Министерство образования и науки Российской Федерации
федеральное государственное автономное образовательное учреждение высшего образования
**«НАЦИОНАЛЬНЫЙ ИССЛЕДОВАТЕЛЬСКИЙ
ТОМСКИЙ ПОЛИТЕХНИЧЕСКИЙ УНИВЕРСИТЕТ»**

Институт **Физико-технический**
Направление подготовки **14.04.02 Ядерные физика и технологии**
Кафедра **Физико-энергетические установки**

МАГИСТЕРСКАЯ ДИССЕРТАЦИЯ

Тема работы
Исследование экономичности энергоблока в переменном режиме после установки дополнительного подогревателя высокого давления

УДК 621.311.25:621.039.577-048.35

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Руководитель

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Томск – 2017 г.

Результаты обучения

Код результата	**Результат обучения** (выпускник должен быть готов)
Профессиональные компетенции	
Р1	Применять глубокие математические, естественнонаучные, социально-экономические и профессиональные знания для теоретических и экспериментальных исследований в области использования ядерной науки и техники
Р2	Способность определять, формулировать и решать междисциплинарные инженерные задачи в ядерной области с использованием профессиональных знаний и современных методов исследования
Р3	Уметь планировать и проводить аналитические, имитационные и экспериментальные исследования в сложных и неопределённых условиях с использованием современных технологий, а также критически оценивать полученные результаты
Р4	Использовать основные и специальные подходы, навыки и методы для идентификации, анализа и решения технических проблем в ядерной науке и технике
Р5	Готовность к эксплуатации современного физического оборудования и приборов, к освоению технологических процессов в ходе подготовки производства новых материалов, приборов, установок и систем
Р6	Способность разрабатывать многовариантные схемы для достижения поставленных производственных целей, с эффективным использованием имеющихся технических средств
Общекультурные компетенции	
Р7	Способность использовать творческий подход для разработки новых идей и методов проектирования объектов ядерного

	комплекса, а также модернизировать и совершенствовать применяемые технологии ядерного производства
Общепрофессиональные компетенции	
Р8	Самостоятельно учиться и непрерывно повышать квалификацию в течение всего периода профессиональной деятельности.
Р9	Активно владеть иностранным языком на уровне, позволяющем работать в иноязычной среде, разрабатывать документацию, презентовать результаты профессиональной деятельности.
Р10	Демонстрировать независимое мышление, эффективно функционировать в командно-ориентированных задачах и обладать высоким уровнем производительности в профессиональной (отраслевой), этической и социальной средах, а также руководить командой, формировать задания, распределять обязанности и нести ответственность за результаты работы

Министерство образования и науки Российской Федерации
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Кафедра **Физико-энергетические установки**

УТВЕРЖДАЮ:
Зав. кафедрой ФЭУ ФТИ
О.Ю. Долматов

_________________ _______________
(Подпись) (Дата)

ЗАДАНИЕ
на выполнение выпускной квалификационной работы

В форме:

магистерской диссертации

(бакалаврской работы, /работы, магистерской диссертации)

Студенту:

Группа	ФИО
0АМ5И	**Игбоануго Принсвилл Фиделис**

Тема работы:

Игбоануго Принсвилл Фиделис	
Утверждена приказом директора (дата, номер)	959/с от 16.02.17

Срок сдачи студентом выполненной работы:	**01 июня 2017 года**

ТЕХНИЧЕСКОЕ ЗАДАНИЕ:

Исходные данные к работе *(наименование объекта исследования или проектирования; производительность или нагрузка; режим работы (непрерывный, периодический, циклический и т. д.); вид сырья или материал изделия; требования к продукту, изделию или процессу; особые требования к особенностям функционирования (эксплуатации) объекта или изделия в плане безопасности эксплуатации, влияния на окружающую среду, энергозатратам; экономический анализ и т. д.).*	Целью обзора является сбор и обобщение информации об опыте совершенствования действующих электрических станций. Объектом исследования является тепловая схема блока К-800-240. Предмет работы – исследование экономичности блока К-800-240 в номинальном и переменном режиме, сравнительный анализ, проработка мероприятия по повышению экономичности.

Перечень подлежащих исследованию, проектированию и разработке вопросов *(аналитический обзор по литературным источникам с целью выяснения достижений мировой науки техники в рассматриваемой области; постановка задачи исследования, проектирования, конструирования; содержание процедуры исследования, проектирования, конструирования; обсуждение результатов выполненной работы; наименование дополнительных разделов, подлежащих разработке; заключение по работе).*	1. Описание тепловой схемы 2. Экономическое обоснование выбранного варианта включения подогревателя высокого давления при снижении нагрузки блока до 60% от номинальной 3. Расчет тепловой схемы в номинальном режиме 4. Расчет тепловой схемы в режиме частичной нагрузки 5. Расчет тепловой схемы с дополнительным ПВД в режиме частичной нагрузки 6. Анализ полученных результатов 7. Разработка конструкции и расчёт ПВД 8. Расчёт трубопровода отборного пара дополнительного ПВД 9. Технико-экономические показатели работы модернизированного блока 10. Анализ других путей повышения экономичности 11. Заключение

Консультанты по разделам выпускной квалификационной работы

(с указанием разделов)

Раздел	Консультант
Финансовый менеджмент	
Социальная ответственность	

Дата выдачи задания на выполнение выпускной квалификационной работы по линейному графику	

Задание выдал руководитель:

Должность	ФИО	Ученая степень, звание	Подпись	Дата
Доцент кафедры АТЭС	**Губин В.Е.**	**к.т.н.**		

Задание принял к исполнению студент:

Группа	ФИО	Подпись	Дата
0АМ5И	**Игбоануго Принсвилл Фиделис**		

Реферат

Выпускная квалификационная работа включает 111 страниц, 21 рисунков, 11 таблиц, 26 источников.

Ключевые слова: подогреватель, расход топлива, экономичность.

Объектом исследования является тепловая схема блока К-800-240.

Цель работы – исследование экономичности блока К-800-240 в номинальном и переменном режиме, сравнительный анализ, проработка мероприятия по повышению экономичности.

Электроэнергетика является основой составляющей развития мировой экономики. В настоящее время в связи с невозможностью снижения эффективности выработки электроэнергии, наиболее экономически приемлемым является не введение новых мощностей, а модернизация существующих блоков. При модернизации необходимо реализовать продление ресурса работы блока, повышение мощности и эффективности, улучшение технико-экономических показателей работы блока. Особенно это актуально при работе энергоблоков в режимах отличных от номинальных

В результате исследования получена положительная динамика показателей экономичности в результате предлагаемого варианта модернизации, а именно включения в тепловую схему дополнительного подогревателя высокого давления.

Область применения: проектирование и эксплуатация новых, модернизация существующих атомных и тепловых электрических станций.

Theme: Improving the economy of the K-800-240 in variable mode by installing an additional high-pressure heater.

Content

Introduction

Electric power industry is the basis component of the global economy. It accumulated enormous wealth, cutting-edge science and technology.

The largest share in the generation of electrical and thermal energy in Russia accounts for thermal power stations (up to 80%).

Thermal Power Plants (TPP) are a set of electrical facilities, linked by a steam-water, air and gas paths. The main type of thermal power plants is steam-turbine power plant, which can run on any fuel, have a very large capacity and be constructed where there is demand for electricity.

Power units applied at thermal power plants can be divided into two groups: the condensing power units that produce (and release to consumer) only electricity and district heating power units that produce (and release to consumer) electricity and heat.

TPP reliably operate for decades at the temperatures and pressures close to the maximum permissible for the materials from which their equipment is made. The negative impact of thermal power plants on the environment where due attention is given can be minimized even when using contaminated fuel.

A constant stimulus for improvement of TPP cycles, circuits and equipment, till the present day, is to reduction of production costs through fuel savings, increase of reserve maintenance campaigns and decrease of maintenance costs. In the market conditions necessary for these innovations are introduced only in those cases where they increase competitiveness and funding spent can be returned within a reasonable time. Therefore, at this time, due to insufficient investment in the energy sector, as well as inability to reduce the efficiency of power generation, the most economically feasible decision is not the introduction of new facilities but modernization of existing units. While upgrading it is necessary to extend the service life of the unit, enhance power and efficiency, improve technical and economic performance of the unit.

Within the power units non-equilibrium processes of energy transformation are performed: chemical energy of fuel non-equilibrium (with losses) is transformed into heat and then into mechanical work, which is converted into electricity. To perform a continuous process of energy transformation in the form of heat into mechanical work, it is necessary to have a "hot" and "cold" thermodynamic heat sources and use the working fluid in the power unit. The last in a circular thermodynamic process (cycle) changes its state as a result of the work and the input (or output) of energy in the form of heat.

1. Description of the thermal circuit

Unit consists of a once-through boiler and a single-shaft condensing unit of supercritical steam parameters with a single-stage gas reheats (Figure 1).

The turbine has five cylinders. Fresh steam with parameters 23.54 MPa and 560 ° C through the stopper and the group fed the control valves in the pressure cylinder (HPC), and then at 3.66 MPa and 299 ° C is sent to the reheater of the boiler. After reheating steam of 3.25 MPa and 540 ° C is fed through a shut-off and control valves in the middle of the double-beam medium pressure cylinder (MPC) of the IPC at 0.20 MPa and 197 ° C steam is given in three double-flow low-pressure cylinder (LPC) having five stages in each stream. From LPC pairs sent to the three capacitors 800 MCC-2. The final pressure in the condenser is 3.5 kPa. Turbine speed n = 50 s-1.

The turbine has eight regenerative steam extraction, two - from HPC, four - from the CSD and two - from LPC. Thermal installation scheme provides the heating of the feed water to 274 ° C, which is produced in glands heaters, four low pressure surface heaters, deaerator-DP-1 1600m (2 pcs.), (At 0.7 MPa) and the three high pressure heaters (CVP). To supply the two pumps steam generators each half performance, developing pressure of 35 MPa, are installed. The drive pumps used condensing steam turbine power Pe = 15 MW each, fueled by steam from the IPC main turbine steam with parameters P = 1.6 MPa, t = 440 ° C.

There are «cold» feed water lines to bypass the group of LDPE and line bypass of LDPE automatic protection system. There is a bypass line HDPE surface type main condensate.

The booster pump works through a gearbox, reducing the rotational speed.

The steam in high pressure cylinder is fed into the middle portion of the cylinder and passes sequentially single-row regulatory stage and five stages of the active type, which apertures are located in the inner housing. The steam is then rotated 180 °, washes the inner casing and passes six more stages, the apertures which are located in the outer housing of CVP. All the blades of the turbine, except regulating stage are variable in height profile.

From the reheater steam is supplied through two pipes to two boxes of MPC isolation valves installed on both sides of the turbine from which steam enters the box four control valves, placed directly on the IPC. Double-flow IPC with nine stages s in each stream. After MPC steam is sent to the three low-pressure cylinders. LPC of turbine has the last blade 960 mm long.

Drains LDPE blend cascaded to deaerator and drains IPA 4, 3 and 2 to the mixing point with help of pump. Coolers drains of steam seals and the IPA 1 enter the condenser.

The added water that makes up the loss of the water-steam cycle of the wturbine goes to the condenser.

2. Economic assessment for the option of included high pressure heater with load reduction up to 60% of the nominal

Installation of an optional high pressure heater is carried out in order to obtain more power in a reduced load conditions, namely better use of regeneration turbine system, thereby the specific fuel consumption reduces and efficiency of the heat flow, electricity production and supply increase. As a result high pressure heater installation the efficiency of the entire unit was increased. Selection of steam for high pressure heater is made in HPC after the fifth stage.

2.1. The block with an additional (fourth) high pressure heater

2.1.1. Economic assessment for the option of the additional heater is reduced to determining the value of the reduced costs:

$$З_3 = E_н \cdot \sum K_3 + \sum И_3;$$

Where: $E_н = 0{,}15 -$ efficiency factor equal to the average annual interest rate;

$\sum K_3 -$ the amount of capital expenditures in the block with the additional high pressure heater; $\sum И_3 -$ the sum of annual costs of the option with the additional high pressure heater.

Price is given in prices of 2017 year.

2.1.2. Capital costs for the installation of an additional high pressure heater

High pressure heater cost is the cost of the heater, as well as the cost of its installation. High pressure heater cost in taken on the basis of (Table 1. 5.21). The costs of installation, lead-in steam piping and piping and commensurate with the cost of the heater:

$$К_{ПВД} = К_{нов.} + К_{монт.} = 118{,}3 \cdot 10^3 + 118{,}3 \cdot 10^3 = 236{,}6 \cdot 10^3\, руб.$$

2.1.3. The total capital cost of the unit with an additional high pressure heater

Thus, the capital cost of the unit with the additional high pressure heater is defined by capital expenditure on additional high pressure heater and its installation:

$$\sum К_3 = К_{ПВД} = 236{,}6 \cdot 10^3\, руб.$$

2.1.4. Annual costs

The fuel component costs:

$$И_{т3} = B_{год3} \cdot \frac{7000}{Q_р^н} \cdot Ц_т = 10{,}753 \cdot 10^5 \cdot \frac{7000}{7760{,}44} \cdot 19{,}5 = 18913{,}676 \cdot 10^3\, руб.\ ;$$

where: $B_{год3}$ – annual consumption of fuel per unit;

$$B_{год3} = b_Э^{отп} \cdot T_{год} \cdot N_3 = 0{,}31753 \cdot 7000 \cdot 483{,}79 = 10{,}753 \cdot 10^5\, т.у.т/год;$$

$Ц_т = 19{,}5 руб./т.у.т$ – the price per ton of fuel equivalent (9);

$T_{год} = 7000ч$ – the number of work hours of block per year;

$Q_р^н = 7760{,}44 ккал/кг$ – lower calorific value of fuel equivalent.

2.1.5. The given unit costs with the additional high pressure heater

$$З_{пр3} = \sum К_3 \cdot Е_н + И_{т3} = \left(236{,}6 \cdot 0{,}15 + 18913{,}676\right) \cdot 10^3 = 18949{,}167 \cdot 10^3 \; руб / год.$$

2.2. Unit without additional high pressure heater

Costs for unit without additional high pressure heater comprise fuel costs:

$$З_2 = \sum И_2;$$

The fuel component of costs:

$$И_{т2} = В_{год2} \cdot \frac{7000}{Q_р^н} \cdot Ц_т = 11{,}381 \cdot 10^5 \cdot \frac{7000}{7760{,}44} \cdot 19{,}5 = 20018{,}3 \cdot 10^3 \; руб. \; ;$$

$$В'_{год2} = b_{Э2}^{отп} \cdot T_{год} = 0{,}33607 \cdot 7000 \cdot 482{,}28 = 11{,}345 \cdot 10^5 \; т.у.т / год;$$

Reduced generation of electricity:

$$В_{год}^{недов} = b_{Э}^{отп} \cdot T_{год} = 336{,}07 \cdot 7000 \cdot \left(483{,}79 - 482{,}28\right) = 0{,}036 \cdot 10^5 \; т.у.т / год.$$

$$В_{год2} = В'_{год2} + В_{год2}^{недов} = (11{,}345 + 0{,}036) \cdot 10^5 = 11{,}381 \cdot 10^5 \; т.у.т / год.$$

2.3. Money saving

Money saving depends on fuel costs and the capital costs of high pressure heater:

$$\Delta З = З_{пр2} - З_{пр3} = И_{т2} - И_{т3} - К_{ПВД} =$$
$$= 20018{,}3 \cdot 10^3 - 18949{,}167 \cdot 10^3 = 1069{,}133 \cdot 10^3 \; руб. / год.$$

2.4. Calculation of pay back period

Depreciation deductions:

$$А_г = 0{,}12 \cdot К = 0{,}12 \cdot 236{,}6 \cdot 10^3 = 28{,}392 \cdot 10^3 \ руб/год.$$

Annual contributions provided by the bank for a loan:

$$К_р = 0{,}2 \cdot К = 0{,}2 \cdot 236{,}6 \cdot 10^3 = 47{,}32 \cdot 10^3 \ руб/год;$$

where: $i = 0{,}2$ – the average annual interest rate of the bank.

Money flow:

$$Д_п = \Delta З - А_г - К_р = 1069{,}133 \cdot 10^3 - 28{,}392 \cdot 10^3 - 47{,}32 \cdot 10^3 = 993{,}421 \cdot 10^3 \ руб./год.$$

Payback period:

$$Т_{ок} = -\frac{\lg \cdot \left(1 - \frac{К_{ПВД} + К_р}{Д_п} \cdot i\right)}{\lg(1+i)} = -\frac{\lg \cdot \left(1 - \frac{236{,}6 \cdot 10^3 + 47{,}32 \cdot 10^3}{993{,}421 \cdot 10^3} \cdot 0{,}2\right)}{\lg(1+0{,}2)} = 0{,}41 года.$$

Thus, the payback period is $Т_{ок} = \frac{0{,}41 \cdot 365}{30} = 4{,}9 месяца.$

3. Calculation of the thermal scheme in the nominal mode

The main purpose of calculating the basic thermal scheme (PTS) of the power unit is to determine the technical characteristics of the heat equipment (steam, water and fuel consumption) and the energy parameters of the block and its parts (efficiency and specific heat and fuel consumption).

The first step in the calculation is to determine the state of water vapor in the turbine stages. To do this, the process of steam expansion in the turbine in the h-s diagram is constructed.

The second stage of the basic thermal scheme calculation is to compile a free table of steam and water parameters in the turbine unit.

The third stage of the basic thermal scheme is to draw up the correlations between the material and thermal balances of steam, condensate and water flows.

The fourth stage of the calculation includes the compilation, sequential and joint solution of the equations of thermal and material balances of the basic thermal scheme heat exchangers in order to determine their relative steam flow.

3.1. Initial data (6)

Type of turbine: K-800-240.

Type of boiler: П - 67 (ТГМП-204).

Initial steam pressure: $P_0 = 23{,}54 МПа.$

Temperature of fresh steam: $t_0 = 560°C.$

Steam temperature after intermediate superheater: $t_{ПП} = 540°C.$

Steam flow rate per the turbine unit: G0н = 2300 t / h.

Turbine condenser pressure: $P_0 = 0{,}00345 МПа.$

Table 3.1. Types of heaters (1).

Heater Number	Type of heater	F
П1	ПВ–1600–380–66	1650
П2	ПВ–2100–380–40	2135
ТП ПН	–	–
П3	ПВ–1600–380–17	1560
Д	ДП–1600–1	–
П4	ПН–2400–32–7–I	2330
П5	ПН–2200–32–7–II	2233
П6	ПН–1500–32–7–III	1550
ОЭ	ЭВ–4–1100	–
П7	ПН–1600–32–7–IV	1630
ОУ	ПС–220	–

The basic thermal scheme of the turbine installation is shown in Figure 3.1.

The process of steam expansion in the turbine system in the nominal mode is shown in Figure 3.2.

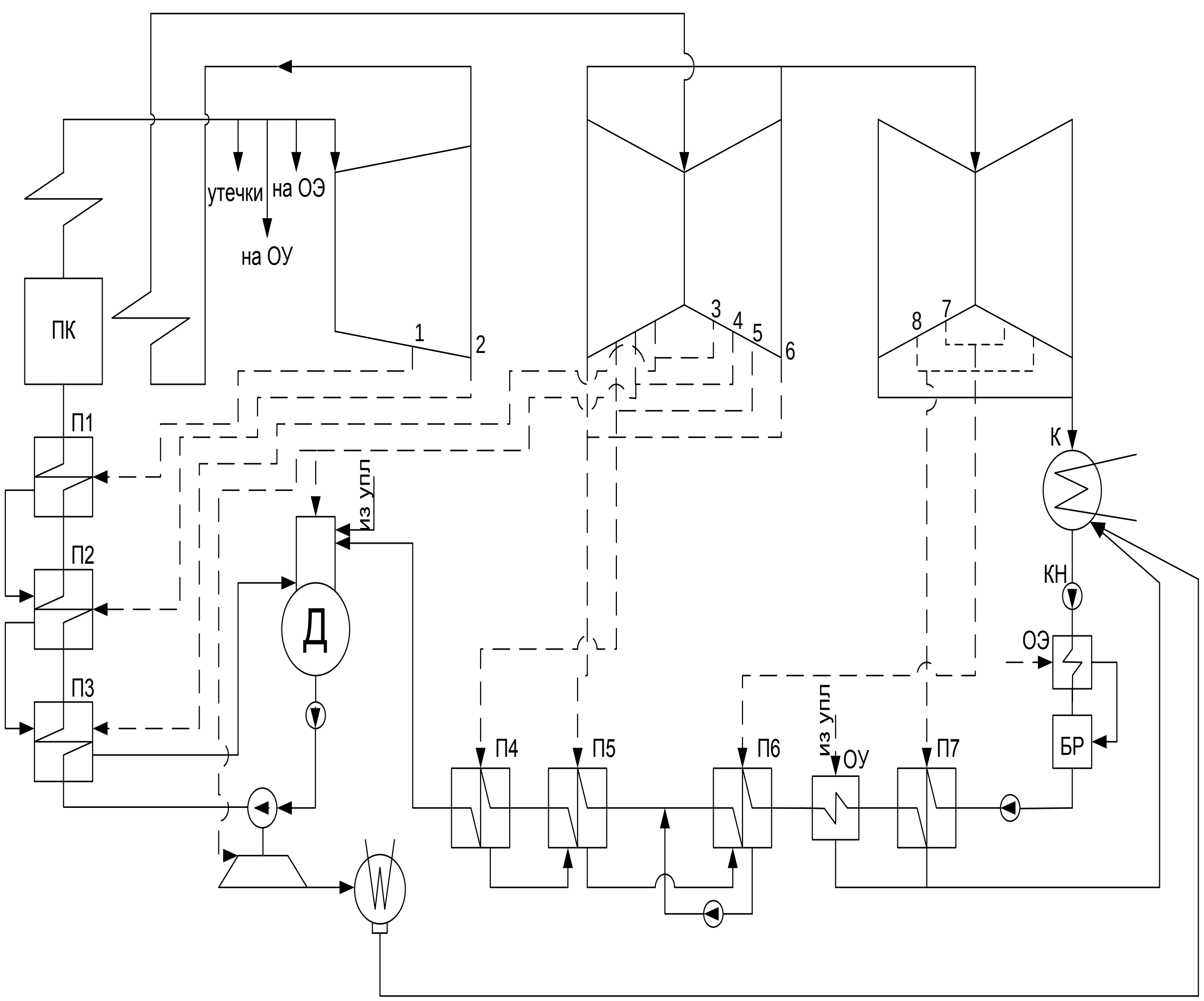


Figure 3.1. The basic thermal scheme of the turbine unit K-800-240.

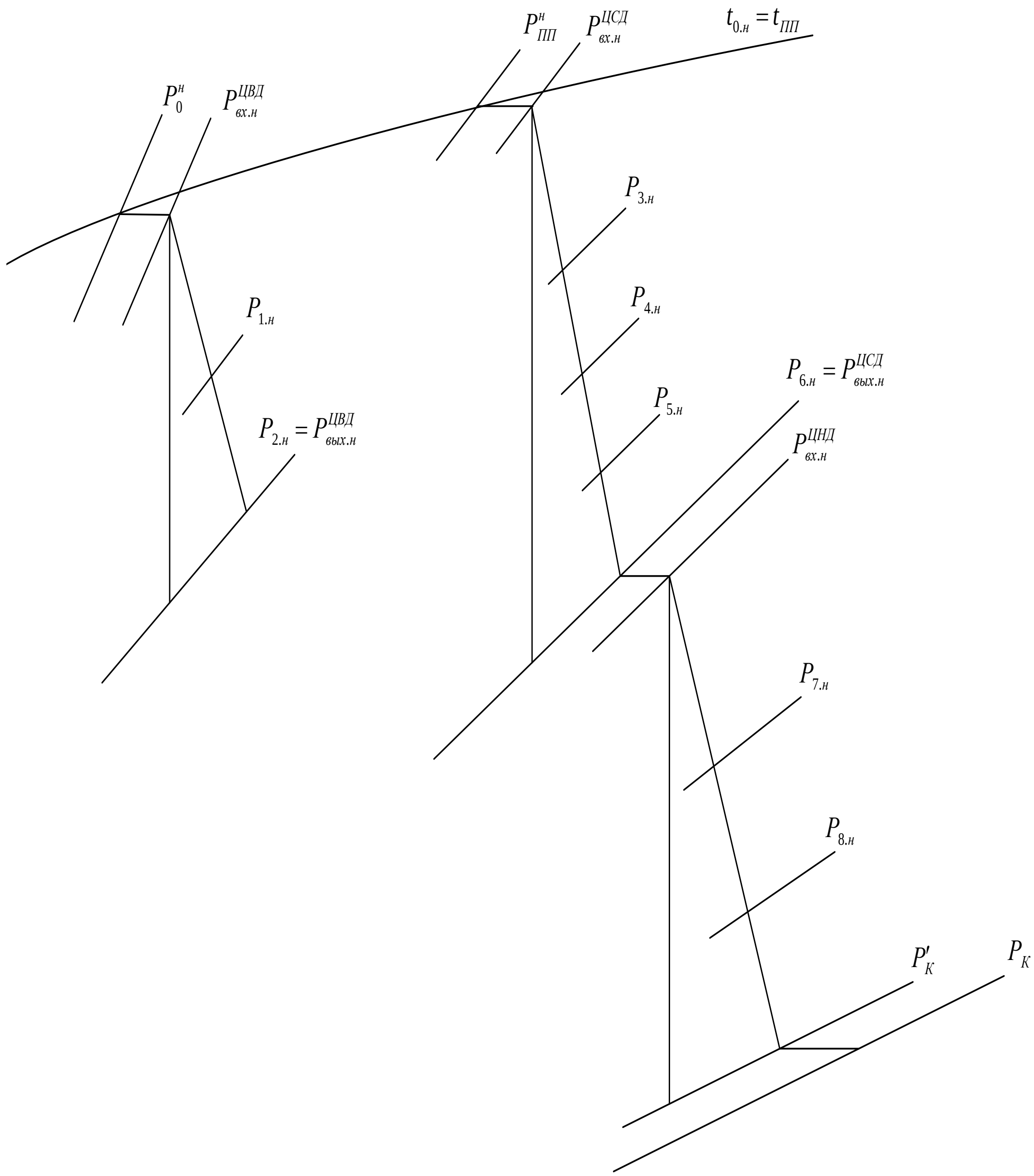


Figure 3.2. The process of steam expansion in a turbine in the nominal mode

Table 3.2. Parameters of steam, condensate and water in nominal conditions.

Elements of the thermal scheme	Selection number	Steam in the turbine (selections)				Steam in Heater	Drainage of heating steam		Feed water, main condensate			Heat capacity	Underheating
		$P_{отб}$	$t_{отб}$	$h_{отб}$	$G_{отб}$	$P_п$	t_s	h´	$t_{пв}$	$P_{пв}$	$h_{пв}$	C_p	θ
Ед. изм.	-	МПа	ºС	$\frac{кДж}{кг}$	кг/с	МПа	ºС	$\frac{кДж}{кг}$	ºС	МПа	$\frac{кДж}{кг}$	$\frac{кДж}{кг \cdot К}$	ºС
П1	1	5,97	362	3076	48,61	5,59	271,5	1192,9	266,5	33,0	1155,3	4,498	5
П2	2	3,66	299	2967	60,55	3,41	242,7	1050,2	237,7	33,0	1029,0	4,322	5
П3	3	1,93	469	3402	30,27	1,85	198,8	846,9	193,8	33,0	838,3	4,267	5
ТП ПН	4				29,18				173,16	-	751,1		
Д	4	0,90	368	3194	5,53	0,70	165,0	697,1	165,0	0,70	697,1	4,353	
П4	5	0,47	288	3040	24,72	0,423	154,6	652,7	153,6	1,40	650,6	4,258	1
П5	6	0,20	207	2860	26,11	0,179	128,8	541,1	126,8	1,40	539,4	4,213	2
П6	7	0,084	141	2716	30,55	0,076	101,0	423,3	99,0	1,40	417,8	4,181	2
П7	8	0,024	61	2538	24,16	0,022	59,5	249,2	58,5	1,40	246,2	4,177	1

3.2. Heat circuit calculation

3.2.1. Balance of steam and water

Relative feed water flow rate:

$$\alpha_{пв} = \alpha_{ту} + \alpha_{ут} + \alpha_{упл} + \alpha_{э};$$

Where: $\alpha_{ту}$ is the relative steam flow per the turbine unit, we take

$$\alpha_{ту} = 1;$$

$\alpha_{ут}$ = (0,005 ÷ 0,012) - relative leakage rate, we accept

$$\alpha_{ут} = 0{,}01;$$

$\alpha_{упл}$ = (0,02 ÷ 0,04) - relative steam consumption from the turbine seals, we accept $\alpha_{упл}$ = 0,028;

$\alpha_{э}$ = (0,002 ÷ 0,003) - relative steam consumption for ejectors, we accept $\alpha_{э}$ = 0,002;

$$\alpha_{дв} = \alpha_{ут};$$

$$\alpha_{пв} = 1 + 0{,}01 + 0{,}028 + 0{,}002 = 1{,}04.$$

3.2.2. Preliminary estimation of steam consumption per turbine

$$G_0' = K_p \left(\frac{N_э \cdot 10^3}{H_i \cdot \eta_м \cdot \eta_г} \right);$$

Where: $N_э$ - the given electric power in MW; H_i is the actual heat transfer of the turbine in kJ / kg; $\eta_м$, $\eta_г$ - mechanical efficiency and generator efficiency (0,98 ÷ 0,995 can be adopted); k_p - coefficient of regeneration, it depends on many factors and is in the range from 1.15 to 1.4.

I accept: $\eta_м$ = 0,98; $\eta_г$ = 0,98; k_p = 1,306;

$$G_0' = 1{,}306\left(\frac{800\cdot 10^3}{1638{,}6\cdot 0{,}98\cdot 0{,}98}\right) = 663{,}91\frac{кг}{с} = 2390\frac{т}{ч}.$$

The actual heat drops of the turbine for the cylinders:

$$H_i^{ЦВД} = h_{вх.н}^{ЦВД} - h_2 = 3385{,}3 - 2967 = 418{,}3\frac{кДж}{кг};$$

$$H_i^{ЦСД} = h_{вх.н}^{ЦСД} - h_6 = 3543{,}7 - 2860 = 683{,}7\frac{кДж}{кг};$$

$$H_i^{ЦНД} = h_{вх.н}^{ЦНД} - h_к' = 2860 - 2323{,}4 = 536{,}6\frac{кДж}{кг};$$

$$H_i = H_i^{ЦВД} + H_i^{ЦСД} + H_i^{ЦНД} = 418{,}3 + 683{,}7 + 536{,}6 = 1638{,}6\frac{кДж}{кг}.$$

3.2.3. Calculation of regenerative heaters

Parameters of the working fluid in the nodes of the scheme are determined using the program WaterSteamPro.

Heating water in the MA and OS is: $\Delta t_{оэ} = \Delta t_{оу} = 3°C$

Pressure loss in the steam pipeline:

$P_п = (0{,}92 \div 0{,}95)\cdot P_{отб}$ - for heaters;

$P_п = P_д$ - for the deaerator.

The water temperature behind the heater is determined by taking into account the underheating:

$$t_{в.вых}^{П.i} = t_{S\cdot_i} - \Theta_{П.i},\ ^oC.$$

We ask inadequate heating in heaters:

$\Theta_{ПВД} = 2 - 5^oC;\ \ \Theta_{ПНД} = 1 - 3^oC;\ \ \Theta_д = 0^oC.$

The heat capacity of the main condensate and feed water is determined by:

$$C_в = f\left(P_{ок}\left(P_{пв}\right), t_{в.вх.}\right);\quad \frac{кДж}{кг\cdot K}.$$

Based on known pressures and temperatures, we determine the enthalpy of selection:

$h_{отб.i} = f\left(P_{отб.i}, t_{отб.i}\right).$

The vapor pressure in the heater is determined taking into account the hydraulic losses in the pipeline selection:

$$P_{П.i} = 0{,}95 \cdot P_{отб.i}\text{, МПа.}$$

The temperature and enthalpy of drainage of heaters is determined by:

$$t_{др.i} = t_S = f\left(P_{П.i}\right),\ {}^{o}\text{C};$$

$$h' = f\left(P_{П.i}\right) .$$

Pressure of feed water and main condensate (6. 13-2).

The enthalpy at the outlet of the heater is determined by:

$h^{П.i}_{в.вых} = f\left(t^{П.i}_{в.вых}; P_{ок}\left(P_{пв}\right)\right)$.

Condensate temperature at the condenser outlet:

$t_к = f\left(P^н_к\right) = 25{,}9°C$

3.2.3.1. High-pressure heaters

For the calculation of high-pressure heater a system of equations for the steam cooler, the heater itself and the drainage cooler is compiled (Fig. 3):

HPH 1: $\begin{cases} a_1 \cdot \left(h_1 - h^1_{по}\right) \cdot \eta = a_{пв} \cdot \left(h^{по1}_{пв} - h^1_{пв}\right); \\ a_1 \cdot \left(h^1_{по} - h^1_{др}\right) \cdot \eta = a_{пв} \cdot \left(h^1_{пв} - h^{по2}_{пв}\right). \end{cases}$

HPH 2: $\begin{cases} a_2 \cdot \left(h_2 - h^2_{по}\right) \cdot \eta = a_{пв} \cdot \left(h^{по2}_{пв} - h^2_{пв}\right); \\ a_2 \cdot \left(h^2_{по} - h^2_{др}\right) \cdot \eta + a_1 \cdot \left(h^1_{др} - h^2_{др}\right) \cdot \eta = a_{пв} \cdot \left(h^2_{пв} - h^{по3}_{пв}\right). \end{cases}$

HPH 3: $\begin{cases} a_3 \cdot \left(h_3 - h^3_{по}\right) \cdot \eta = a_{пв} \cdot \left(h^{по3}_{пв} - h^3_{пв}\right); \\ a_3 \cdot \left(h^3_{по} - h^3_{др}\right) \cdot \eta + \left(a_1 + a_2\right) \cdot \left(h^2_{др} - h^3_{др}\right) \cdot \eta = a_{пв} \cdot \left(h^2_{пв} - h^{по3}_{пв}\right). \end{cases}$

Where: a_i – the relative consumption of steam going to i heater;

h_i – enthalpy of the selected steam going to the i heater;

$h_{по}^{i}$ – enthalpy of steam behind i heater;

$h_{др}^{i}$ – Enthalpy of drainage for i drain cooler;

$h_{пв}^{i}$ – Enthalpy of water for i proper heater;

$h_{пв}^{поi}$ – Enthalpy of water behind i desuperheater;

$h_{пв}^{пн}$ – Enthalpy of water behind feed pumps;

$\eta = 0{,}98$ – Efficiency of surface heater, we accept.

As a result of solving these systems of equations, the relative flow of steam for heaters and the enthalpy of water behind the steam coolers are determined:

$$a_1 = 0{,}07126;\ \ h_{пв}^{по1} = 1198{,}37\ кДж/кг;$$
$$a_2 = 0{,}07181;\ \ h_{пв}^{по2} = 1053{,}96\ кДж/кг;$$
$$a_3 = 0{,}06052;\ \ h_{пв}^{по3} = 851{,}16\ кДж/кг.$$

3.2.3.2. Feeding turbo pump

Temperature increase of feed water in the feeding pump:

$$\Delta h_{ТП} = \frac{\upsilon' \cdot (P_{ТН} - P_{д}) \cdot 10^3}{\eta_{нi}},$$

Where: υ' – specific volume of water with the temperature at the outlet from the deaerator;

$\eta_{нi}$ – "Internal" (hydraulic) efficiency of the pump (6. 13-2);

$$\Delta h_{ТП} = \frac{0{,}001108 \cdot (33{,}0 - 0{,}7) \cdot 10^3}{0{,}85} = 42{,}1 \frac{кДж}{кг}.$$

Enthalpy of feedwater behind the pump:

$$h_{пв.н} = h_S^{д} + \Delta h_{ТП} = 697{,}1 + 42{,}1 = 739{,}2 \frac{кДж}{кг}.$$

Water parameters behind the feed pump:

$$P_{ТП} = 33{,}0 МПа; \;\; t_{пв.н} = 170{,}5°C.$$

The proportion of steam extraction for the turbo drive of the feed pump:

$$a_{ТП} = \frac{a_{пв} \cdot h_н^a}{H_i^{ТП} \cdot \eta_н \cdot \eta_м} = \frac{1{,}04 \cdot 35{,}53}{954{,}5 \cdot 0{,}833 \cdot 0{,}98} = 0{,}04742;$$

Where: $h_н^a = \upsilon' \cdot \left(p_н - p_в\right) \cdot 10^3 = 0{,}001108 \cdot \left(33 - 0{,}7\right) \cdot 10^3 = 35{,}53 кДж/кг$ – adiabatic compression work in the pump;

$H_{ТП} = 3194 - 2239{,}5 = 954{,}5 кДж/кг$ – The actual heat drop of steam in the drive turbine of the feed pump;

$\eta_н = \eta_{нi} \cdot \eta'_н = 0{,}85 \cdot 0{,}98 = 0{,}833$ – Full efficiency of the pump, taking into account the volume and mechanical losses (6. 13-4);

$\eta_м$ – Mechanical efficiency of the drive turbine (6. 13-4).

3.2.3.3. Deaerator of feed water

Material balance of the deaerator:

$$a_{др3} + a_д + a_{ок} = a_{пв} + a_{выпар};$$

$$a_{ок} = 0{,}83841 - a_д;$$

Where: $a_{др3} = 0{,}20359$ – relative drainage rate to deaerator;

$a_{д}$ – Relative steam flow to the deaerator;

$a_{выпар} = 0{,}002$ – The relative flow rate of the vapor from the deaerator is equal to the steam flow to the ejector;

$a_{ок}$ – Relative consumption of the main condensate.

Equation of thermal balance of the deaerator:

$$a_{др3} \cdot h_{др3} + a_{д} \cdot h_4 + a_{ок} \cdot h_{ок4} = a_{пв} \cdot h_S^{д} + a_{выпар} \cdot h_{выпар};$$

Where: $h_{выпар} = 2761{,}7\ кДж/кг$ – enthalpy vapors;

$$0{,}20359 \cdot 846{,}9 + a_{д} \cdot 3194 + a_{ок} \cdot 650{,}6 = 1{,}04 \cdot 697{,}1 + 0{,}002 \cdot 2761{,}7.$$

Solving together the equations of material and heat balances, we determine the relative steam consumption in the deaerator and the relative consumption of the main condensate:

$$a_{д} = 0{,}01666; \quad a_{ок} = 0{,}82175.$$

The share of selection for the turbo and deaerator:

$$a_4 = a_{ТП} + a_{д} = 0{,}04742 + 0{,}01666 = 0{,}06408.$$

3.2.3.4. Low-pressure heaters

The relative steam consumption for Low-pressure heater 4:

$$a_5 = \frac{a_{ок} \cdot (h_{ок4} - h_{ок5})}{(h_5 - h_{др4}) \cdot \eta} = \frac{0{,}82175 \cdot (650{,}6 - 539{,}4)}{(3040 - 652{,}7) \cdot 0{,}98} = 0{,}03953.$$

Calculation of the LPH group:

We compose a system of equations for low pressure heaters 5 and 6:

$$\begin{cases} a_6 \cdot \left(h_6 - h_{\partial p5}\right) \cdot \eta + a_5 \cdot \left(h_{\partial p4} - h_{\partial p5}\right) \cdot \eta = a_{o\kappa} \cdot \left(h_{o\kappa 5} - h_{в.вх5}\right); \\ a_7 \cdot \left(h_7 - h_{\partial p6}\right) \cdot \eta + a_{\partial p6} \cdot \left(h_{\partial p5} - h_{\partial p6}\right) \cdot \eta = a^1_{o\kappa} \cdot \left(h_{o\kappa 6} - h_{oy}\right); \end{cases}$$

$$\begin{cases} a^1_{o\kappa} + a_5 + a_6 + a_7 = a_{o\kappa}; \\ a^1_{o\kappa} \cdot h_{oy} + \left(a_5 + a_6 + a_7\right) \cdot h_{\partial p6} = a_{o\kappa} \cdot h_{в.вх5}; \end{cases}$$

Where: $h_{oy} = 258{,}57\,кДж/кг$ – enthalpy of condensate behind the seal cooler.

As a result of the joint solution of these equations, we determine the relative costs of steam for low-pressure heaters and the enthalpy of water beyond the mixing point:

$a_6 = 0{,}02392;\ a_7 = 0{,}02458;\ a^1_{o\kappa} = 0{,}73372;$

$$h_{в.вх5} = 418{,}67\,кДж/кг.$$

The relative consumption of LPH 7:

$$a_8 = \frac{a^1_{o\kappa} \cdot \left(h_{o\kappa 7} - h_{oэ}\right)}{\left(h_8 - h_{\partial p7}\right) \cdot \eta} = \frac{0{,}73372 \cdot \left(246{,}2 - 117{,}8\right)}{\left(2538 - 249{,}2\right) \cdot 0{,}98} = 0{,}05758;$$

Where: $h_{oэ} = 117{,}85\,кДж/кг$ – the enthalpy behind the cooler of the ejectors.

3.2.4. Material balance of steam consumption in the condenser

Steam flow to the condenser by the material balance of the turbine:

$$a^T_\kappa = 1 - \sum a_i = 1 - (0{,}07126 + 0{,}07181 + 0{,}06052 + 0{,}06408 +$$

$$+ 0{,}03953 + 0{,}02392 + 0{,}02458 + 0{,}05758) = 0{,}58672.$$

Steam flow to the condenser by the material balance of the condenser:

$$a^K_\kappa = a^1_{o\kappa} - a_8 - a_{упл} - a_{ут} - a_э - a_{ТП} - a_{выпар} =$$

$$= 0{,}73372 - 0{,}05758 - 0{,}028 - 0{,}01 - 0{,}002 - 0{,}04742 - 0{,}002 = 0{,}58672.$$

Relative calculation error:

$$\delta a_{к} = \frac{a_{к}^{T} - a_{к}^{K}}{a_{к}^{K}} \cdot 100\% = \frac{0{,}58672 - 0{,}58672}{0{,}58672} \cdot 100\% = 0\%.$$

3.3. Calculation of technical and economic indicators

The underdevelopment of turbine power is determined by the formula:

$$y_i = \frac{h_{отб.i} - h_{к} + \Delta h_{пп}}{H_i}$$; Where

$\Delta h_{ПП} = h_{вх.н}^{ЦСД} - h_2 = 3543{,}7 - 2967 = 576{,}7 \frac{кДж}{кг}$ – Increase of enthalpy in the superheater.

Relative steam fows for selections are transferred into absolute values according to the formula:

$$G_{отб.i} = G'_0 \cdot \alpha_i$$

Table 3.3.

№	$G_{отб.i}$	α_i	y_i	$\alpha_i \cdot y_i$
1	47,31	0,07126	0,79623	0,05674
2	47,68	0,07181	0,72971	0,05240
3	40,18	0,06052	0,64323	0,03893
4 ТП	31,48	0,04742	0,51629	0,02449
4 Д	11,06	0,01666	0,51629	0,00857
5	26,24	0,03953	0,42231	0,01669

6	15,88	0,02392	0,31246	0,00747
7	16,32	0,02458	0,22458	0,00552
8	38,23	0,05758	0,11595	0,00688

Checking the steam flow rate:

$$G_0 = \frac{N_э \cdot 10^3}{H_i \cdot \eta_м \cdot \eta_г \cdot (1-\sum \alpha_j \cdot y_j)};$$

$$G_0 = \frac{800 \cdot 10^3}{1638{,}6 \cdot 0{,}98 \cdot 0{,}982 \cdot (1-0{,}23649)} = 664{,}33\frac{кг}{с};$$

The error of steam consumption per turbine is determined by the formula:

$\delta\ G_0 = \left(\frac{664{,}33-663{,}91}{664{,}33}\right) \cdot 100\% = 0{,}07\%$ <2% material calculation is performed correctly.

Consumption of feedwater through the steam generator:

$$G_{пв} = G_0 \cdot a_{пв} = 664{,}33 \cdot 1{,}04 = 690{,}90\, {кг}/{с}.$$

Steam consumption for overheating:

$$G_{пп} = G_0 - G_1 - G_2 = 664{,}33 - 47{,}31 - 47{,}68 = 569{,}34\, {кг}/{с}.$$

Heat, supplied in the steam generator:

$$Q_{ПГ} = G_{пв} \cdot \left(h_0^{ПГ} - h_{пв}\right) + \Delta h_{пп} \cdot G_{пп};$$

$$Q_{ПГ} = \frac{690{,}90 \cdot (3384{,}8 - 1155{,}3) + (3543{,}7 - 2967) \cdot 569{,}34}{10^3} = 1868{,}70 МВт.$$

Heat supplied to the turbine unit:

$$Q_{ТУ} = G_0 \cdot \left[\left(\alpha_{ту} + \alpha_{упл}\right) \cdot \left(h_0 - h_{пв}\right) + \alpha_{дв} \cdot \left(h_{дв} - h_{пв}\right) + \Delta h_{пп} \cdot \alpha_{пп}\right];$$

Where: $t_{дв} = 15°C$;

$$h_{дв} = 15 \cdot 4{,}19 = 62{,}85 \frac{кДж}{кг}.$$

$$Q_{ТУ} = \frac{664{,}33 \cdot (3385{,}3 - 1155{,}3) + (3543{,}7 - 2967) \cdot 569{,}34}{10^3} = 1809{,}79 МВт.$$

Power consumed by the turbo drive:

$$N_{ТП} = \frac{\alpha_{пв} \cdot G_0 \cdot \Delta h_{ПН}}{\eta_{ТП}} = \frac{1{,}04 \cdot 664{,}33 \cdot 42{,}1}{0{,}833} \cdot 10^{-3} = 34{,}90 \quad МВт;$$

Where: $\eta_{ТП} = 0{,}833$- The efficiency of the turbo drive (6. 13-6).

The enthalpy at the exit from the PN:

$$h_{ПН} = f\left(P_{пв}, t_{ПН}\right) = 751{,}13 \quad \frac{кДж}{кг}$$

.

Absolute electrical efficiency of the turbine unit:

$$\eta_{ТУ}^{э} = \frac{N_э + N_{ТП}}{Q_{ТУ}} = \frac{800 + 34{,}9}{1809{,}79} = 0{,}461.$$

Heat transfer efficiency:

$$\eta_{ТР} = \frac{Q_{ТУ}}{Q_{ПГ}} = \frac{1809{,}79}{1868{,}70} = 0{,}968.$$

Effective efficiency for electricity supply, taking into account generation for its own needs (net):

$$\eta_e^э = \eta_{ТУ}^э \cdot \eta_{ТР} \cdot \eta_{ПГ} \cdot (1 - k_{сн}) \; ;$$

Where: $\eta_{ПГ} = 0{,}925 -$ efficiency of SG (steam generator), we accept;

$k_{сн} = 0{,}03 -$ A factor that takes into account the consumption of electricity for own its needs, we accept;

$$\eta_e^э = 0{,}461 \cdot 0{,}968 \cdot 0{,}925 \cdot (1 - 0{,}03) = 0{,}401.$$

Conventional fuel consumption per unit:

$$B_у = \frac{Q_{ПГ}}{\eta_{ПГ} \cdot Q_р^н} = \frac{1868700}{0{,}925 \cdot 32491{,}4} = 62{,}18\frac{кг}{с};$$

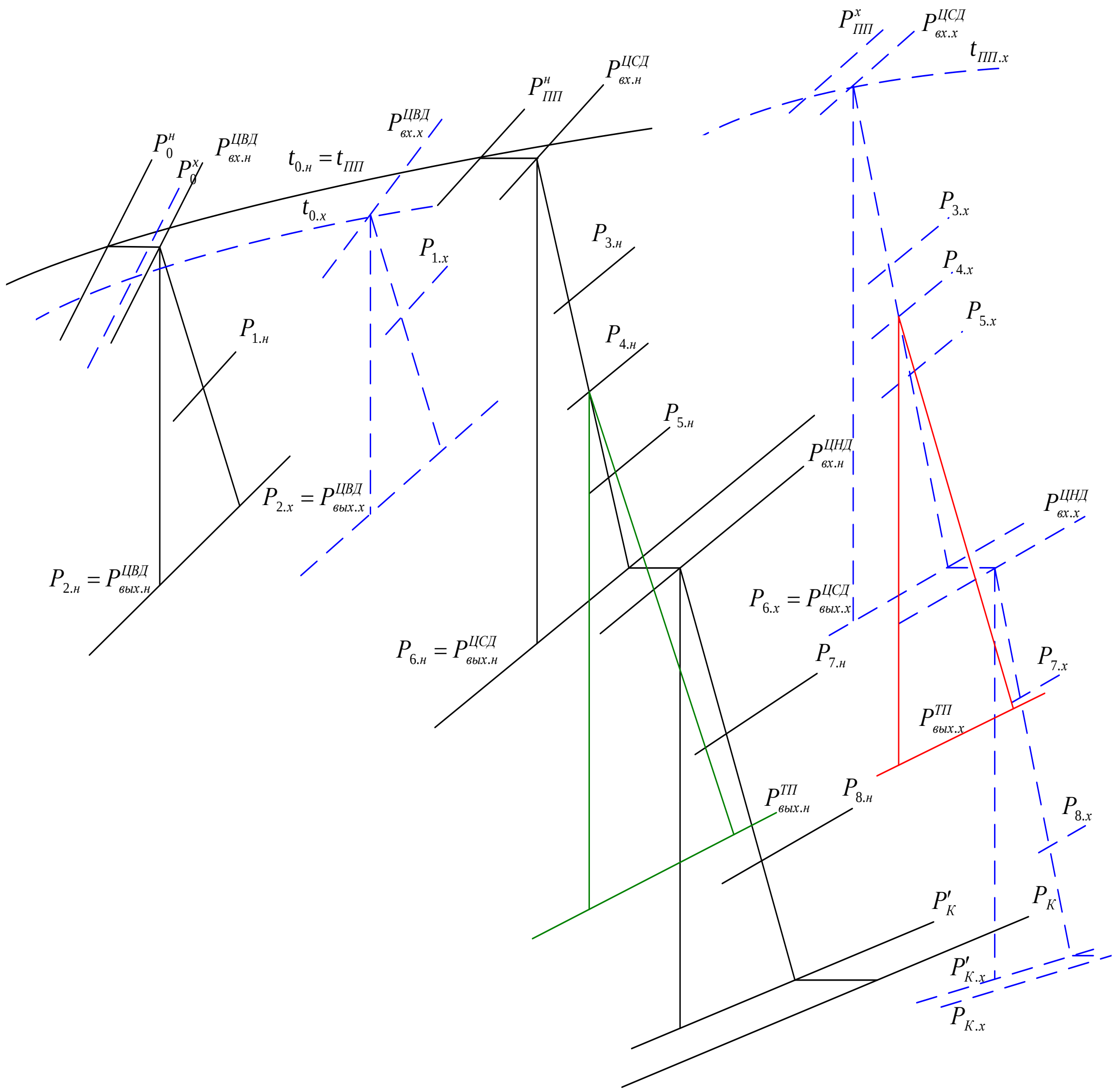
Where: $Q_р^н = 32491{,}4\frac{кДж}{кг} -$ calorific value of conventional fuel.

Specific fuel consumption for electricity generation per unit:

$$b_Э^{отп} = \frac{123}{\eta_{ТУ}^э} = \frac{123}{0{,}461} = 266{,}81\frac{г.у.т}{кВт \cdot ч}.$$

Specific fuel consumption per kilowatt-hour:

$$b_Э^{отп} = \frac{123}{\eta_е^э} = \frac{123}{0{,}401} = 306{,}50\frac{г.у.т}{кВт \cdot ч}.$$



The process of expanding the steam in:

- nominal mode;

- a turbo-drive of a nominal mode;

- partial mode 60% of the nominal;

- a partial-mode turbo.

Figure 4.1. The process of expansion of steam in a turbine in nominal and variable mode.

4. Calculation of the heat circuit in a new and partial modes

4.1. Recalculation of basic characteristics into partial mode

Relative steam flow rate to the turbine in partial mode: $d_0 = 0,6.$
In the first approximation, the calculation of vapor pressures in the selections for the new regime is determined according to the dependence:

$$P^X_{отб1} = P_{отб1} \cdot d_0 = 5,97 \cdot 0,6 = 3,582 МПа;$$
$$P^X_{отб2} = P_{отб2} \cdot d_0 = 3,66 \cdot 0,6 = 2,196 МПа;$$
$$P^X_{отб3} = P_{отб3} \cdot d_0 = 1,93 \cdot 0,6 = 1,158 МПа;$$
$$P^X_{отб4} = P_{отб4} \cdot d_0 = 0,9 \cdot 0,6 = 0,54 МПа;$$
$$P^X_{отб5} = P_{отб5} \cdot d_0 = 0,47 \cdot 0,6 = 0,282 МПа;$$
$$P^X_{отб6} = P_{отб6} \cdot d_0 = 0,2 \cdot 0,6 = 0,12 МПа;$$
$$P^X_{отб7} = P_{отб7} \cdot d_0 = 0,084 \cdot 0,6 = 0,05 МПа;$$
$$P^X_{отб8} = P_{отб8} \cdot d_0 = 0,024 \cdot 0,6 = 0,014 МПа.$$

According to the recalculation of pressures according to the Stodol-Flugel formula for calculation in the new regime, we take the refined pressures (Table 4).

Saturation temperature in the condenser:

$$t_{sx} = \frac{t_{sн} \cdot G_{кх} - t_{в1н} \cdot G_{кх} + G_{кн} \cdot t_{в1х}}{G_{кн}}, \ ^oC;$$

Where: $t_{sн}$ - saturation temperature at the condenser pressure in the nominal mode;
$t_{в1н}$, $t_{в1х}$, - the temperature of the water cooling the condenser in the nominal and new modes, we assume $t_{в1н} = t_{в1х}$ =10 °C;
$G_{кн}$, $G_{кх}$ - steam flow to the condenser in the nominal and new mode;

$$G_{кх} = G_{кн} \cdot d_0 = 389,95 \cdot 0,6 = 233,97 \frac{кг}{с};$$

Then the saturation temperature in the condenser:

$$t_{sx} = \frac{25,1 \cdot 233,97 - 10 \cdot 233,97 + 10 \cdot 389,95}{389,95} = 19,1^o C.$$

Vapor pressure in the condenser:

P_k = f(t_{sx}) = 0,0022 МПа.
The vapor pressure at the output of the LPC:

P_k = 1,05 · P_k = 0,0023 МПа.
Heat transfer coefficients in new modes:

$$К_x = К_x \cdot \left(\frac{G_{вх}}{G_{вн}} \right)^{0,5};$$

For HPH:

$$К_x = 3 \cdot (0,6)^{0,5} = 2,32 \, кВт \big/ (м^2 \cdot к);$$

For LPH:

$$К_x = 4 \cdot (0,6)^{0,5} = 3,1 \, кВт \big/ (м^2 \cdot к).$$

To maintain constant pressure in the deaerator we switch it to the third selection, leaving only TP at the fourth selection.
Table 4.1.

Таблица 4.1.

	$P^н_{отб.i}$, $МПа$	$G^н_{отб.i}$, $\frac{кг}{с}$	$P^x_{отб.i}$, $МПа$	$P^x_{П.i}$, $МПа$	t_S, ^oC
k	0,00345	347,59	0,0023	0,0022	19,1
8	0,024	38,23	0,014	0,013	51,0
7	0,084	16,32	0,05	0,048	80,3
6	0,20	15,88	0,12	0,114	103,3
5	0,47	26,24	0,282	0,268	129,7
Д	0,9	11,06	1,158	0,70	165,0
ТП		31,48	0,54	0,513	170,5
3	1,93	40,18	1,158	1,10	184,1
2	3,66	47,68	2,196	2,086	214,5

1	5,97	47,31	3,582	3,403	241,0
0	23,54	664,33	14,124	–	–

To determine the underheating of water in the new regime, we take the water flow through the heaters (Table 5):

$G_{в.х.} = d_0 \cdot G_{в.н}$.

Feed water pressure:

$P_{пв} = 1{,}4 \cdot P_0 = 1{,}4 \cdot 23{,}54 = 33 \;\; МПа.$

Main condensate pressure:

$P_{ок} = 2{,}0 \cdot P_д = 2{,}0 \cdot 0{,}7 = 1{,}4 \;\; МПа.$

Heating water in the OE and OU is $\Delta t_{оэ} = \Delta t_{оу} = 3°C$.

The water temperature behind the heater is determined by taking into account the underheating:

$t^{П.i}_{в.вых} = t_{S.i} - \Theta_{П.i}$.

Underheating of heaters is determined by the formula:

$\Theta_i = \left(t_{S.i} - t^{П.i}{}_{в.вх}\right) \cdot e^{-\frac{K \cdot F}{C_в \cdot G_в}}$

.

The heat capacity of the main condensate and feed water is determined by:

$C_в = f\left(P_{ок}(P_{пв}), t_{в.вх.}\right), \frac{кДж}{кг \cdot К}.$

Increasing the temperature of feed water in the PN:

$\Delta t_{ПН} = \frac{V' \cdot (P_{ПН} - P_д) \cdot 100}{C_в}, {}^oC;$

Where: V' - the specific volume of water at the temperature at the outlet from the deaerator.

Feedwater temperature for PN:

$t_{ПН} = t^д_{в.вых} + \Delta t_{ПН}$, °C.

The vapor pressure in the heater is determined taking into account the hydraulic losses in the pipeline selection:

$P^x{}_{П.i} = 0{,}95 \cdot P^x{}_{отбi}$, МПа.

Enthalpy at the outlet of the heater:

$$h_{вых} = f\left(P_в^x, t_{вых}\right), \frac{кДж}{кг}.$$

4.2. Calculation of underheats and temperatures behind the heaters

The inlet temperature in the heater P7:

$$t_{в.вх}^{П7} = t_к + \Delta t_{оэ} = 19{,}1 + 3 = 22{,}1°C.$$

Underheating of heater P7:

$$\Theta_{П7} = \left(t_{S.7} - t^{П.7}{}_{в.вх}\right) \cdot e^{-\frac{K \cdot F}{C_в \cdot G_в}} = \left(51{,}0 - 22{,}1\right) \cdot e^{-\frac{3{,}1 \cdot 1630}{4{,}176 \cdot 308{,}3}} = 0{,}6°C.$$

Temperature at the outlet of the heater P7:

$$t_{в.вых}^{П7} = t_{S.7} - \Theta_{П.7} = 51{,}0 - 0{,}6 = 50{,}4°C.$$

The inlet temperature in the preheater P6:

$$t_{в.вх}^{П6} = t_{в.вых}^{П7} + \Delta t_{оу} = 50{,}4 + 3 = 53{,}4°C.$$

Underheating of heater P6:

$$\Theta_{П6} = \left(t_{S.6} - t^{П.6}{}_{в.вх}\right) \cdot e^{-\frac{K \cdot F}{C_в \cdot G_в}} = \left(80{,}3 - 53{,}4\right) \cdot e^{-\frac{3{,}1 \cdot 1550}{4{,}192 \cdot 308{,}3}} = 0{,}7°C.$$

Temperature at the outlet of the heater P6:

$$t_{в.вых}^{П6} = t_{S.6} - \Theta_{П.6} = 80{,}3 - 0{,}7 = 79{,}6°C.$$

The temperature at the inlet to the heater P5:

$$t_{в.вх}^{П5} = f\left(h_{в.вх}^{П5}, P_{ок}\right) = 81{,}7°C.$$

Underheating of heater P5:

$$\Theta_{П5} = \left(t_{S.5} - t^{П.5}{}_{в.вх}\right) \cdot e^{-\frac{K \cdot F}{C_в \cdot G_в}} = \left(103{,}3 - 81{,}7\right) \cdot e^{-\frac{3{,}1 \cdot 2233}{4{,}218 \cdot 344{,}8}} = 0{,}6°C.$$

Temperature at the outlet of the heater P5:

$$t_{в.вых}^{П5} = t_{S.5} - \Theta_{П.5} = 103{,}3 - 0{,}6 = 102{,}7°C;$$

$$t^{П.4}{}_{в.вх} = t_{в.вых}^{П5}.$$

Underheating of heater P4:

$$\Theta_{П4} = \left(t_{S.4} - t^{П.4}{}_{в.вх}\right) \cdot e^{-\frac{K \cdot F}{C_в \cdot G_в}} = \left(129{,}7 - 102{,}7\right) \cdot e^{-\frac{3{,}1 \cdot 2330}{4{,}261 \cdot 344{,}8}} = 0{,}5°C.$$

Temperature at the outlet of the heater P4:

$$t^{П4}_{в.вых} = t_{S.4} - \Theta_{П.4} = 129{,}7 - 0{,}5 = 129{,}2°C.$$

Underheating of heater P3:

$$\Theta_{П3} = \left(t_{S.3} - t^{П.3}{}_{в.вх}\right)\cdot e^{-\frac{K\cdot F}{C_в \cdot G_в}} = (184{,}1 - 170{,}5)\cdot e^{-\frac{2.32\cdot 1560}{4{,}296\cdot 414{,}54}} = 1{,}8°C.$$

Temperature at the outlet of the heater P3:

$$t^{П3}_{в.вых} = t_{S.3} - \Theta_{П.3} = 184{,}1 - 1{,}8 = 182{,}3°C.$$

The temperature at the inlet to the heater P2:

$$t^{П2}_{в.вх} = t^{П3}_{в.вых}.$$

Underheating of heater P2:

$$\Theta_{П2} = \left(t_{S.2} - t^{П.2}{}_{в.вх}\right)\cdot e^{-\frac{K\cdot F}{C_в \cdot G_в}} = (214{,}5 - 182{,}3)\cdot e^{-\frac{2{,}32\cdot 2135}{4{,}395\cdot 414{,}54}} = 2{,}1°C.$$

Temperature at the outlet of the heater P2:

$$t^{П2}_{в.вых} = t_{S.2} - \Theta_{П.2} = 214{,}5 - 2{,}1 = 212{,}4°C.$$

The temperature at the inlet to the heater P1:

$$t^{П1}_{в.вх} = t^{П2}_{в.вых}.$$

Underheating of heater P1:

$$\Theta_{П1} = \left(t_{S.1} - t^{П.1}{}_{в.вх}\right)\cdot e^{-\frac{K\cdot F}{C_в \cdot G_в}} = (241{,}0 - 212{,}4)\cdot e^{-\frac{2{,}32\cdot 1650}{4{,}523\cdot 414{,}54}} = 3{,}7°C.$$

Temperature at the outlet of the heater P1:

$$t^{П1}_{в.вых} = t_{S.1} - \Theta_{П.1} = 241{,}0 - 3{,}7 = 237{,}3°C.$$

Таблица. 4.2.

	$G_{в.н}, \frac{кг}{с}$	$G_{в.х}, \frac{кг}{с}$	$P^x_в$, $МПа$	$t_{вых}$, °C	$C_p, \frac{кДж}{кг\cdot К}$	f	Θ_x, °C	$h_{вых}$, кДж/кг
ОЭ	512,51	307,51	1,40	22,1	-	-	-	94,0
П7	513,84	308,30	1,40	50,4	4,18	1630	0,6	212,2
ОУ	513,84	308,30	1,40	53,4	4,18	-	-	224,7
П6	513,84	308,30	1,40	79,6	4,18	1550	0,7	334,4
СМ	574,66	344,80	1,40	81,7	4,20	-	-	343,2
П5	574,66	344,80	1,40	102,7	4,21	2233	0,6	431,5

П4	574,66	344,80	1,40	129,2	4,23	2330	0,5	543,7
Д	690,9	414,54	0,70	165,0	-	-	-	697,1
ПН	690,9	414,54	33,0	170,05	4,25	-	-	739,2
П3	690,9	414,54	33,0	182,3	4,27	1560	1,8	790,2
П2	690,9	414,54	33,0	212,4	4,27	2135	2,1	920,7
П1	690,9	414,54	33,0	237,3	4,40	1650	3,7	1031,3

Calculation of the scheme is carried out according to the equation of thermal balances.

We accept:

Efficiency of surface heaters $\eta_п = 0{,}98$;

Deaerator efficiency and flow mixing points: $\eta_{см} = 0{,}995$.

Steam consumption per turbine:

$$G_{0x} = d_0 \cdot G_0^н = 0{,}6 \cdot 664{,}33 = 398{,}60\,{кг}/{c}.$$

Feedwater consumption:

$$G_{пв}^x = G_{0x} \cdot \alpha_{пв} = 398{,}6 \cdot 1{,}04 = 414{,}54\,{кг}/{c}.$$

The enthalpies of steam in the selections are determined from the process of vapor expansion in the "h-s" diagram at known pressures in the selections and efficiency of the cylinders.

$$\eta_{oi.x}^{ЦВД} = \eta_{oi.н}^{ЦВД} \cdot \left(0{,}629 + 0{,}897 \cdot d_0^2 - 0{,}528 \cdot d_0^3\right) =$$

$$= 0{,}86 \cdot \left(0{,}629 + 0{,}897 \cdot 06^2 - 0{,}528 \cdot 0{,}6^3\right) = 0{,}72;$$

$$\eta_{oi.x}^{ЦСД} = \eta_{oi.н}^{ЦСД} \cdot \left(1{,}04 - 0{,}04 \cdot d_0\right) = 0{,}9 \cdot \left(1{,}04 - 0{,}04 \cdot 0{,}6\right) = 0{,}91;$$

$$\eta_{oi.x}^{ЦНД} = \eta_{oi.н}^{ЦНД} \cdot \left(1{,}1 - 0{,}1 \cdot d_0\right) = 0{,}85 \cdot \left(1{,}1 - 0{,}1 \cdot 0{,}6\right) = 0{,}88.$$

We determine the enthalpy of drainage from the vapor pressure in the heater as a function of the saturation pressure:

$$h' = f\left(P^x{}_{П.i}\right).$$

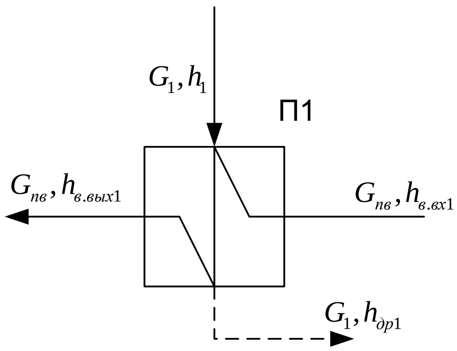
Table 4.3. The process of steam expansion in the turbine and the enthalpy of drainage.

	h, *кДж/кг*	*s*, *кДж/(кг·К)*	*P, МПа*	*t*, ^oC	*h', кДж/кг*
0	3387,4	6,26	23,54	560,0	-
вход в ЦВД	3496,3	6,63	14,124	560,0	-
1	3156,3	6,73	3,582	372,2	1056,2
2	3053,2	6,78	2,196	314,8	930,5
вход в ЦСД	3557,2	7,56	1,939	540,0	-
3	3406,0	7,60	1,158	467,1	791,4
4	3236,1	7,71	0,54	383,0	652,8
5	3078,0	7,74	0,282	303,9	552,5
6	2898,0	7,80	0,12	212,3	439,3
вход в ЦНД	2898,0	7,82	0,114	212,2	-
7	2745,3	7,86	0,05	132,1	340,5
8	2613,7	8,09	0,014	61,8	220,0

k'	2537,0	8,67	0,0023	19,8	-
k	2535,8	8,69	0,0022	19,1	-

4.3. Calculation of the thermal scheme of partial mode

4.3.1. High-pressure heaters

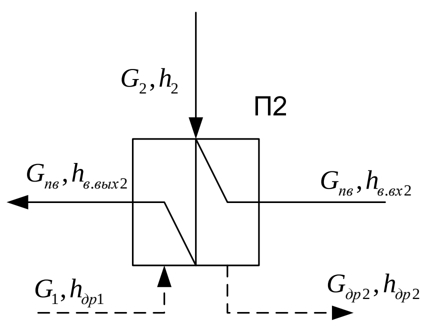


Heat balance equation for preheater P1:

$$G_1^{П} \cdot \left(h_1 - h_{др1}\right) \cdot \eta_п = G_{пв} \cdot \left(h_{в.вых1} - h_{в.вх1}\right).$$

Steam flow for the heater:

$$G_1 = G_1^{П} = \frac{G_{пв} \cdot \left(h_{в.вых1} - h_{в.вх1}\right)}{\left(h_1 - h_{др1}\right) \cdot \eta_п} = \frac{414{,}54 \cdot (1031{,}3 - 920{,}7)}{(3156{,}3 - 1056{,}2) \cdot 0{,}98} = 22{,}28 \frac{кг}{с}.$$



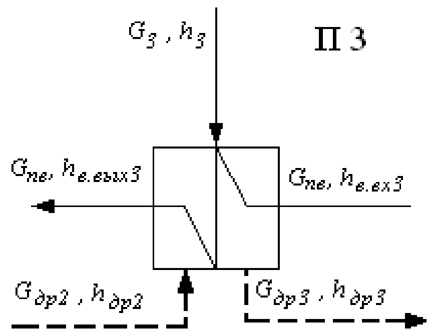
Heat balance equation for preheater P2:

$$\left(G_2^{П} \cdot \left(h_2 - h_{др2}\right) + G_1 \cdot \left(h_{др1} - h_{др2}\right)\right) \cdot \eta_п = G_{пв} \cdot \left(h_{в.вых2} - h_{в.вх2}\right).$$

Steam flow for the heater:

$$G_2 = G_2^{П} = \frac{\dfrac{G_{пв} \cdot (h_{в.вых2} - h_{в.вх2})}{\eta_n} - G_1 \cdot (h_{др1} - h_{др2})}{(h_2 - h_{др2})} =$$

$$= \frac{\dfrac{414{,}54 \cdot (920{,}7 - 790{,}2)}{0{,}98} - 22{,}28 \cdot (1056{,}2 - 930{,}5)}{(3053{,}2 - 930{,}5)} = 24{,}69\,\frac{кг}{с}.$$



Heat balance equation for preheater P3:

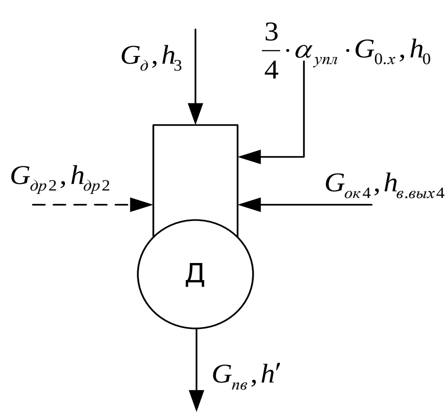
$$\left(G_3^{П} \cdot (h_3 - h_{др3}) + G_2 \cdot (h_{др2} - h_{др3})\right) \cdot \eta_n = G_{пв} \cdot (h_{в.вых3} - h_{в.вх3})\,..$$

Steam flow for the heater:

$$G_3^{П} = \frac{\dfrac{G_{пв} \cdot (h_{в.вых3} - h_{в.вх3})}{\eta_n} - G_2 \cdot (h_{др2} - h_{др3})}{(h_3 - h_{др3})} =$$

$$= \frac{\dfrac{414{,}54 \cdot (790{,}2 - 739{,}2)}{0{,}98} - 24{,}69 \cdot (930{,}5 - 791{,}4)}{(3406{,}0 - 791{,}4)} = 6{,}94\,\frac{кг}{с}.$$

4.3.2. Deaerator of feed water



The equation of the material balance of deaerator:

$$G_{пв} + G_{выпар} = G_{др3} + G_{д} + G_{ок4},\ \frac{кг}{c};$$

$$G_{др3} = G_{др1} + G_{др2} + G_{др3} = 22{,}28 + 24{,}69 + 6{,}94 = 53{,}91\frac{кг}{c};$$

$$G_{выпар} = \alpha_{э} \cdot G_{0x} = 0{,}002 \cdot 398{,}6 = 0{,}80\frac{кг}{c}.$$

Heat balance equation of deaerator D:

$$G_{пв} \cdot h' = \left(G_{др3} \cdot h_{др3} + G_{д} \cdot h_{3} + G_{ок4} \cdot h_{в.вых4} - G_{выпар} \cdot h_{выпар}\right) \cdot \eta_{см}.$$

Solving the equations of heat and material balances we find the flow of steam for the deaerator and main condensate flow through the heater P4:

$$G_{ок4} = 344{,}80\ \frac{кг}{c};$$

$$G_{д} = 15{,}03\ \frac{кг}{c}.$$

The steam consumption in the 3rd selection:

$$G_{3} = G_{3}^{П} + G_{д} = 6{,}94 + 15{,}03 = 21{,}97\frac{кг}{c}.$$

4.3.3. Turbine-driven feed pump

Increase of the feed water temperature in the PN:

$$\Delta h_{ТП} = \frac{\upsilon' \cdot (P_{ТН} - P_{д}) \cdot 10^{3}}{\eta_{нi}},$$

where: $\upsilon' -$ specific volume of water at the outlet of the deaerator;;

$\eta_{нi}$ – “Internal" (hydraulic) pump efficiency (6. 13-2);

$$\Delta h_{ТП} = \frac{0{,}001108 \cdot (33{,}0 - 0{,}7) \cdot 10^3}{0{,}85} = 42{,}1 \frac{кДж}{кг}.$$

The enthalpy of the feedwater behind the pump:

$$h_{пв.н} = h_S^{д} + \Delta h_{ТП} = 697{,}1 + 42{,}1 = 739{,}2 \frac{кДж}{кг}.$$

Water parameters for the feed pump:

$$P_{ТП} = 33{,}0 МПа; \; t_{пв.н} = 170{,}5°C.$$

Consumption of steam for turbine drive feed pump:

$$G_{ТП} = \frac{G_{пв}^{x} \cdot h_{н}^{a}}{H_i^{ТП} \cdot \eta_н \cdot \eta_м} = \frac{414{,}54 \cdot 35{,}53}{534{,}5 \cdot 0{,}833 \cdot 0{,}98} = 33{,}76 \frac{кг}{с};$$

where: $h_н^a = \upsilon' \cdot (p_н - p_в) \cdot 10^3 = 0{,}001108 \cdot (33 - 0{,}7) \cdot 10^3 = 35{,}53 кДж/кг$ – adiabatic compression work in the pump;

$H_{ТП} = 3236{,}1 - 2701{,}6 = 534{,}5 кДж/кг$ – the actual heat drop of the steam in the turbine driven feed pump;

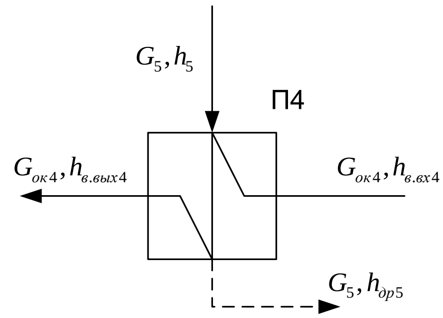
$\eta_н = \eta_{нi} \cdot \eta'_н = 0{,}85 \cdot 0{,}98 = 0{,}833$ – total efficiency of the pump with volumetric and mechanical losses (6 13-4);

$\eta_м$ – mechanical efficiency of the turbine drive (6 13-4).

Power of turbine drive feed pump:

$$N_{ТП}^{x} = G_{ТП} \cdot H_i^{ТП} = 33{,}76 \cdot 534{,}5 \cdot 10^{-3} = 18{,}05 МВт.$$

4.3.4. Low pressure heaters

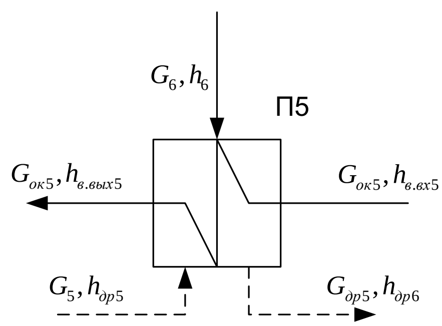


The equation of thermal balance for the heater P4:

$$G_4^{П} \cdot \left(h_5 - h_{др5}\right) \cdot \eta_n = G_{ок4} \cdot \left(h_{в.вых4} - h_{в.вх4}\right).$$

The steam consumption for the heater:

$$G_5 = G_4^{П} = \frac{G_{ок4} \cdot \left(h_{в.вых4} - h_{в.вх4}\right)}{\left(h_5 - h_{др5}\right) \cdot \eta_n} = \frac{344{,}8 \cdot (543{,}7 - 431{,}5)}{\left(3078{,}0 - 552{,}5\right) \cdot 0{,}98} = 15{,}63 \frac{кг}{с}.$$



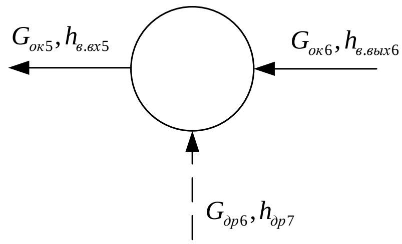
The equation of thermal balance for P5:

$$\left(G_5^{П} \cdot \left(h_6 - h_{др6}\right) + G_5 \cdot \left(h_{др5} - h_{др6}\right)\right) \cdot \eta_n = G_{ок5} \cdot \left(h_{в.вых5} - h_{в.вх5}\right).$$

The steam consumption for the heater:

$$G_6 = G_5^{П} = \frac{\dfrac{G_{ок5} \cdot \left(h_{в.вых5} - h_{в.вх5}\right)}{\eta_n} - G_5 \cdot \left(h_{др5} - h_{др6}\right)}{\left(h_6 - h_{др6}\right)} =$$

$$= \frac{\dfrac{344{,}8 \cdot \left(431{,}5 - 343{,}2\right)}{0{,}98} - 15{,}63 \cdot \left(552{,}5 - 439{,}3\right)}{\left(2898{,}0 - 439{,}3\right)} = 11{,}92 \frac{кг}{с}.$$



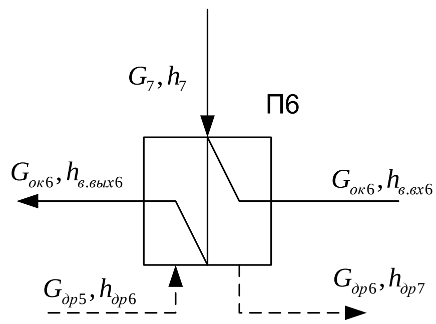
Material balance of mixing point:

$$G_{др6} = G_5 + G_6 + G_7 = 15{,}63 + 11{,}92 + 12{,}70 = 40{,}25\, ^{кг}\!/_{c};$$

$$G_{др6} + G_{ок6} = G_{ок5} = G_{ок4}.$$

The equation of the heat balance of the mixing point:

$$\left(G_{др6} \cdot h_{др7} + G_{ок6} \cdot h_{в.вых6}\right) \cdot \eta_{см} = G_{ок5} \cdot h_{в.вх5}.$$



Solving the equation for the mixing point and the heater P6:

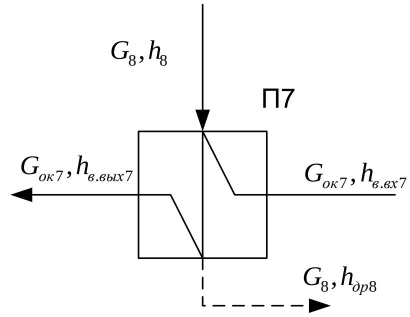
Heat balance equation for heater P6:

$$\left(G_6^{П} \cdot \left(h_7 - h_{др7}\right) + G_{др5} \cdot \left(h_{др6} - h_{др7}\right)\right) \cdot \eta_n = G_{ок6} \cdot \left(h_{в.вых6} - h_{в.вх6}\right);$$

steam consumtion for the heater:

$$G_7 = G_6^{П} = \frac{\dfrac{G_{ок6} \cdot \left(h_{в.вых6} - h_{в.вх6}\right)}{\eta_n} - G_{др6} \cdot \left(h_{др6} - h_{др7}\right)}{\left(h_7 - h_{др7}\right)} =$$

$$= \frac{\dfrac{308{,}30 \cdot \left(334{,}4 - 224{,}7\right)}{0{,}98} - 40{,}25 \cdot \left(439{,}3 - 340{,}5\right)}{\left(2745{,}3 - 340{,}5\right)} = 12{,}70\, \frac{кг}{c}.$$



The equation of thermal balance for the heater P7:

$$G_7^{П} \cdot \left(h_8 - h_{др8}\right) \cdot \eta_n = G_{ок7} \cdot \left(h_{в.вых7} - h_{в.вх7}\right).$$

The steam consumption for the heater:

$$G_8 = G_7^{П} = \frac{G_{ок7} \cdot \left(h_{в.вых7} - h_{в.вх7}\right)}{\left(h_8 - h_{др8}\right) \cdot \eta_n} = \frac{308{,}30 \cdot (212{,}2 - 94{,}0)}{\left(2613{,}74 - 220{,}0\right) \cdot 0{,}98} = 15{,}53 \frac{кг}{с}.$$

4.4. The material balance of steam consumption in the condenser

The steam consumption in the capacitor of the turbine material balance:

$$G_{кх}^{T} = G_0^{x} - \sum_{i=1}^{8} G_{отб.i}^{x} = 398{,}6 -$$
$$-\left(22{,}28 + 24{,}69 + 21{,}97 + 33{,}76 + 15{,}63 + 11{,}92 + 12{,}70 + 15{,}53\right) = 240{,}12 \,{}^{кг}\!/_{с}.$$

Steam flow to the condenser from the balance of the condenser:

$$G_{кх}^{К} = G_{вх.Э}^{x} - G_{ТП} - G_{отб.8}^{x} - \alpha_{выпар} \cdot G_0^{x} - a_{э} \cdot G_0^{x} - \alpha_{дв} \cdot G_0^{x} =$$
$$= 307{,}51 - 33{,}76 - 15{,}53 - 0{,}028 \cdot 398{,}6 - 0{,}002 \cdot 398{,}6 - 0{,}01 \cdot 398{,}6 = 240{,}12 \,{}^{кг}\!/_{с}.$$

Зная расходы пара через отсеки в номинальном и новом режиме, рассчитываем давления пара в отборах по формуле Стодола – Флюгеля:

Knowing the steam flow through the compartments in the nominal and the new regime, we calculate the vapor pressure in the selections with formula of Stodol - Flyugel:

the steam pressure in the 8th selection:

$$P_{8x} = \sqrt{\left(\frac{G_{8-\kappa x}}{G_{8-\kappa н}}\right)^2 \cdot \left(P_{8н}^2 - P_{\kappa н}^2\right) + P_{\kappa x}^2};\ \text{МПа};$$

the steam pressure in the 7th selection:

$$P_{7x} = \sqrt{\left(\frac{G_{7-8x}}{G_{7-8н}}\right)^2 \cdot \left(P_{7н}^2 - P_{8н}^2\right) + P_{8x}^2};\ \text{МПа};$$

the steam pressure in the 6th selection:

$$P_{6x} = \sqrt{\left(\frac{G_{6-7x}}{G_{6-7н}}\right)^2 \cdot \left(P_{6н}^2 - P_{7н}^2\right) + P_{7x}^2};\ \text{МПа};$$

the steam pressure in the 5th selection:

$$P_{5x} = \sqrt{\left(\frac{G_{5-6x}}{G_{5-6н}}\right)^2 \cdot \left(P_{5н}^2 - P_{6н}^2\right) + P_{6x}^2};\ \text{МПа};$$

the steam pressure in the 4th selection:

$$P_{4x} = \sqrt{\left(\frac{G_{4-5x}}{G_{4-5н}}\right)^2 \cdot \left(P_{4н}^2 - P_{5н}^2\right) + P_{5x}^2};\ \text{МПа};$$

the steam pressure in the 3rd selection:

$$P_{3x} = \sqrt{\left(\frac{G_{3-4x}}{G_{3-4н}}\right)^2 \cdot \left(P_{3н}^2 - P_{4н}^2\right) + P_{4x}^2};\ \text{МПа};$$

the steam pressure in the 2nd selection:

$$P_{2x} = \sqrt{\left(\frac{G_{2-3x}}{G_{2-3н}}\right)^2 \cdot \left(P_{2н}^2 - P_{3н}^2\right) + P_{2x}^2};\ \text{МПа};$$

the steam pressure in the 1st selection:

$$P_{1x} = \sqrt{\left(\frac{G_{1-2x}}{G_{1-2н}}\right)^2 \cdot \left(P_{1н}^2 - P_{2н}^2\right) + P_{2x}^2}\ ;\ \text{МПа};$$

Moving pressure:

$$P_{ск} = \sqrt{\left(\frac{G_{0-1x}}{G_{0-1н}}\right)^2 \cdot \left(P_{0н}^2 - P_{1н}^2\right) + P_{1x}^2}\ ;\ \text{МПа}.$$

Table 4.4. Determination of the pressure.

Таблица 4.4. Определение давлений.

	Nominal mode	New mode	$P_{ном}$, МПа	$P_{х}$, МПа
G0-1	664,33	398,6	23,54	17,655
G1-2	617,02	376,32	5,97	3,582
G2-3	569,34	351,63	3,66	2,196
G3-4	529,16	329,66	1,93	1,158
G4-5	486,62	295,90	0,90	0,54
G5-6	460,38	280,27	0,47	0,282
G6-7	444,50	268,35	0,20	0,12
G7-8	428,18	255,65	0,084	0,05
G8-к	389,95	240,12	0,024	0,014

Действительные теплоперепады турбины по цилиндрам:

$$H_i^{ЦВД} = h_{вх.х}^{ЦВД} - h_2 = 3496{,}3 - 3053{,}2 = 443{,}1 \ \frac{кДж}{кг};$$

$$H_i^{ЦСД} = h_{вх.х}^{ЦСД} - h_6 = 3557{,}2 - 2898{,}0 = 659{,}2 \ \frac{кДж}{кг};$$

$$H_i^{ЦНД} = h_{вх.х}^{ЦНД} - h_к' = 2898{,}0 - 2535{,}8 = 362{,}2 \ \frac{кДж}{кг};$$

$$H_i = H_i^{ЦВД} + H_i^{ЦСД} + H_i^{ЦНД} = 443{,}1 + 659{,}2 + 362{,}2 = 1464{,}5 \ \frac{кДж}{кг}.$$

Недовыработка определяется по формуле:

$$y_i = \frac{h_{отб.i} - h_к + \Delta h_{пп}}{H_i};$$

где: $\Delta h_{пп} = h_{вх.х}^{ЦСД} - h_2 = 3557{,}2 - 3053{,}2 = 504{,}0 \frac{кДж}{кг}$ – повышение энтальпии в промперегревателе.

Относительные расходы пара по отборам определяются по формуле:

$$\alpha_i = \frac{G^x{}_{отб.i}}{G^x{}_0}.$$

Таблица 4.5.

№	$G_{отб.i}$	α_i	y_i	$\alpha_i \cdot y_i$
1	22,28	0,0559	0,7670	0,04287
2	24,69	0,0620	0,6966	0,04319
3	6,94	0,0174	0,5934	0,01033
Д	15,03	0,0377	0,5934	0,02238
ТП	33,76	0,0847	0,4774	0,04044

5	15,63	0,0392	0,3694	0,01449
6	11,92	0,0299	0,2465	0,00738
7	12,70	0,0319	0,1422	0,00454
8	15,53	0,0390	0,0524	0,00204

We calculate the turbine power:

$$N_Э = \frac{G_0^x \cdot \left[H_i \cdot \eta_м \cdot \eta_г \cdot \left(1 - \sum \alpha_i \cdot y_i\right)\right]}{10^3} =$$

$$= \frac{398{,}6 \cdot \left[1464{,}5 \cdot 0{,}982 \cdot 0{,}98 \cdot \left(1 - 0{,}18766\right)\right]}{10^3} = 465{,}67 MBm.$$

where: $\eta_м = 0{,}982$ – the mechanical efficiency;

$\eta_г = 0{,}98$ – The efficiency of the generator.

4.5. Determination of indicators of thermal efficiency in the partial mode

Heat let in the steam generator:

$$Q_{ПГ} = G_{пв}^x \cdot \left(h_{0x}^{ПГ} - h_{пв}\right) + \Delta h_{пп} \cdot G_{пп};$$

where:

$$G_{пп} = G_{ту} - G_1 - G_2 = 398{,}6 - 22{,}28 - 24{,}69 = 351{,}63;$$

$$Q_{ПГ} = \frac{414{,}54 \cdot \left(3386{,}9 - 1031{,}3\right) + 504 \cdot 351{,}63}{10^3} = 1153{,}92 MBm.$$

Heat supplied to the turbine:

$$Q_{ТУ} = G_{0.x} \cdot \left(h_{0.x} - h_{пв}\right) + \Delta h_{пп} \cdot G_{пп};$$

$$Q_{ТУ} = \frac{398{,}6 \cdot \left(3387{,}4 - 1031{,}3\right) + 504 \cdot 351{,}63}{10^3} = 1116{,}36 MBm.$$

Electrical efficiency of the installation:

$$\eta_{ту}^{Э} = \frac{N_{Э} + N_{ТП}}{Q_{ТУ}} = \frac{465,67 + 18,53}{1116,36} = 0,434.$$

Efficiency of transport:

$$\eta_{ТР} = \frac{Q_{ТУ}}{Q_{ПГ}} = \frac{1116,36}{1153,92} = 0,968.$$

Effective efficiency for electricity output:

$$\eta_e^{э} = \eta_{ту}^{э} \cdot \eta_{ТР} \cdot \eta_{ПГ} \cdot (1 - k_{сн});$$

where: $\eta_{ПГ} = 0,925$ – the efficiency of the steam generator (SG), accept; $k_{сн} = 0,03$ – for the coefficient that takes into account electricity consumption for its own needs, we accept;

$$\eta_e^{э} = 0,434 \cdot 0,968 \cdot 0,925 \cdot (1 - 0,03) = 0,366.$$

Equivalent fuel consumption per block:

$$B_у = \frac{Q_{ПГ}}{\eta_{ПГ} \cdot Q_р^н} = \frac{1153920}{0,925 \cdot 32491,4} = 38,39 \frac{кг}{с}.$$

where: $Q_р^н = 32491,4 \frac{кДж}{кг}$ – calorific value of fuel equivalent.

Specific fuel consumption for electric power generation per unit:

$$b_Э^{отп} = \frac{123}{\eta_{ТУ}^{э}} = \frac{123}{0,434} = 283,41 \frac{г.у.т}{кВт \cdot ч}.$$

Specific fuel consumption per generated kWh:

$$b_Э^{отп} = \frac{123}{\eta_e^{э}} = \frac{123}{0,366} = 336,07 \frac{г.у.т.}{кВт \cdot ч}.$$

In the transition from the nominal mode operation of the unit to the moving parameters (by reducing the steam flow through the boiler, and therefore, the pressure, constant steam temperature is maintained) unit profitability decreased due to an increase $b_э^{отп}$ in fuel consumption per generated kWh, increased by 29,57 $\frac{г.у.т.}{кВт \cdot ч}$, and is $b_Э^{отп} = 336{,}07\,\frac{г.у.т.}{кВт \cdot ч}$.

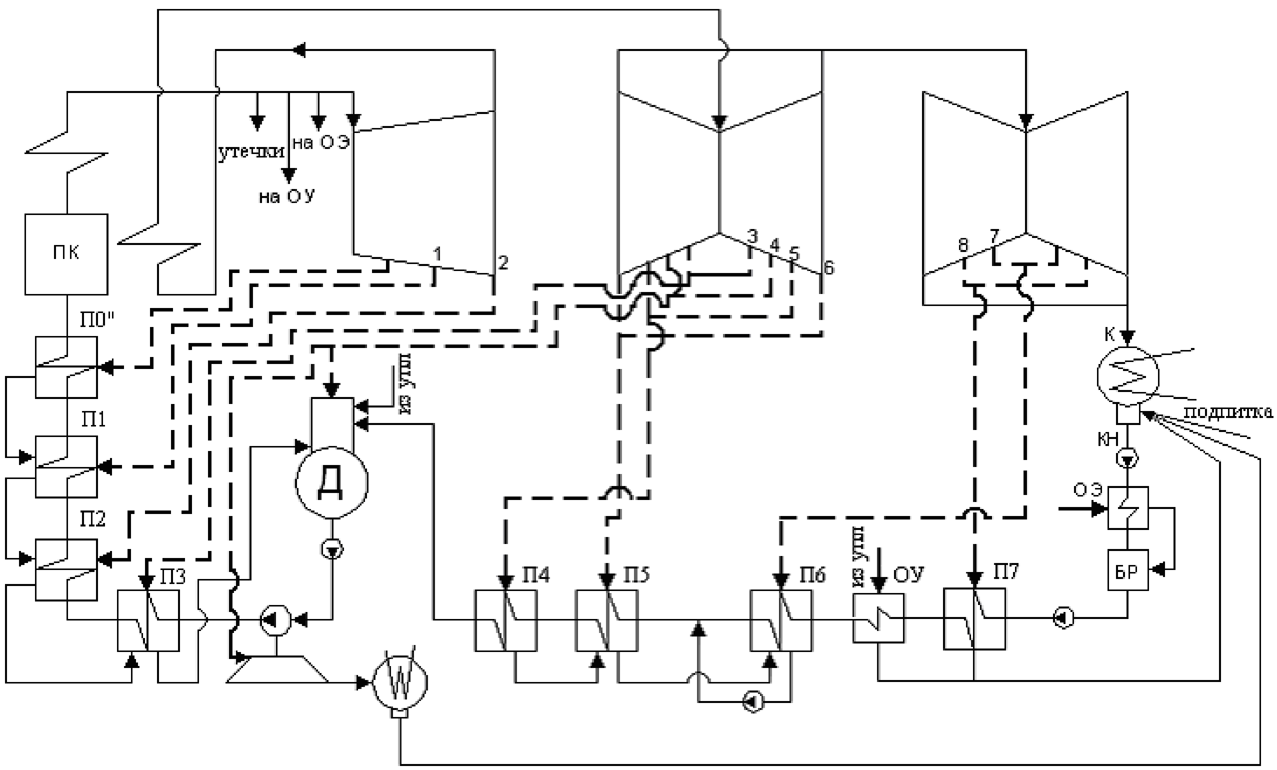


Figure 5.1.Principal thermal circuit of the turbine K-800-240 with an additional high pressure heater.

5. Calculation of the thermal block scheme with an additional high pressure heater

in a partial load operation (60%)

5.1. Calculation of steam parameters for selection

We accept the feed water temperature at the outlet of additional heater (6. 13):

$$t_{пв.0} = t_{S.0} - \Theta_0 = 266{,}5^oC;$$

where: $t_{S.0}, {}^oC -$ the saturation temperature in the additional heater;

$\Theta_0 -$ subcooling in the additional heater, we accept = 5 ° C.

$t_{S.0} = t_{пв} + \Theta_0 = 266,5 + 5 = 271,5^o C.$

The enthalpy of the feedwater at the outlet of the heater П0:

$$h_{в.вых.0} = 1188,8 \frac{кДж}{кг}.$$

The steam pressure in the heater:

$$P_{П.0} = f(t_S) = f(271,5°C) = 5,63 МПа.$$

The steam pressure in the selection:

$$P_{отб.0} = 1,05 \cdot P_{П.0} = 1,05 \cdot 5,63 = 5,91 МПа.$$

Enthalpy of drainage heater П0:

$$h_{др.п.0} = 1208,7 \frac{кДж}{кг}.$$

Parameters of steam selection:

$$t_{отб.0} = 438,3^o C;$$

$$h_{отб.0} = 3275,5 \frac{кДж}{кг}.$$

5.2. Calculation of high pressure heaters

The equation of thermal balance for the heater П0:

$$G_{п.0} \cdot (h_{отб.0} - h_{др.п.0}) \cdot \eta_п = G_{пв} \cdot (h_{в.вых.п.0} - h_{в.вх.п.0}).$$

The steam consumption in the heater П0:

$$G'_{п.0} = \frac{G_{пв} \cdot \left(h_{в.вых.п.0} - h_{в.вх.п.0}\right)}{\left(h_{отб.0} - h_{др.п.0}\right) \cdot \eta_п} = \frac{414{,}54 \cdot (1188{,}8 - 1031{,}3)}{(3275{,}5 - 1208{,}7) \cdot 0{,}98} = 32{,}24 \frac{кг}{с}.$$

After clarification of the initial steam flow through the turbine by the approximate interpolation, we have $G_0 = 449{,}49\, кг/с$.

$$G_{п.0} = \frac{G_{пв} \cdot \left(h_{в.вых.п.0} - h_{в.вх.п.0}\right)}{\left(h_{отб.0} - h_{др.п.0}\right) \cdot \eta_п} = \frac{451{,}02 \cdot (1188{,}8 - 1031{,}3)}{(3275{,}5 - 1208{,}7) \cdot 0{,}98} = 35{,}07 \frac{кг}{с}.$$

The equation of thermal balance for the heater П1:

$$\left(G_1^П \cdot \left(h_1 - h_{др1}\right) + G_0^П \cdot \left(h_{др0} - h_{др1}\right)\right) \cdot \eta_п = G_{пв} \cdot \left(h_{в.вых1} - h_{в.вх1}\right).$$

The steam consumption in the heater П1:

$$G_1 = G_1^П = \frac{\dfrac{G_{пв} \cdot \left(h_{в.вых1} - h_{в.вх1}\right)}{\eta_п} - G_0^П \cdot \left(h_{др0} - h_{др1}\right)}{\left(h_1 - h_{др1}\right)} =$$

$$= \frac{\dfrac{451{,}02 \cdot (1031{,}3 - 920{,}7)}{0{,}98} - 35{,}07 \cdot (1208{,}7 - 1056{,}2)}{(3156{,}3 - 1056{,}2)} = 21{,}69 \frac{кг}{с}.$$

The equation of thermal balance for the heater П2:

$$\left(G_2^П \cdot \left(h_2 - h_{др2}\right) + G_1 \cdot \left(h_{др1} - h_{др2}\right)\right) \cdot \eta_п = G_{пв} \cdot \left(h_{в.вых2} - h_{в.вх2}\right).$$

The steam consumption in the heater:

$$G_2 = G_1^П = \frac{\dfrac{G_{пв} \cdot \left(h_{в.вых2} - h_{в.вх2}\right)}{\eta_п} - G_1 \cdot \left(h_{др1} - h_{др2}\right)}{\left(h_2 - h_{др2}\right)} =$$

$$= \frac{\dfrac{451{,}02 \cdot (920{,}7 - 790{,}2)}{0{,}98} - 21{,}69 \cdot (1056{,}2 - 930{,}5)}{(3053{,}2 - 930{,}5)} = 27{,}01 \frac{кг}{с}.$$

The equation of thermal balance for the heater П3:

$$\left(G_3^П \cdot \left(h_3 - h_{др3}\right) + G_2 \cdot \left(h_{др2} - h_{др3}\right)\right) \cdot \eta_п = G_{пв} \cdot \left(h_{в.вых3} - h_{в.вх3}\right).$$

The steam consumption in the heater:

$$G_3^П = \frac{\dfrac{G_{пв} \cdot \left(h_{в.вых3} - h_{в.вх3}\right)}{\eta_п} - G_2 \cdot \left(h_{др2} - h_{др3}\right)}{\left(h_3 - h_{др3}\right)} =$$

$$= \frac{\dfrac{450{,}02 \cdot (790{,}2 - 739{,}2)}{0{,}98} - 27{,}01 \cdot (930{,}5 - 791{,}4)}{(3406{,}0 - 791{,}4)} = 7{,}54\,\frac{кг}{c}.$$

5.3. Feedwater deaerator

The equation of the deaerator material balance:

$$G_{пв} + G_{выпар} = G_{др3} + G_д + G_{ок4},\ \frac{кг}{c};$$

$$G_{др3} = G_{др0} + G_{др1} + G_{др2} + G_{др3} = 35{,}07 + 21{,}69 + 27{,}01 + 7{,}54 = 91{,}31\,\frac{кг}{c};$$

$$G_{выпар} = \alpha_э \cdot G_{0x} = 0{,}002 \cdot 443{,}67 = 0{,}88\,\frac{кг}{c};$$

$$G_{ок4} = 359{,}59 - G_д.$$

Heat balance equation of deaerator Д:

$$G_{пв} \cdot h_S^д = \left(G_{др3} \cdot h_{др3} + G_д \cdot h_3 + G_{ок4} \cdot h_{в.вых4} - G_{выпар} \cdot h_{выпар}\right) \cdot \eta_{см}.$$

Solving the equations of heat and material balances, we find the flow of steam to the deaerator and condensate flow through the heater П4:

$$G_{ок4} = 357{,}20\ \ \frac{кг}{c};$$

$$G_д = 2{,}39\ \ \frac{кг}{c}.$$

The steam consumption in the 3rd selection:

$$G_3 = G_3^П + G_д = 7{,}54 + 2{,}39 = 9{,}93\,\frac{кг}{c}.$$

5.4. Turbine-driven feed pump

The feed water temperature increase in the feed pump:

$$\Delta h_{ТП} = \frac{\upsilon' \cdot (P_{ТН} - P_{д}) \cdot 10^3}{\eta_{нi}},$$

where: $\upsilon' -$ specific volume of water at the outlet of the deaerator;

$\eta_{нi} -$ "Internal" (hydraulic) pump efficiency (6. 13-2);

$$\Delta h_{ТП} = \frac{0{,}001108 \cdot (33{,}0 - 0{,}7) \cdot 10^3}{0{,}85} = 42{,}1 \frac{кДж}{кг}.$$

The enthalpy of the feedwater behind the pump:

$$h_{пв.н} = h_S^{д} + \Delta h_{ТП} = 697{,}1 + 42{,}1 = 739{,}2 \frac{кДж}{кг}.$$

Water parameters for the feed pump:

$$P_{ТП} = 33{,}0 МПа; \; t_{пв.н} = 170{,}5°C.$$

Расход пара на турбопривод питательного насоса:

Steam consumption for turbine drive of feed pump

$$G_{ТП} = \frac{G_{пв}^{x} \cdot h_{н}^{a}}{H_{i}^{ТП} \cdot \eta_{н} \cdot \eta_{м}} = \frac{451{,}02 \cdot 35{,}53}{534{,}5 \cdot 0{,}833 \cdot 0{,}98} = 36{,}72 \frac{кг}{с};$$

where: $h_{н}^{a} = \upsilon' \cdot (p_{н} - p_{в}) \cdot 10^3 = 0{,}001108 \cdot (33 - 0{,}7) \cdot 10^3 = 35{,}53 \, кДж/кг$ – adiabatic compression work in the pump;

$H_{ТП} = 3236{,}1 - 2701{,}6 = 534{,}5 \, кДж/кг -$ действительное теплопадение пара в приводной турбине питательного насоса

the actual heat drop of the steam in turbine driven feed pump;

$\eta_{н} = \eta_{нi} \cdot \eta_{н}' = 0{,}85 \cdot 0{,}98 = 0{,}833 -$

total efficiency of the pump with volumetric and mechanical losses (6 13-4);

$\eta_м -$ mechanical efficiency of the turbine drive (6 13-4).

Мощность турбопривода питательного насоса:

Turbo drive power of the feed pump:

$$N_{ТП}^{x} = G_{ТП} \cdot H_i^{ТП} = 36{,}72 \cdot 534{,}5 \cdot 10^{-3} = 19{,}63 МВт.$$

5.5. Low pressure heaters

The equation of thermal balance for the heater П4:

$$G_4^{П} \cdot \left(h_5 - h_{др5}\right) \cdot \eta_п = G_{ок4} \cdot \left(h_{в.вых4} - h_{в.вх4}\right).$$

The steam consumption in the heater:

$$G_5 = G_4^{П} = \frac{G_{ок4} \cdot \left(h_{в.вых4} - h_{в.вх4}\right)}{\left(h_5 - h_{др5}\right) \cdot \eta_п} = \frac{357{,}20 \cdot (543{,}7 - 431{,}5)}{\left(3078{,}0 - 552{,}5\right) \cdot 0{,}98} = 16{,}19 \frac{кг}{с}.$$

The equation of thermal balance for П5:

$$\left(G_5^{П} \cdot \left(h_6 - h_{др6}\right) + G_5 \cdot \left(h_{др5} - h_{др6}\right)\right) \cdot \eta_п = G_{ок5} \cdot \left(h_{в.вых5} - h_{в.вх5}\right).$$

The steam consumption in the heater:

$$G_6 = G_5^{П} = \frac{\dfrac{G_{ок5} \cdot \left(h_{в.вых5} - h_{в.вх5}\right)}{\eta_п} - G_5 \cdot \left(h_{др5} - h_{др6}\right)}{\left(h_6 - h_{др6}\right)} =$$

$$= \frac{\dfrac{357{,}2 \cdot \left(431{,}5 - 343{,}2\right)}{0{,}98} - 16{,}19 \cdot \left(552{,}5 - 439{,}3\right)}{\left(2898{,}0 - 439{,}3\right)} = 12{,}34 \frac{кг}{с}.$$

Material balance of mixing point:

$$G_{\partial p6} = G_5 + G_6 + G_7 = 16{,}19 + 12{,}34 + 13{,}57 = 42{,}1 \,{}^{\kappa г}\!/_{c};$$

$$G_{\partial p6} + G_{o\kappa 6} = G_{o\kappa 5} = G_{o\kappa 4}.$$

The equation of the heat balance of the mixing point:

$$\left(G_{\partial p6} \cdot h_{\partial p7} + G_{o\kappa 6} \cdot h_{в.вых6}\right) \cdot \eta_{см} = G_{o\kappa 5} \cdot h_{в.вх5}.$$

Solving the equation for the mixing point and the heater П6:

heat balance equation for heater П6:

$$\left(G_6^{П} \cdot \left(h_7 - h_{\partial p7}\right) + G_{\partial p5} \cdot \left(h_{\partial p6} - h_{\partial p7}\right)\right) \cdot \eta_п = G_{o\kappa 6} \cdot \left(h_{в.вых6} - h_{в.вх6}\right);$$

steam flow to the heater:

$$G_7 = G_6^{П} = \frac{\dfrac{G_{o\kappa 6} \cdot \left(h_{в.вых6} - h_{в.вх6}\right)}{\eta_п} - G_{\partial p6} \cdot \left(h_{\partial p6} - h_{\partial p7}\right)}{\left(h_7 - h_{\partial p7}\right)} =$$

$$= \frac{\dfrac{328{,}67 \cdot \left(334{,}4 - 224{,}7\right)}{0{,}98} - 42{,}1 \cdot \left(439{,}3 - 340{,}5\right)}{\left(2745{,}3 - 340{,}5\right)} = 13{,}57\,\frac{\kappa г}{c}.$$

The equation of thermal balance for the heater П7:

$$G_7^{П} \cdot \left(h_8 - h_{\partial p8}\right) \cdot \eta_п = G_{o\kappa 7} \cdot \left(h_{в.вых7} - h_{в.вх7}\right).$$

The steam consumption in the heater:

$$G_8 = G_7^{П} = \frac{G_{o\kappa 7} \cdot \left(h_{в.вых7} - h_{в.вх7}\right)}{\left(h_8 - h_{\partial p8}\right) \cdot \eta_п} = \frac{328{,}67 \cdot (212{,}2 - 94{,}0)}{\left(2613{,}74 - 220{,}0\right) \cdot 0{,}98} = 16{,}56\,\frac{\kappa г}{c}.$$

5.6. Material balance of steam consumption in the condenser

Steam flow to the condenser by the material balance of the turbine:

$$G_{кх}^{T} = G_0^x - \sum_{i=1}^{9} G_{отб.i}^x = 433{,}67 -$$

$$-\left(35{,}07 + 21{,}69 + 27{,}01 + 9{,}93 + 36{,}72 + 16{,}19 + 12{,}34 + 13{,}57 + 16{,}56\right) = 244{,}59\, {кг}/{с}.$$

Steam flow to the condenser of the capacitor balance: 298,54

$$G_{кх}^{К} = G_{вх.Э}^x - G_{ТП} - G_{отб.8}^x - \alpha_{выпар} \cdot G_{пв}^x - a_э \cdot G_{пв}^x - \alpha_{дв} \cdot G_{пв}^x =$$

$$= 315{,}919 - 36{,}72 - 16{,}56 - 0{,}028 \cdot 451{,}02 - 0{,}002 \cdot 451{,}02 - 0{,}01 \cdot 451{,}02 = 244{,}59\, {кг}/{с}.$$

We determine the underworking and relative costs of steam for the selections and enter them in Table 5.1.

Таблица 5.1.

№	$G_{отб.i}$	α_i	y_i	$\alpha_i \cdot y_i$
0´´	35,07	0,0777	0,8493	0,06599
1	21,69	0,0481	0,7670	0,03689
2	27,01	0,0599	0,6966	0,04173
3	7,54	0,0167	0,5934	0,00991
Д	2,39	0,0053	0,5934	0,00314
ТП	36,72	0,0846	0,4774	0,04042
5	16,19	0,0359	0,3694	0,01326
6	12,34	0,0273	0,2465	0,00674
7	13,57	0,0301	0,1422	0,00428
8	16,56	0,0367	0,0524	0,00192

We calculate the power of the turbine:

$$N_Э = \frac{G_0^x \cdot \left[H_i \cdot \eta_м \cdot \eta_г \cdot \left(1 - \sum \alpha_i \cdot y_i\right)\right]}{10^3} =$$

$$= \frac{433{,}67 \cdot \left[1464{,}5 \cdot 0{,}982 \cdot 0{,}98 \cdot \left(1 - 0{,}22430\right)\right]}{10^3} = 483{,}79 MBm.$$

5.7. Determination of heat efficiency in partial mode with additional high pressure heater

Heat let in the steam generator:

$$Q_{ПГ} = G_{пв}^x \cdot \left(h_{0x}^{ПГ} - h_{пв}\right) + \Delta h_{пп} \cdot G_{пп};$$

where:

$$G_{пп} = G_0 - G_0^П - G_1 - G_2 = 433{,}67 - 35{,}07 - 21{,}69 - 27{,}01 = 349{,}90;$$

$$Q_{ПГ} = \frac{451{,}02 \cdot \left(3386{,}9 - 1188{,}8\right) + 504 \cdot 349{,}9}{10^3} = 1167{,}74 MBm.$$

Heat supplied to the turbine:

$$Q_{ТУ} = G_{0.x} \cdot \left(h_{0.x} - h_{пв}\right) + \Delta h_{пп} \cdot G_{пп};$$

$$Q_{ТУ} = \frac{433{,}67 \cdot \left(3387{,}4 - 1188{,}8\right) + 504 \cdot 349{,}9}{10^3} = 1129{,}82 MBm.$$

Electrical efficiency of the set:

$$\eta_{ту}^Э = \frac{N_Э + N_{ТП}}{Q_{ТУ}} = \frac{483{,}79 + 19{,}63}{1129{,}82} = 0{,}446.$$

Efficiency of transport:

$$\eta_{ТР} = \frac{Q_{ТУ}}{Q_{ПГ}} = \frac{1129{,}82}{1167{,}74} = 0{,}968.$$

Effective efficiency for electricity output:

:

$$\eta_e^э = \eta_{ту}^э \cdot \eta_{ТР} \cdot \eta_{ПГ} \cdot \left(1 - k_{сн}\right);$$

where: $\eta_{ПГ} = 0{,}9$ – efficiency of the steam generator (SG).

$k_{сн} = 0{,}03$ – coefficient that takes into account electricity consumption for its own needs, we accept;

$$\eta_e^э = 0{,}446 \cdot 0{,}968 \cdot 0{,}925 \cdot (1 - 0{,}03) = 0{,}387.$$

Equivalent fuel consumption per block:

$$B_у = \frac{Q_{ПГ}}{\eta_{ПГ} \cdot Q_р^н} = \frac{1167740}{0{,}925 \cdot 32491{,}4} = 38{,}85 \frac{кг}{с}.$$

where: $Q_р^н = 32491{,}4 \frac{кДж}{кг}$ – calorific value of fuel equivalent.

Specific fuel consumption for electric power generation in the unit:

$$b_Э^{отп} = \frac{123}{\eta_{ТУ}^э} = \frac{123}{0{,}446} = 275{,}78 \frac{г.у.т}{кВт \cdot ч}.$$

Specific fuel consumption per generated kWh:

$$b_Э^{отп} = \frac{123}{\eta_e^э} = \frac{123}{0{,}387} = 317{,}53 \frac{г.у.т.}{кВт \cdot ч}.$$

The fuel consumption increased by $11{,}03 \frac{г.у.т.}{кВт \cdot ч}$ compared to the nominal mode, and decreased by $18{,}54 \frac{г.у.т.}{кВт \cdot ч}$ compared to the partial mode without additional HPH.

6. Analysis of the results of the calculation

We recalculate the pressure using the Stodola-Flugel formula to determine the steam selection of additional HPH:

$$P_{П0''} = \sqrt{\left(\frac{G_{0''-1x}}{G_{1-2н}}\right)^2 \cdot \left(P_{0'н}^2 - P_{1н}^2\right) + P_{1x}^2};$$

$$P_{П0''} = \sqrt{\left(\frac{376{,}91}{617{,}02}\right)^2 \cdot \left(15{,}74^2 - 5{,}97^2\right) + 3{,}04^2} = 9{,}40 МПа;$$

where: $P_{0'н} = 15{,}74 МПа -$ steam pressure after the regulating stage (6. Table 5-6).

Analyzing the obtained results of the calculation, it can be concluded that the most optimal place for the selection of steam for the additional (fourth) HPH is the option, when steam is taken behind the fifth stage of the turbine's high-pressure cylinder at a pressure of 9.23 MPa in the nominal mode (6. Table 5-6). At the same time, the steam flow rate to the turbine increased by an amount equal to the steam flow rate to the additional heater. This makes it possible to obtain the maximum possible efficiency of the unit with an additional selection of steam from the turbine's HPC in the partial load mode. In this case, the relative steam flow to the fourth HPH is $\alpha_i = 0{,}0808$, the feed water temperature after the fourth HPH reaches the feedwater temperature of the nominal regime before the boiler.

Improvement of the technical and economic performance of the unit with the installation and connection of additional LDPE allowed reducing the specific consumption of conventional fuel and increasing the efficiency of the unit in partial mode.

7. Design and calculation of additional high pressure heater

7.1. Thermal calculation of HPH

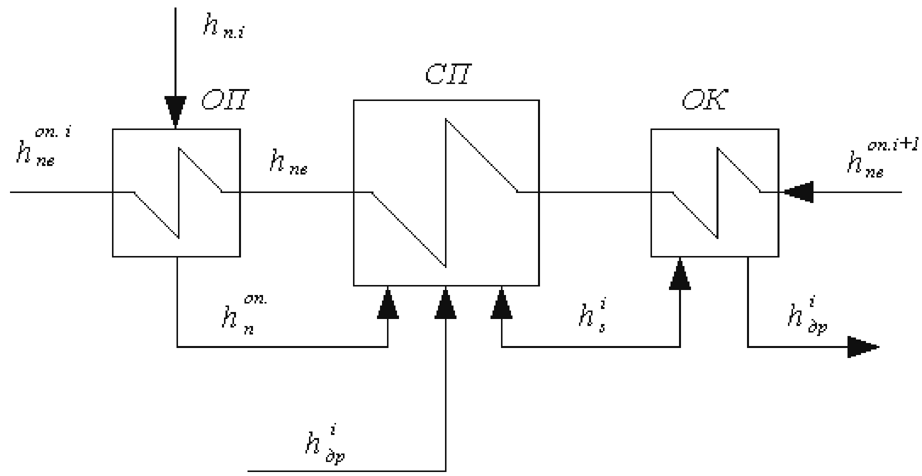


Figure 7.3. Scheme of HPH with steam and drain coolers.

7.1.1. Thermal loads of different HPH zones

Steam consumption per 1 heater:

$$G_n^{1n} = \frac{G_n}{2} = \frac{35{,}07}{2} = 17{,}54\,\frac{\text{кг}}{c}.$$

Thermal load of the cooler zone:

$$Q_{on} = G_n^{1n} \cdot \left(h_n - h_n^{no}\right) \cdot \eta_n = 17{,}54 \cdot \left(3275{,}5 - 2870{,}7\right) \cdot 0{,}98 = 6956{,}2\,\text{кВт}.$$

The thermal load of the zone of the heater itself:

$$Q_{cn} = G_n^{1n} \cdot \left(h_n^{no} - h_s\right) \cdot \eta_n = 17{,}54 \cdot \left(2870{,}7 - 1208{,}7\right) \cdot 0{,}98 = 28560{,}3\,\text{кВт}.$$

Thermal load of the condensate cooler zone:

$$Q_{ок} = G_п^{1п} \cdot \left(h_s - h_{др}\right) \cdot \eta_п = 17{,}54 \cdot \left(1208{,}7 - 1077{,}5\right) \cdot 0{,}98 = 2255{,}2 кВт.$$

7.1.1.1. Determine the water temperature at the outlet of the condenser cooler, at the inlet to the heater itself and at the outlet of the steam cooler

Consumption of feedwater through the heater itself:

$$G_{пв}^{1п} = \frac{G_{пв}}{2} = \frac{451{,}02}{2} = 225{,}51 \frac{кг}{с}.$$

A part of the feed water enters the condensate cooler at a flow rate of:

$$G_{ок} = 32{,}70 \,{}^{кг}\!/_{с} \left(14{,}5\%\right).$$

From the heat balance equation for the condensate cooler (drainage):

$$Q_{ок} = G_{ок} \cdot \left(h_{од} - h_{пв}^{по1}\right); \;\; кВт.$$

The enthalpy of water at the outlet of the condensate cooler:

$$h_{од} = h_{пв.вх.0} + \frac{Q_{ок}}{G_{ок}} = 1056{,}2 + \frac{2255{,}2}{32{,}70} = 1125{,}2 \frac{кДж}{кг}.$$

Water temperature at the outlet of the condensate cooler:

$$t_{ок}^{вых} = 254{,}3^0 C.$$

The enthalpy of water at the entrance to the heater itself:

$$h_{сп}^{вх} = h_{пв.вх.0} + \frac{G_{ок} \cdot \left(h_{од} - h_{пв.вх.0}\right)}{G_{пв}^{1п}} = 1056{,}2 + \frac{32{,}7 \cdot \left(1125{,}2 - 1056{,}2\right)}{225{,}51} = 1066{,}0 \frac{кДж}{кг}.$$

The water temperature at the inlet in heater itself:

$$t_{сп}^{вх} = 244{,}5^0 C.$$

The water flow through the steam cooler is:

$$G_{on} = 0{,}1 \cdot G_{пв}^{1п} = 0{,}1 \cdot 225{,}51 = 22{,}551 \,{}^{кг}\!/_{c}.$$

The enthalpy of water at the inlet of the steam cooler:

$$h_{on}^{вых} = h_{пв}^{cn} + \frac{Q_{on}}{G_{on}} = 1188{,}8 + \frac{6956{,}2}{22{,}551} = 1497{,}3 \frac{кДж}{кг};$$

where: $h_{пв}^{cn} = 1188{,}8\,кДж/кг$ $-$ enthalpy of feedwater at the outlet from the heater itself.

Water temperature at the outlet of the steam cooler:

$$t_{пв}^{on} = 332{,}1^{0}C.$$

7.1.2. Calculation of the average logarithmic temperature head for various HPH zones

The schematic diagram of the regenerative heater and the graph of the temperatures of the heated water and the heating steam are shown in Figure 7.1.

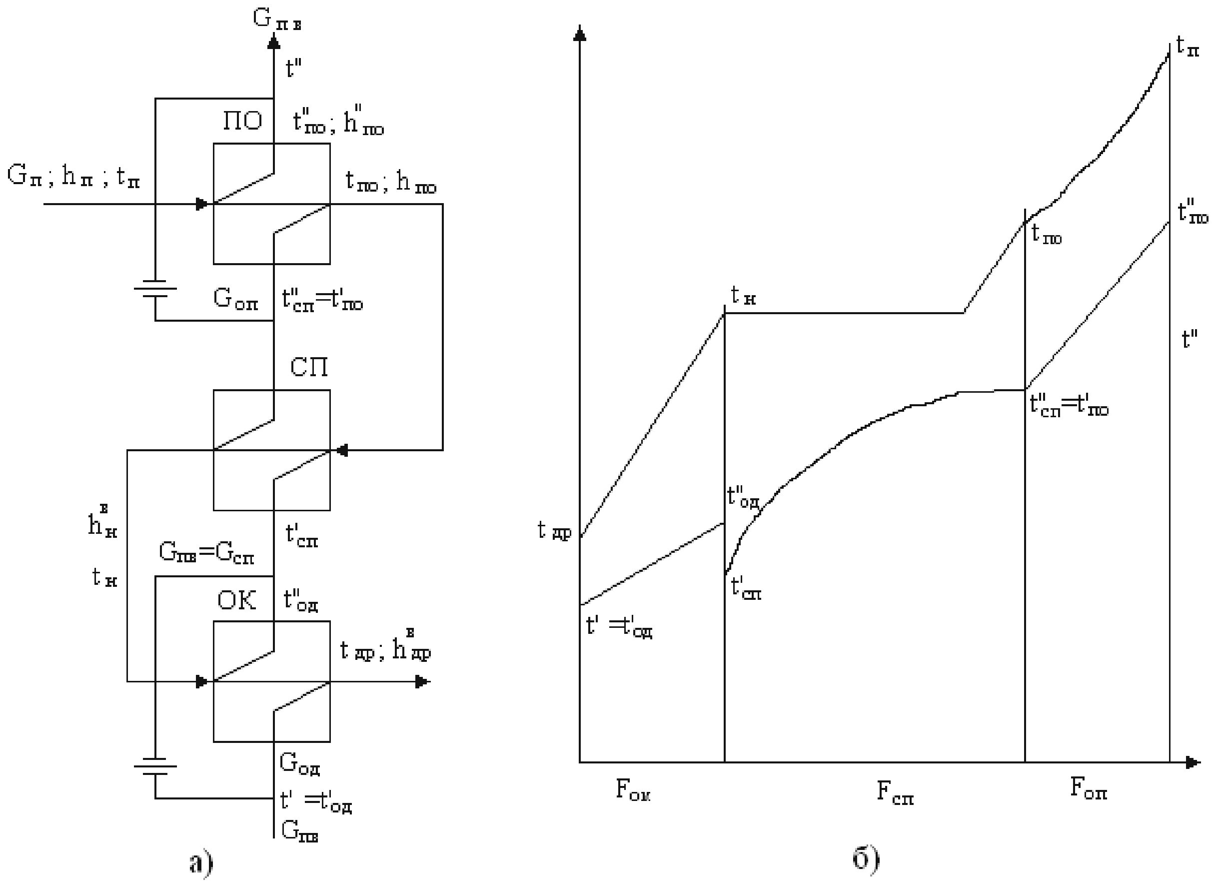


Figure 7.1. Schematic diagram of the regenerative heater (a) and the graphs of the temperatures of the heated water and the heating steam (b).

7.1.2.1. Zone of steam cooler

$$\Delta t_б = t_п - t''_{по} = 438{,}3 - 332{,}1 = 106{,}2^0 C;$$

where: $t_п = 438{,}3^0 C -$ the temperature of steam in the selection;

$t''_{по} = 332{,}1^0 C -$ Feed water temperature at the outlet of the steam cooler.

$$\Delta t_м = t_{по} - t'_{по} = 291{,}8 - 266{,}5 = 25{,}3^0 C;$$

where: $t_{по} = 291{,}8^0 C -$ steam temperature behind the steam cooler;

$t'_{по} = 266{,}5^0 C -$ The temperature of the feed water at the outlet of the heater itself..

The average logarithmic temperature head of the steam cooler zone:

$$\Delta t_{ср.л} = \frac{\Delta t_б - \Delta t_м}{\ln \frac{\Delta t_б}{\Delta t_м}} = \frac{106,2 - 25,3}{\ln \frac{106,2}{25,3}} = 56,39^0 C;$$

7.1.2.2. The zone of the heater itself

$$\Delta t_б = t_s - t'_{сп} = 271,5 - 244,5 = 27,0^0 C;$$

where: $t_s = 271,5^0 C -$ steam saturation temperature;

$t'_{сп} = 244,5^0 C$ – Feed water temperature at the inlet to the heater itself.

$$\Delta t_м = t_s - t''_{сп} = 271,5 - 266,5 = 5^0 C;$$

where: $t''_{сп} = 266,5^0 C -$ the temperature of the feed water at the output from the heater itself.Среднелогарифмический температурный напор зоны собственно подогревателя:

$$\Delta t_{ср.л} = \frac{\Delta t_б - \Delta t_м}{\ln \frac{\Delta t_б}{\Delta t_м}} = \frac{27,0 - 5}{\ln \frac{27,0}{5}} = 13,05^0 C;$$

7.1.2.3. Condenser cooling zone

$$\Delta t_б = t_s - t''_{ок} = 271,5 - 254,3 = 17,2^0 C;$$

where: $t''_{ок} = 254,3^0 C -$ feed water temperature at the outlet of the condensate cooler.

$$\Delta t_м = t_{др} - t'_{ок} = 246{,}7 - 237{,}3 = 9{,}4^0 C;$$

where: $t''_{др} = 246{,}7^0 C -$ drainage temperature of the heating steam at the outlet of the condensate cooler;

$t'_{ок} = 237{,}3^0 C -$ Feed water temperature at the condensate cooler inlet.

Average logarithmic temperature head of the condensate cooler zone:

$$\Delta t_{ср.л} = \frac{\Delta t_б - \Delta t_м}{\ln \frac{\Delta t_б}{\Delta t_м}} = \frac{17{,}2 - 9{,}4}{\ln \frac{17{,}2}{9{,}4}} = 12{,}9^0 C;$$

7.1.3 Thermal calculation of the heater itself

The purpose of the calculation is to determine the heat transfer coefficient of the heater itself, as well as its geometric dimensions.

To determine the coefficient of heat transfer from the walls of pipes to water, it is necessary to set the mode of its movement.

The speed of water in the heater tubes is assumed to be within $1{,}3 - 1{,}8 м/с$; we take $w_в = 1{,}3 м/с$.

7.1.3.1. Calculation of the heat transfer coefficient from the pipe walls to water

Average temperature of water in the zone of the heater itself:

$$t_{в.ср} = \frac{t'_{сп} - t''_{сп}}{2} = \frac{244{,}5 + 266{,}5}{2} = 255{,}5^0 C.$$

With $t_{в.ср} = 255{,}5^o C$ determine the thermophysical parameters of water (5. Table 1):

$\mu_{ж} = 1111{,}43 \cdot 10^{-7} \frac{кг}{м \cdot с} -$ specific dynamic viscosity of water;

$\Pr_{ж} = 0{,}7932 -$ Prandtl number;

$\lambda_{ж} = 645{,}66 \cdot 10^{-3} \ \frac{Вт}{м \cdot {}^0 C} -$ coefficient of water thermal conductivity;

$\vartheta_{ж} = 0{,}0012787 \frac{м^3}{кг} -$ specific volume of water;

$\nu_{ж} = 1{,}353 \cdot 10^{-7} \ \frac{м^2}{с} -$ coefficient of water kinematic viscosity.

Determine the regime of water flow:

$$\mathrm{Re}_{ж} = \frac{w_в \cdot d_{вн}}{\nu_{ж}} = \frac{1{,}3 \cdot 0{,}02}{1{,}353 \cdot 10^{-7}} = 1{,}9217 \cdot 10^5.$$

With $\mathrm{Re}_{ж} > 10^4$ the coefficient of heat transfer from the wall to the water:

$$a_2 = 0{,}023 \cdot \mathrm{Re}_{ж}^{0,8} \cdot \Pr_{ж}^{0,43} \cdot \frac{\lambda_{ж}}{d_{вн}} \cdot \varepsilon_{сп} =$$

$$= 0{,}023 \cdot \left(1{,}9217 \cdot 10^5\right)^{0,8} \cdot 0{,}7932^{0,43} \cdot \frac{0{,}64566}{0{,}02} \cdot 1{,}35 = 15301{,}0 \ \frac{Вт}{м^2 \cdot К};$$

where: $\varepsilon_{сп} = 1{,}35 -$ correction for the flow turbulence due to its rotation in the spiral tubes, obtained from the calculation:

$$\varepsilon_{сп} = 1 + 11{,}11 \cdot \frac{D \cdot n_в \cdot n_п}{I_{сп}} = 1 + 11{,}11 \cdot \frac{0{,}06 \cdot 8 \cdot 1}{15} = 1{,}35;$$

where: $D_{вн} = 0{,}06 м -$ inner diameter of the smallest spiral turn;

$n_в = \sqrt{\frac{\beta \cdot I_{сп}}{2 \cdot \pi \cdot t} + x^2} - x = \sqrt{\frac{0{,}98 \cdot 15}{2 \cdot 3{,}14 \cdot 0{,}032} + 0{,}44^2} - 0{,}44 = 8 -$ number of spiral turns;

$\beta = 0{,}98 -$ Coefficient that takes into account the fact that not the entire length of the tube is curled into a spiral;

$I_{сп} = 15 м -$ The previously adopted length of the spiral;

$t = 0{,}032 м -$ distance between tubes;

$$x = 0,5 \cdot \left(\frac{D_{вн}}{t} - 1 \right) = 0,5 \cdot \left(\frac{0,06}{0,032} - 1 \right) = 0,44;$$

$n_п = 1 -$ Number of planes at the spiral tube (single spiral).

7.1.3.2. Calculation of the heat transfer coefficient from the heating medium to the wall of the pipes

We set the coefficient of heat transfer: $K_{сп} = 2876,7 \frac{Вт}{м^2 \cdot К}$.

Specific heat flow in the heater itself:

$$q = K_{сп} \cdot \Delta t_{ср.л} = 2876,7 \cdot 13,05 = 37540,9 \frac{Вт}{м^2}.$$

Outside tube wall temperature:

$$t_{ст}^{нар} = t'_{сп} + \frac{q}{R_{тр}} + \frac{q}{a_2} = 244,4 + \frac{37540,9}{13000} + \frac{37540,9}{15301} = 249,7^o C;$$

where: $R_{тр} = \frac{\lambda_{тр}}{\delta_{ст}} = \frac{52}{0,004} = 13000 -$ thermal resistance of the coil pipe;

$\lambda_{тр} = 52 \frac{Вт}{м \cdot К} -$ Coefficient of thermal conductivity of metal pipes;

$\delta_{ст} = 0,004 м -$ pipe wall thickness.

Condensate film temperature:

$$t_{пл}^{к} = \frac{t_s + t_{ст}^{нар}}{2} = \frac{271,5 + 249,7}{2} = 260,6^o C.$$

With $t_{пл}^{к} = 260,6^o C,$ determine the thermo physical parameters of steam condensate according to (5 of Table 1):

$\mu_к = 1016,4 \cdot 10^{-7} \frac{кг}{м \cdot с} -$ Specific dynamic viscosity of water;

$Pr_к = 0{,}8344 -$ Prandtl number;

$\lambda_к = 606{,}0 \cdot 10^{-3}\ \frac{Вт}{м \cdot {}^0C} -$ Coefficient of water Thermal conductivity;

$\vartheta_к = 0{,}00127604\ \frac{м^3}{кг} -$ Specific volume of water;

$\nu_к = 1{,}29698 \cdot 10^{-7}\ \frac{м^2}{с} -$ Coefficient of water kinematic viscosity.

We determine the mode of washing of the coil tubes with condensate steam:

$$\mathrm{Re}_{ж} = \frac{q \cdot H_{акт}}{\mu_к \cdot r} = \frac{39698{,}5 \cdot 0{,}224}{1016{,}4 \cdot 10^{-7} \cdot 1560{,}4 \cdot 10^3} = 56{,}37;$$

where: $H_{акт} = m \cdot d_{нар} = 8 \cdot 0{,}028 = 0{,}224 м -$ active pipe height;

$r = 1560{,}4 \cdot 10^3\ \frac{Дж}{кг} -$ Heat of vaporization.

Coefficient of heat transfer from the heating medium to the pipes walls:

$$a_1 = (\varepsilon_r \cdot C)^{\frac{3}{4}} \cdot \lambda_к \cdot \left(\frac{g}{(\nu_к)^2} \right) \cdot (\mathrm{Re}_к)^{-\frac{1}{3}} =$$

$$= (0{,}8 \cdot 0{,}728)^{\frac{4}{3}} \cdot 0{,}606 \cdot \left(\frac{9{,}81}{(1{,}29698 \cdot 10^{-7})^2} \right) \cdot 56{,}37^{-\frac{1}{3}} = 4482{,}9\ \frac{Вт}{м^2 \cdot К};$$

where: $\varepsilon_r = 0{,}8 -$ correction for the pipe type (pipe made of carbon steel);

$C = 0{,}728 -$ Correction for the position of the pipe (spiral).

7.1.3.3. Determination of the heat transfer coefficient

$$K'_{сп} = \left(d_{ср} \cdot \left(\frac{1}{a_1 \cdot d_н} + \frac{1}{2 \cdot \lambda_{тр}} \cdot \ln \frac{d_н}{d_{вн}} + \frac{1}{a_2 \cdot d_{вн}} \right) \right)^{-1} =$$

$$= \left(0{,}024 \cdot \left(\frac{1}{4482{,}9 \cdot 0{,}028} + \frac{1}{2 \cdot 52} \cdot \ln \frac{28}{20} + \frac{1}{15301 \cdot 0{,}02} \right) \right)^{-1} = 2879{,}55\ \frac{Вт}{м^2 \cdot К}.$$

Computational error:

$$\delta = \frac{K'_{сп} - K_{сп}}{K_{сп}} \cdot 100\% = \frac{2879{,}55 - 2876{,}7}{2879{,}55} \cdot 100\% = 0{,}10\%.$$

Area of heat exchange of the heater itself:

$$F_{сп} = \frac{Q_{сп}}{K'_{сп} \cdot \Delta t_{ср.л}} = \frac{28560{,}3}{2{,}87955 \cdot 13{,}08} = 758{,}3 м^2.$$

Number of spirals in the heater itself:

$$N_{спир} = \frac{G^{1п}_{пв} \cdot \vartheta_в}{d^2_{вн} \cdot w_в} = \frac{225{,}51 \cdot 0{,}00127931}{0{,}02^2 \cdot 1{,}3} = 554{,}8 шт;$$

We take the number of spirals as multiple of the number of sections and the number of rows in each section:

$$8 \cdot 12 = 96.$$

Consequently, the number of spirals in the heater itself:

$$N_{спир} = 576 \quad шт.$$

Length of each spiral:

$$I_{сп} = \frac{F_{сп}}{\pi \cdot d_{ср} \cdot N_{спир}} = \frac{758{,}3}{3{,}14 \cdot 0{,}024 \cdot 576} = 13{,}28 м.$$

7.1.4. Thermal calculation of steam cooler

Thermal load of steam cooler:

$$Q_{оп} = 6956{,}2 \quad кВт \ .$$

Steam consumption:

$$G^{1п}_{п} = 17{,}54 \frac{кг}{с}.$$

Sectional area for steam passage:

$$F = I_{cn} \cdot 0{,}004 \cdot \beta = 13{,}28 \cdot 0{,}004 \cdot 0{,}98 = 0{,}0522 \ \text{м} \ .$$

7.1.4.1. Calculation of coefficient of steam emission from hot surroundings to the wall of the pipe.

Average temperature of water in coolant steam-system:

$$t_{cp} = \frac{t_n + t_{no}}{2} = \frac{438{,}3 + 291{,}5}{2} = 364{,}9^o C.$$

for $t_{cp} = 364{,}9^o C$ Definition of thermo-physical parameters of steam in coolant in (5 тable.1):

$\mu_n = 227{,}93 \cdot 10^{-7} \dfrac{кг}{м \cdot c}$ − Specific dynamic volume of steam property;

$\Pr_n = 1{,}064$ − Pradtles number;

$\lambda_n = 57{,}34 \cdot 10^{-3} \dfrac{Bm}{м \cdot {}^0 C}$ − Coefficient of heat capacity of steam;

$\vartheta_n = 0{,}04712 \dfrac{м^3}{кг}$ − Specific heat capacity;

$\nu_n = 10{,}74 \cdot 10^{-7} \dfrac{м^2}{c}$ − Coefficient of kinematic property of stem to the velocity of steam in coolant system.

Velocity of steam in coolant system:

$$w_n = \frac{G_n^{1n} \cdot \vartheta_n}{F} = \frac{17{,}54 \cdot 0{,}04712}{0{,}0522} = 15{,}83 \frac{м}{c}.$$

Equivalent diameter:

$$d_{экв} = \frac{4 \cdot F}{2} = \frac{4 \cdot 0{,}0522}{2} = 0{,}1044 м.$$

Determination of the duration of steam in coolant system:

$$Re_п = \frac{w_п \cdot d_{экв}}{\nu_п} = \frac{15{,}83 \cdot 0{,}1044}{10{,}74 \cdot 10^{-7}} = 1{,}539 \cdot 10^6.$$

Coefficient of heat emission form hot surroundings to the wall of the pipe:

$$a_1 = 0{,}027 \cdot Re_п^{0{,}84} \cdot Pr_п^{0{,}4} \cdot \frac{\lambda_п}{d_{экв}} =$$

$$= 0{,}027 \cdot \left(1{,}539 \cdot 10^6\right)^{0{,}84} \cdot 1{,}064^{0{,}4} \cdot \frac{57{,}34 \cdot 10^{-3}}{0{,}1152} = 2169{,}83 \frac{Вт}{м^2 \cdot К}.$$

7.1.4.2. Calculation of heat emission from pipe to water

Average temperature of water in coolant-steam system:

$$t_{ср} = \frac{t'_{по} + t''_{по}}{2} = \frac{266{,}5 + 332{,}1}{2} = 299{,}3^o C.$$

For $t_{ср} = 299{,}3^o C$ determining thermo-physical parameters of water in (5тabl.1):

$\mu_{ж} = 942{,}55 \cdot 10^{-7} \frac{кг}{м \cdot с}$ – Specific dynamic property of water;

$Pr_{ж} = 0{,}79976$ – Pradtles number;

$\lambda_{ж} = 591{,}19 \cdot 10^{-3} \frac{Вт}{м \cdot {}^0C}$ – Coefficient of heat conduction of water;

$\vartheta_{ж} = 0{,}0013225 \frac{м^3}{кг}$ – Specific volume of water;

$\nu_{ж} = 1{,}2466 \cdot 10^{-7} \frac{м^2}{с}$ – Coefficient of kinematic property of water.

Determine the duration of water regime:

$$Re_{ж} = \frac{w_в \cdot d_{вн}}{\nu_{ж}} = \frac{1{,}3 \cdot 0{,}02}{1{,}2466 \cdot 10^{-7}} = 2{,}086 \cdot 10^5.$$

Coefficient of steam emission from walls of pipe to watere:

$$a_2 = 0{,}023 \cdot \mathrm{Re}_{ж}^{0,8} \cdot \mathrm{Pr}_{ж}^{0,43} \cdot \frac{\lambda_{ж}}{d_{вн}} \cdot \varepsilon_{сп} =$$

$$= 0{,}023 \cdot \left(2{,}086 \cdot 10^5\right)^{0,8} \cdot 0{,}79976^{\,0,43} \cdot \frac{0{,}59119}{0{,}02} \cdot 1{,}35 = 15013{,}5 \frac{Вт}{м^2 \cdot К};$$

7.1.4.3. Determination of the heat transfer coefficient

$$K_{оп} = \left(d_{ср} \cdot \left(\frac{1}{a_1 \cdot d_н} + \frac{1}{2 \cdot \lambda_{тр}} \cdot \ln \frac{d_н}{d_{вн}} + \frac{1}{a_2 \cdot d_{вн}} \right) \right)^{-1} =$$

$$= \left(0{,}024 \cdot \left(\frac{1}{2169{,}83 \cdot 0{,}028} + \frac{1}{2 \cdot 52} \cdot \ln \frac{28}{20} + \frac{1}{15013{,}5 \cdot 0{,}02} \right) \right)^{-1} = 1809{,}62 \frac{Вт}{м^2 \cdot К}.$$

Area of heat exchange of the steam cooler:

$$F_{оп} = \frac{Q_{оп}}{K_{оп} \cdot \Delta t_{ср.л}} = \frac{6956{,}2}{1809{,}62 \cdot 56{,}39} = 68 м^2.$$

Number of spirals in the steam cooler:

$$N_{спир} = \frac{G_{оп} \cdot \vartheta_в}{d_{вн}^2 \cdot w_в \cdot 0{,}785} = \frac{225{,}51 \cdot 0{,}0012466}{0{,}02^2 \cdot 1{,}3 \cdot 0{,}785} = 69 шт;$$

The length of each spiral in the steam cooler:

$$I_{сп} = \frac{F_{оп}}{\pi \cdot d_{ср} \cdot N_{спир}} = \frac{68}{3{,}14 \cdot 0{,}024 \cdot 69} = 13{,}2 \ м.$$

7.1.5. Thermal calculation of condensate cooler

7.1.5.1. Calculation of the heat transfer coefficient from the pipe walls to water

Average temperature of water in the condensate cooler zone:

$$t_{cp} = \frac{t'_{o\kappa} + t''_{o\kappa}}{2} = \frac{246{,}7 + 254{,}3}{2} = 250{,}5^{o}C.$$

With $t_{cp} = 250{,}5^{o}C$ we determine the thermo physical parameters of water according to (5 Table 1):

$\mu_{ж} = 1133{,}4 \cdot 10^{-7}\, \frac{кг}{м \cdot с} -$ Specific dynamic viscosity of water;

$\Pr_{ж} = 0{,}79711 -$ Prandtl number;

$\lambda_{ж} = 650{,}64 \cdot 10^{-3}\, \frac{Вт}{м \cdot {}^{0}C} -$ coefficient of thermal conductivity of water;

$\vartheta_{ж} = 0{,}0012083 \frac{м^{3}}{кг} -$ Specific volume of water;

$\nu_{ж} = 1{,}3695 \cdot 10^{-7}\, \frac{м^{2}}{с} -$ Coefficient of kinematic viscosity of water.

We determine the regime of water flow:

$$\mathrm{Re}_{ж} = \frac{w_{в} \cdot d_{вн}}{\nu_{ж}} = \frac{1{,}3 \cdot 0{,}02}{1{,}3695 \cdot 10^{-7}} = 1{,}899 \cdot 10^{5}.$$

With $\mathrm{Re}_{ж} > 10^{4}$ coefficient of heat transfer from the wall to the water:

$$a_{2} = 0{,}023 \cdot \mathrm{Re}_{ж}^{0{,}8} \cdot \Pr_{ж}^{0{,}43} \cdot \frac{\lambda_{ж}}{d_{вн}} \cdot \varepsilon_{cn} =$$

$$= 0{,}023 \cdot \left(1{,}899 \cdot 10^{5}\right)^{0{,}8} \cdot 0{,}79711^{0{,}43} \cdot \frac{0{,}65064}{0{,}02} \cdot 1{,}35 = 15305{,}42 \frac{Вт}{м^{2} \cdot К};$$

7.1.5.2. Calculation of the heat transfer coefficient from the heating medium to the wall of the pipes

We set the heat transfer coefficient:

$$K_{ок} = 3386\frac{Вт}{м^2 \cdot К}.$$

Specific heat flow in the actual heater:

$$q = K_{ок} \cdot \Delta t_{ср.л} = 3386 \cdot 12,9 = 43679,4\frac{Вт}{м^2}.$$

Outside tube wall temperature:

$$t_{ст}^{нар} = t'_{ок} + \frac{q}{R_{тр}} + \frac{q}{a_2} = 255,38 + \frac{43679,4}{13000} + \frac{43679,4}{15305,42} = 261,5^o C;$$

where: $R_{тр} = \frac{\lambda_{тр}}{\delta c_{ст}} = \frac{52}{0,004} = 13000 -$ thermal resistance of the coil tube;

$\lambda_{тр} = 52\frac{Вт}{м \cdot К} -$ Coefficient of thermal conductivity of metal pipes;

$\delta_{ст} = 0,004 м -$ Thickness of pipe walls.

Condensate film temperature:

$$t_{пл}^{к} = \frac{t_s + t_{ст}^{нар}}{2} = \frac{271,5 + 261,5}{2} = 266,5^o C.$$

With $t_{пл}^{к} = 266,5^o C$ determine the thermophysical parameters of the condensate according to (5. Table 1):

$\mu_к = 990,21 \cdot 10^{-7}\frac{кг}{м \cdot с} -$ Specific dynamic viscosity of water;

$\Pr_к = 0,83865 -$ Prandtl number;

$\lambda_к = 597,54 \cdot 10^{-3}\frac{Вт}{м \cdot {}^0 C} -$ Coefficient of water thermal conductivity;

$\vartheta_к = 0,0012926\frac{м^3}{кг} -$ Specific volume of water;

$\nu_к = 1,2799 \cdot 10^{-7}\frac{м^2}{с} -$ Coefficient of water kinematic viscosity.

We determine the mode of washing the coil tubes with the condensate steam:

$$\mathrm{Re}_{ж} = \frac{q \cdot H_{акт}}{\mu_{к} \cdot r} = \frac{43999{,}32 \cdot 0{,}224}{990{,}21 \cdot 10^{-7} \cdot 1578{,}4 \cdot 10^{3}} = 63{,}06;$$

where: $H_{акт} = m \cdot d_{нар} = 8 \cdot 0{,}028 = 0{,}224 м -$ active pipe height;

$r = 1578{,}9 \cdot 10^{3} \frac{Дж}{кг} -$ Heat of vaporization.

Coefficient of heat transfer from the heating medium to the walls of pipes:

$$a_1 = (\varepsilon_r \cdot C)^{\frac{3}{4}} \cdot \lambda_{к} \cdot \left(\frac{g}{(\nu_{к})^2}\right)^{\frac{1}{3}} \cdot (\mathrm{Re}_{к})^{-\frac{1}{3}} =$$

$$= (0{,}8 \cdot 0{,}728)^{\frac{4}{3}} \cdot 597{,}54 \cdot 10^{-3} \cdot \left(\frac{9{,}81}{(1{,}2799 \cdot 10^{-7})^2}\right)^{\frac{1}{3}} \cdot 63{,}06^{-\frac{1}{3}} = 6154{,}4 \frac{Вт}{м^2 \cdot К};$$

where: $\varepsilon_r = 0{,}8 -$ correction for the type of pipe (pipe made of carbon steel);

$C = 0{,}728 -$ Correction for the position of the pipe (spiral).

7.1.5.3. Determination of the heat transfer coefficient

$$K'_{ок} = \left(d_{ср} \cdot \left(\frac{1}{a_1 \cdot d_{н}} + \frac{1}{2 \cdot \lambda_{тр}} \cdot \ln \frac{d_{н}}{d_{вн}} + \frac{1}{a_2 \cdot d_{вн}}\right)\right)^{-1} =$$

$$= \left(0{,}024 \cdot \left(\frac{1}{6154{,}4 \cdot 0{,}028} + \frac{1}{2 \cdot 52} \cdot \ln \frac{28}{20} + \frac{1}{15305{,}42 \cdot 0{,}02}\right)\right)^{-1} = 3386{,}11 \frac{Вт}{м^2 \cdot К}.$$

Computational error:

$$\delta = \frac{K'_{ок} - K_{ок}}{K'_{ок}} \cdot 100\ \% = \frac{3386{,}11 - 3386}{3386{,}11} \cdot 100\ \% = 0{,}0033\ \%.$$

Area of heat exchange of the condensate cooler:

$$F_{ок} = \frac{Q_{ок}}{K'_{ок} \cdot \Delta t_{ср.л}} = \frac{2255,2}{3,38611 \cdot 12,9} = 51,63 м^2.$$

Number of coils in condensate cooler:

$$N_{сп}^{ок} = \frac{G_{пв}^{ок} \cdot \vartheta_{в}}{0,785 \cdot d_{вн}^2 \cdot w_{в}} = \frac{32,7 \cdot 0,0012926}{0,785 \cdot (0,02)^2 \cdot 1,3} = 103,55 \approx 104 шт;$$

The length of each coil in the condensate cooler:

$$I_{сп} = \frac{F_{сп}}{\pi \cdot d_{ср} \cdot N_{спир}} = \frac{51,63}{3,14 \cdot 0,024 \cdot 104} = 6,59 м.$$

7.2. Determination of the geometric dimensions of the heater

Number of collectors:

$$N_{кол} = 4 \;\; шт.$$

The speed of water in the reservoir is:

$$w_{в} = 4 \;\; {}^{м}\!/_{с}.$$

Feedwater consumption:

$$G_{пв}^{1п} = 225,51 \, {}^{кг}\!/_{с}.$$

Internal diameter of the collector:

$$d_{вн} = \sqrt{\frac{4 \cdot G_{пв}^{1п} \cdot \vartheta_{в} \cdot 2}{\pi \cdot N_{кол} \cdot w_{в}}} = \sqrt{\frac{4 \cdot 225,51 \cdot 0,00127931 \;\; \cdot 2}{3,14 \cdot 4 \cdot 4}} = 0,214 м;$$

where: $\vartheta_{в} = 0,00127931 \;\; {}^{м^3}\!/_{кг}$ – specific volume of feedwater (5. table 1).

We select the standard pipe:

outside diameter $d_{нар} = 0,273 \;\; м;$

Collector wall thickness $s_{кол} = 0,012 \;\; м;$

inner diameter $d_{вн} = 0{,}250\ м.$

Determine the number of sections:

$$N_{сек} = \frac{N}{N_{кол} \cdot n_{тр}} = \frac{576}{4 \cdot 12} = 12\ \ шт\ ;$$

wher: $n_{тр} = 12\ \ шт\ -$ the number of tubes in the section.

Determine the number of partitions in the heater itself:

$$N_{пер} = \frac{N_{сек}}{N_{кол}} + 1 = \frac{12}{4} + 1 = 4\ \ шт\ .$$

Determine the height of the collector in the zone of the heater itself:

$$H_{кол} = \left(\frac{2 \cdot N}{N_{кол}} - 1 \right) \cdot t + \delta_{пер} \cdot N_{пер} = \left(\frac{2 \cdot 576}{4} - 1 \right) \cdot 0{,}032 + 0{,}004\ \ \cdot 4 = 9{,}20 м;$$

where: $\delta_{пер} = 0{,}004\ \ м\ -$ thickness of the partition.

Number of sections in the condensate cooler:

$$N_{сек} = 1\ \ шт\ .$$

Number of partitions in the condensate cooler:

$$N_{пер} = 1\ \ шт\ .$$

We determine the height of the reservoir in the zone of the condensate cooler:

$$H_{кол} = \left(\frac{2 \cdot N}{N_{кол}} - 1 \right) \cdot t + \delta_{пер} \cdot N_{пер} = \left(\frac{2 \cdot 104}{4} - 1 \right) \cdot 0{,}032 + 0{,}004 \cdot 1 = 1{,}64 м;$$

where: $\delta_{пер} = 0{,}004\ \ м\ -$ thickness of the partition.

Number of sections in the steam cooler:

$$N_{сек} = 1 \ шт \ .$$

Number of partitions in the steam cooler:

$$N_{пер} = 1 \ шт \ .$$

We determine the height of the collector in the zone of the steam cooler:

$$H_{кол} = \left(\frac{2 \cdot N}{N_{кол}} - 1 \right) \cdot t + \delta_{пер} \cdot N_{пер} = \left(\frac{2 \cdot 69}{4} - 1 \right) \cdot 0{,}032 + 0{,}004 \cdot 2 = 1{,}076 м;$$

where: $\delta_{пер} = 0{,}004 \ м$ − thickness of the partition.

Internal smallest diameter of turns of a spiral:

$$D_{сп} = 0{,}06 \ м.$$

Outer diameter of the coil turns:

$$D_{н} = D_{вн} + 2 \cdot N_{вит} \cdot t + \delta_{сп} = 0{,}06 + 2 \cdot 8 \cdot 0{,}032 + 0{,}004 = 0{,}576 \ м.$$

Calculation of the internal diameter of the heater housing:

$$D_{вн}^{к} = 3 \cdot D_{н} + 2 \cdot \delta_1' + 2{,}6 \cdot \delta_2 = 3 \cdot 0{,}576 + 2 \cdot 0{,}08 + 2{,}6 \cdot 0{,}004 = 1{,}8984 \ м \ ;$$

$$D_{вн}^{к} = 2 \cdot \left(D_{н} + 2 \cdot \delta_3 \right) + 3 \cdot d_{кол} + 2 \cdot \delta_1 = 2 \cdot \left(0{,}576 + 2 \cdot 0{,}02 \right) + 3 \cdot 0{,}219 + 2 \cdot 0{,}02 = 1{,}929 \ м \ .$$

From the calculated internal diameters, we choose the largest:

$$D_{вн}^{к} = 1{,}929 \ м \ .$$

Calculation of feed water pipes:

$$D_{вн}^{р} = \sqrt{\frac{4 \cdot G_{пв}^{1п} \cdot \vartheta_{в}}{\pi \cdot w_{в}}} = \sqrt{\frac{4 \cdot 225{,}51 \cdot 0{,}00127931}{3{,}14 \cdot 4}} = 0{,}303 \ м \ .$$

By the closest conditional diameter, we select the standard pipe and specify the speed of the water:

outside diameter $d_{нар} = 0{,}325 \ м;$

Collector wall thickness $s_{кол} = 0{,}019$ $м$;

inner diameter $d_{вн} = 0{,}282$ $м$;

$$w_в = \frac{4 \cdot G_{пв}^{1п} \cdot \vartheta}{\pi \cdot D_{вн}^2} = \frac{4 \cdot 225{,}51 \cdot 0{,}00127931}{3{,}14 \cdot 0{,}284^2} = 4{,}001 \frac{м}{с}.$$

Calculation of the heating steam pipe:

$$D_{вн}^р = \sqrt{\frac{4 \cdot G_п^{1п} \cdot \vartheta_в}{\pi \cdot w_в}} = \sqrt{\frac{4 \cdot 17{,}54 \cdot 0{,}04712}{3{,}14 \cdot 40}} = 0{,}162 \ м \ .$$

On the nearest conditional diameter we select the standard pipe and specify the rate of condensate (drainage):

outside diameter $d_{нар} = 0{,}219$ $м$;

Collector wall thickness $s_{кол} = 0{,}019$ $м$;

inner diameter $d_{вн} = 0{,}2$ $м$;

$$w_п = \frac{4 \cdot G_п^{1п} \cdot \vartheta}{\pi \cdot D_{вн}^2} = \frac{4 \cdot 17{,}54 \cdot 0{,}04712}{3{,}14 \cdot 0{,}2^2} = 26{,}31 \frac{м}{с}.$$

Pipe calculation for the drainage of the heating steam:

$$D_{вн}^р = \sqrt{\frac{4 \cdot G_{др}^{1п} \cdot \vartheta_в}{\pi \cdot w_{др}}} = \sqrt{\frac{4 \cdot 17{,}54 \cdot 0{,}0127931}{3{,}14 \cdot 1}} = 0{,}169 \ м \ .$$

where: $w_{др} = 1 м/с -$ the recommended condensate speed of the heating steam.

On the nearest conditional diameter $D_{вн}^у$ we select the standard pipe and specify the rate of condensate (drainage):

outside diameter $d_{нар} = 0{,}219$ $м$;

Collector wall thickness $s_{кол} = 0{,}019$ $м$;

inner diameter $d_{вн} = 0{,}2\ \ м$;

$$w_в = \frac{4 \cdot G_n^{1n} \cdot \vartheta}{\pi \cdot D_{вн}^2} = \frac{4 \cdot 17{,}54 \cdot 0{,}00127931}{3{,}14 \cdot 0{,}2^2} = 0{,}714\,\frac{м}{c}.$$

7.3. Mechanical calculation of the heater housing

Inner diameter of the shell:

$$D_{вн} = 2000\ \ мм.$$

Design temperature:

$$t_{расч} = 500^o C.$$

Allowable voltage for steel 20K:

$$\sigma_{доп} = 11{,}9\,\frac{кгс}{мм^2}.$$

Coefficient of strength of welded seam:

$\varphi = 1 -$ During automatic welding.

Addition to the calculated wall thickness: $C = 1$.

Estimated shell thickness:

:

$$S_0 = \frac{P \cdot D_{вн}}{200 \cdot \varphi \cdot \sigma_{доп} - P} + C = \frac{57{,}41 \cdot 2000}{200 \cdot 1 \cdot 11{,}9 - 57{,}41} + 1 = 49{,}44мм\ ;$$

where: $P = 56{,}3\ \ бар = 57{,}41\frac{кгс}{см^2}$ $-$ calculated steam pressure in the heater housing.

We assume the wall thickness of the shell to be equal to: $S_0 = 65мм.$

Estimated wall thickness of the elliptical bottom:

$$S_0 = \frac{P \cdot}{400 \cdot z \cdot \sigma_{доп} - P} \cdot \frac{D_{вн}}{2 \cdot h_в} + C = \frac{69,58 \cdot 2000}{400 \cdot 1,05 \cdot 11,9 - 69,58} \cdot \frac{2000}{2 \cdot 550} + 1 = 52,3 мм;$$

where: $z = 1,05 -$ coefficient of weakening the bottom by the hole;

$h_в = 550 мм -$ Height of the convex part of the bottom (internal size).

We take the thickness of the bottom wall $S_0 = 65 мм.$

7.4. Hydraulic calculation of the heater

7.4.1. Calculation of the hydraulic resistance of the heater itself

The coefficient of local resistance, taking into account the input of the stream into the spiral and the outlet of the stream from the spiral:

$$\xi_м = N_{сек} \cdot \xi_{вход} + N_{сек} \cdot \xi_{вых} + \xi_{кр} = 12 \cdot 1,25 + 12 \cdot 1 + 0,5 = 27,5;$$

where: $\xi_{кр} = 0,5$ $-$ coefficient of local resistance, taking into account the influence of the curvature of the spiral (3. page 35).

Coefficient that takes friction resistance into account:

$$\lambda = 0,1 \cdot \left(1,46 \cdot \frac{\Delta}{d_э} + \frac{100}{R_e}\right)^{0,25} = 0,1 \cdot \left(1,46 \cdot \frac{0,0002}{0,024} + \frac{100}{1,9863 \cdot 10^5}\right)^{0,25} = 0,034;$$

where: $\Delta = 0,0002 м -$ roughness of the pipe walls of the coil (3. page 35).

Hydraulic resistance in spiral coils:

$$\Delta P_{сп}^{спир} = \left(\lambda \cdot \frac{I_{сп}}{d_э} + \xi_м\right) \cdot \frac{\rho_в \cdot w_в^2}{2} =$$

$$= \left(0{,}034 \cdot \frac{13{,}32}{0{,}024} + 2{,}75\right) \cdot \frac{780{,}3 \cdot 1{,}3^2}{2} = 14255 Па = 0{,}143 бар;$$

where: $\rho_в = 780{,}3 \, кг/м^3 -$ density of feed water (Table 5);

$w_в = 1{,}3 \, м/с -$ Feed water velocity in the pipes of the spiral coil.

Hydraulic resistance in collector pipes:

$$\Delta P_{сп}^{кол} = \left(\lambda \cdot \frac{H_{кол}}{d_э}\right) \cdot \frac{\rho_в \cdot w_в^2}{2} = \left(0{,}034 \cdot \frac{9{,}20}{0{,}21}\right) \cdot \frac{780{,}3 \cdot 4^2}{2} = 17948 Па = 0{,}18 бар;$$

where: $H_{кол} = 9{,}2 м -$ height of the collector in the zone of the heater itself;

$w_в = 4 \, м/с -$ Feedwater velocity in the pipes of the collector.

Hydraulic resistance of the heater zone:

$$\Delta P_{сп} = \Delta P_{сп}^{спир} + \Delta P_{сп}^{кол} = 0{,}143 + 0{,}18 = 0{,}323 бар.$$

7.4.2. Calculation of the hydraulic resistance of the steam cooler

The coefficient of local resistance, taking into account the input of the stream into the spiral and the outlet of the stream from the spiral:

$$\xi_м = N_{сек} \cdot \xi_{вход} + N_{сек} \cdot \xi_{вых} + \xi_{кр} = 1{,}25 + 1 + 0{,}5 = 2{,}75;$$

where: $\xi_{кр} = 0{,}5$ $-$ coefficient of local resistance, taking into account the influence of the curvature of the spiral (3. page 35).

Coefficient that takes friction resistance into account:

$$\lambda = 0{,}1 \cdot \left(1{,}46 \cdot \frac{\Delta}{d_э} + \frac{100}{R_e}\right)^{0{,}25} = 0{,}1 \cdot \left(1{,}46 \cdot \frac{0{,}0002}{0{,}024} + \frac{100}{1{,}9863 \cdot 10^5}\right)^{0{,}25} = 0{,}034;$$

where: $\Delta = 0{,}0002\textit{м} -$ roughness of the pipe walls of the coil (3. page 35).

Hydraulic resistance in spiral coils:

$$\Delta P_{on}^{спир} = \left(\lambda \cdot \frac{I_{сп}}{d_э} + \xi_м \right) \cdot \frac{\rho_в \cdot w_в^2}{2} =$$

$$= \left(0{,}034 \cdot \frac{13{,}2}{0{,}024} + 2{,}75 \right) \cdot \frac{780{,}3 \cdot 1{,}3^2}{2} = 14143, \textit{Па} = 0{,}142\textit{бар};$$

where: $\rho_в = 780{,}3 \, {}^{кг}\!/\!_{м^3} -$ density of feed water (Table 5);

$w_в = 1{,}3 \, {}^{м}\!/\!_{с} -$ Feed water velocity in the pipes of the spiral coil.

Hydraulic resistance in collector pipes:

$$\Delta P_{on}^{кол} = \left(\lambda \cdot \frac{H_{кол}}{d_э} \right) \cdot \frac{\rho_в \cdot w_в^2}{2} = \left(0{,}034 \cdot \frac{1{,}076}{0{,}21} \right) \cdot \frac{780{,}3 \cdot 4^2}{2} = 913{,}5\textit{Па} = 0{,}009\textit{бар};$$

where: $H_{кол} = 0{,}79\textit{м} -$ height of the collector in the zone of the heater itself;

$w_в = 4 \, {}^{м}\!/\!_{с} -$ Feedwater velocity in the pipes of the collector.

Hydraulic resistance of the heater zone:

$$\Delta P_{on} = \Delta P_{on}^{спир} + \Delta P_{on}^{кол} = 0{,}142 + 0{,}009 = 0{,}151\textit{бар}.$$

7.4.3. Calculation of the hydraulic resistance of the condensate cooler

The coefficient of local resistance, taking into account the input of the stream into the spiral and the outlet of the stream from the spiral:

$$\xi_м = N_{сек} \cdot \xi_{вход} + N_{сек} \cdot \xi_{вых} + \xi_{кр} = 1{,}25 + 1 + 0{,}5 = 2{,}75;$$

where: $\xi_{кр} = 0{,}5$ $-$ coefficient of local resistance, taking into account the influence of the curvature of the spiral (3. page 35).

Coefficient that takes friction resistance into account:

$$\lambda = 0{,}1 \cdot \left(1{,}46 \cdot \frac{\Delta}{d_э} + \frac{100}{R_e}\right)^{0,25} = 0{,}1 \cdot \left(1{,}46 \cdot \frac{0{,}0002}{0{,}024} + \frac{100}{1{,}9863 \cdot 10^5}\right)^{0,25} = 0{,}034;$$

where: $\Delta = 0{,}0002 м -$ roughness of the pipe walls of the coil (3. page 35).

Hydraulic resistance in spiral coils:

$$\Delta P_{ок}^{спир} = \left(\lambda \cdot \frac{I_{сп}}{d_э} + \xi_м\right) \cdot \frac{\rho_в \cdot w_в^2}{2} =$$

$$= \left(0{,}034 \cdot \frac{6{,}59}{0{,}024} + 2{,}75\right) \cdot \frac{780{,}3 \cdot 1{,}3^2}{2} = 7968{,}8 Па = 0{,}08 бар;$$

where: $\rho_в = 780{,}3 \, кг/м^3 -$ density of feed water (Table 5);

$w_в = 1{,}3 \, м/с -$ Feed water velocity in the pipes of the spiral coil.

Hydraulic resistance in collector pipes:

$$\Delta P_{ок}^{кол} = \left(\lambda \cdot \frac{H_{кол}}{d_э}\right) \cdot \frac{\rho_в \cdot w_в^2}{2} = \left(0{,}034 \cdot \frac{1{,}64}{0{,}21}\right) \cdot \frac{780{,}3 \cdot 4^2}{2} = 1657{,}5 Па = 0{,}017 бар;$$

where: $H_{кол} = 1{,}02 м -$ height of the collector in the zone of the heater itself;

$w_в = 4 \, м/с -$ Feedwater velocity in the pipes of the collector.

Hydraulic resistance of the heater zone:

$$\Delta P_{ок} = \Delta P_{ок}^{спир} + \Delta P_{ок}^{кол} = 0{,}08 + 0{,}017 = 0{,}097 бар.$$

Hydraulic resistance of the whole heater on water:

$$\Delta P_{пв} = \Delta P_{сп} + \Delta P_{оп} + \Delta P_{ок} = 0{,}323 + 0{,}151 + 0{,}097 = 0{,}571 бар.$$

The loss of pressure in the annular space of the heater and condenser cooler and steam during steam condensation is negligible, and they can be neglected.

On the basis of the above calculations, we select the prototype PVD1: PV-1600-380-66.

8. Calculation of the steam selection pipeline of additional HPH

The pipeline of the selected steam for the fourth HPH is calculated for strength, the compensating capacity of the pipeline is estimated and the thickness of the thermal insulation layer is calculated.

8.1. Calculation of pipeline strength

Internal diameter of the pipeline:

$$d_{вн} = \sqrt{\frac{4 \cdot G_n \cdot V}{\pi \cdot C}} = \sqrt{\frac{4 \cdot 35{,}07 \cdot 0{,}04712}{3{,}14 \cdot 25{,}62}} = 0{,}287 м = 287 мм.$$

where: $V = 0{,}04712 \, м^3/кг -$ specific volume of steam (5. Table 3);

$C = 25{,}62 \, м/с -$ The optimum speed of superheated steam (6. Table 13.4).

The minimum thickness of the pipeline wall (6. F. 13.5).

$$s = \frac{P}{2 \cdot \varphi \cdot [\delta] + P} \cdot d_{вн} + C = \frac{5{,}91}{2 \cdot 0{,}8 \cdot 122 + 5{,}91} \cdot 0{,}287 + 0{,}00085 = 0{,}0093 мм;$$

where: $P = 5{,}91 МПа -$ medium pressure;

$\varphi = 0{,}8 -$ safety limit factor (6. page 200);

$[\delta] = 122 МПа -$ Allowable stresses for steel 12X1MΦ, at a temperature of 480 ° C, (16. Appendix 4);

$C = 0{,}1 \cdot s -$ Correction to the calculated wall thickness (6. page 200).

By the internal diameter, wall thickness and nominal pressure, we select from the range of high pressure pipelines a pipeline with standard dimensions: 325 x 12, where: 325 mm - the outer diameter of the pipeline; 12 mm - the wall thickness of the pipeline.

8.2. Calculation of the pipeline for self-compensation

The tracing of the pipeline of the selected steam of the fourth HPH is shown in Figure 8.1. Calculation of a simple pipeline (without branches) with two fixed supports for self-compensation is performed using two dimensionless parameters.

The first of them - geometric - represents the ratio of the elastic length of the pipeline to the distance between its fixed supports, i.e.

$$X = \frac{L}{a} - 1 = \frac{30{,}9}{18{,}3} - 1 = 0{,}714;$$

where: $L = 30{,}9 м -$ length of the pipeline;;

$a = \sqrt{X_в^2 + Y_в^2 + Z_в^2} = \sqrt{11{,}5^2 + 11{,}2^2 + 8{,}2^2} = 18{,}3 м -$ Distance between supports A and B (figure 8.1).

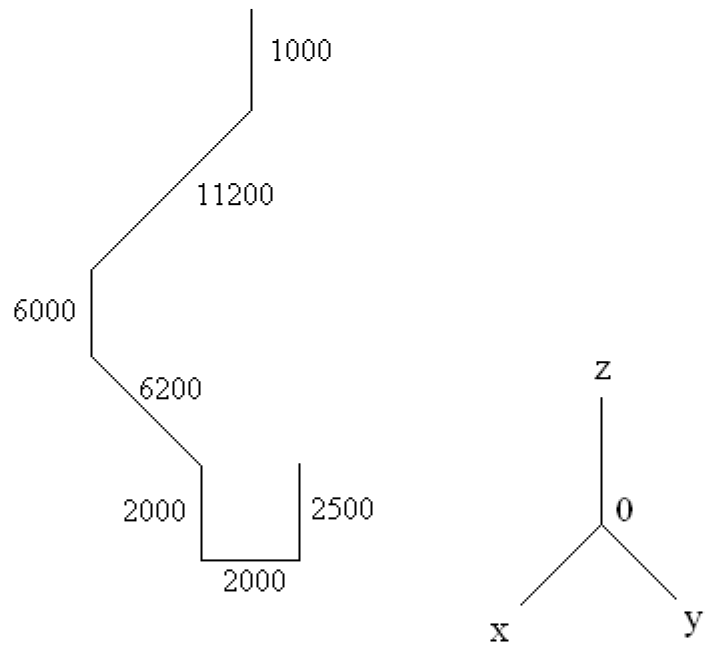


Figure 8.1. Pipeline tracing of selected steam.

The second parameter is a function of the reduced temperature deformation, referred to the unit of the extended length of the pipeline:

$$Y = n \cdot \frac{d_{н}}{L} = 9{,}4 \cdot \frac{0{,}325}{30{,}9} = 0{,}0989;$$

where: $n = 9{,}4 -$ the reduced deformation temperature for 12Kh1MF steel at 480 ° C (14. Figure 5.7).

After calculation of parameters X and Y using the graph (14. Figure 5.4), the area where the point with X and Y coordinates is located, is determined. The point with such coordinates fell into the region of the self-compensation zone and, therefore, the temperature elongations will be compensated by the pipeline itself.

8.3. Calculation of hydraulic losses in the pipeline

The extended length of the pipeline: $L = 30{,}9м.$

Steam pressure in turbine selection:

$$P_{отб} = 5{,}91 МПа = 60{,}2\, {кгс}/{см^2}.$$

Mass steam velocity in the steam pipeline:

$$m = 0{,}354 \cdot \frac{Q}{d_p^2} = 0{,}354 \cdot \frac{63{,}14}{0{,}287^2} = 271{,}36 \frac{кгс}{с \cdot м^2};$$

where: $Q = 63{,}14\, {т}/{ч} -$ steam consumption;

$d_p = 0{,}17815 м -$ Calculated internal diameter of the pipeline.

Hydrodynamic pressure at the beginning of the pipeline route:

$$P_д^1 = \frac{m^2 \cdot \vartheta_1}{2 \cdot g \cdot 10^4} = \frac{271{,}36^2 \cdot 0{,}051959}{2 \cdot 9{,}81 \cdot 10^4} = 0{,}02 \frac{кгс}{см^2} = 0{,}0204 бар;$$

where: $g = 9{,}81\, {м}/{с} -$ acceleration of gravity;

$\vartheta_1 = 0{,}051959\, {м^3}/{кг} -$ Specific volume of steam at the beginning of the pipeline route

Coefficient of pipeline resistance due to friction:

$$\xi_{тр} = \frac{\lambda}{d_p} \cdot L = \frac{0{,}02}{0{,}287} \cdot 30{,}9 = 3{,}47;$$

where: $\lambda = 0{,}02 -$ coefficient of friction, which depends on the relative equivalent roughness of the pipeline, the definition of which is given below.

The ratio of the internal design diameter to the absolute roughness:

$$\frac{d_p}{K_э} = \frac{0{,}287}{0{,}2 \cdot 10^{-3}} = 1435;$$

where: $K_э = 0{,}2 мм -$ absolute equivalent roughness of the pipeline.

Reynolds number:

$$R_e = \frac{m \cdot d_p}{g \cdot \mu} = \frac{271{,}36 \cdot 0{,}287}{9{,}81 \cdot 2{,}4 \cdot 10^{-6}} = 3{,}31 \cdot 10^6;$$

where: $\mu = 2{,}4 \cdot 10^{-6}\, кгс/м^2 -$ dynamic viscosity of steam.

At given values of the ratio of the internal diameter of the pipeline to the absolute roughness and the Reynolds number from (14. Fig. 9.4), the coefficient of friction $\lambda = 0{,}02$.

The total coefficient of resistance of the pipeline:

$$\xi = \xi_{mp} + \sum_1^n \xi_m = 3{,}47 + 1{,}5 = 4{,}97;$$

where: $\sum_1^n \xi_m = 6 \cdot \xi^{90^o} = 6 \cdot 0{,}25 = 1{,}5 -$ coefficient of local resistance at 6 turns by 90 ° (according to Figure 8.1 of the pipeline trace).

Steam pressure at the end of the pipeline:

$$P_2 = P_1 - P_д^1 \cdot \xi = 60{,}2 - 0{,}02 \cdot 4{,}97 = 60{,}10\, кгс/см^2 = 58{,}90\, бар.$$

Loss of pressure in the pipeline:

$$\Delta P = P_1 - P_2 = 60{,}2 - 60{,}1 = 0{,}1\, кгс/см^2 = 0{,}098\, бар.$$

8.4. Calculation of thermal insulation of the pipeline

Determination of the thickness of the insulation layer is carried out for given heat losses.

The first layer of insulation - mineral wool mats of brand 200 (limit is of 600 ° C). The second (cover) layer is asbestos cement plaster with a thickness of 15 mm.

Average design temperature:

$$t_{cp} = \frac{t_{вн} + t_{нар}}{2} = \frac{438{,}3 + 45}{2} = 241{,}65^o C;$$

where: $t_{вн} = 438{,}3^o C -$ temperature on the inside of the insulation;

$t_{нар} = 45^o C -$ Temperature on the outside of the insulation (15. 4.1).

Coefficient of thermal conductivity of the insulating layer (15. Table 2):

$$\lambda = 0{,}05 + 0{,}00016 \cdot t_{ср} = 0{,}05 + 0{,}00016 \cdot 241{,}65 = 0{,}089 \frac{ккал}{ч \cdot м^2 \cdot {}^o C} = 0{,}103 \frac{Вт}{м^2 \cdot {}^o C}.$$

Coefficient of heat transfer from the surface of insulation to the ambient air (15.f. 5):

$$a_{нар} = 8{,}1 + 0{,}045 \cdot (t_{нар} - t_{ос}) = 8{,}1 + 0{,}045 \cdot (45 - 25) = 9 \frac{ккал}{ч \cdot м^2 \cdot {}^o C} = 10{,}467 \frac{Вт}{м^2 \cdot {}^o C}.$$

where: $t_{ос} = 25^o C -$ ambient temperature (15. 4.1).

Normalized heat loss (Table 15):

$$q = q_{таб} \cdot к = 290 \cdot 1{,}06 = 307{,}59 \frac{ккал}{ч \cdot м} = 357{,}62 \frac{Вт}{м};$$

where: $q_{таб} = 290 {}^{ккал}/_{ч \cdot м} -$ thermal loss at and $d_{нар} = 325 мм$ и $t_{вн} = 438{,}3^o C$ (15. p. 277);

$k = 1{,}06 -$ Correction factor for the pipeline of selected steam..

For further calculation, we take the thickness of the insulation layer:

$$\delta = 158 мм.$$

The outer diameter of insulation is:

$$d_{из} = D_н + 2 \cdot \delta_{из} = 325 + 2 \cdot 158 = 641 мм.$$

Then:

$$\ln \frac{d_{из}}{D_н} = 2 \cdot \pi \cdot \lambda \cdot \left(\frac{t_{вн} - t_{нар}}{q} - \frac{1}{a_н \cdot \pi \cdot d_{из}} \right) =$$

$$= 2 \cdot 3{,}14 \cdot 0{,}103 \cdot \left(\frac{438{,}3 - 45}{357{,}62} - \frac{1}{10{,}467 \cdot 3{,}14 \cdot 0{,}641} \right) = 0{,}68103.$$

Hence we determine the ratio:

$$\frac{d_{из}}{D_{н}} = 1{,}9759.$$

And, consequently, the outer diameter of the insulation is:

$$d_{из} = 2{,}16831 \cdot D_{н} = 1{,}9759 \cdot 325 = 642 мм.$$

Computational error:

$$\Delta = \frac{\left| d_{из}^{прин} - d_{из}^{расч} \right|}{d_{из}^{прин}} \cdot 100\% = \frac{\left| 641 - 642 \right|}{641} \cdot 100\% = 0{,}156\%.$$

The thermal resistance of the plaster layer is neglected.

Coefficient of compaction of mineral wool mats: (15. Table 3).

Thickness of mats before compaction when installing them on a steam line:

$$\delta = \delta_{из} \cdot K_{у} \cdot \frac{D_{н} + \delta_{из}}{D_{н} + 2 \cdot \delta_{из}} = 158 \cdot 1{,}3 \cdot \frac{325 + 158}{325 + 2 \cdot 158} = 155 мм.$$

The coefficient $K_{у} \cdot \frac{D_{н} + \delta_{из}}{D_{н} + 2 \cdot \delta_{из}} = 1{,}3 \cdot \frac{325 + 155}{325 + 2 \cdot 155} = 0{,}983;$ is taken equal to one (15. p. 258).

9. Technical and economic performance of the upgraded unit

Table 9.1. The performance of the unit in different modes is given.

Таблица 9.1. показатели работы блока.

№ п.п.	**Name**	60%	60% с 4 ПВД
1.	Output power, *МВт*	482,28	483,79
2.	Power consumed by the turbo-drive of feed pumps, *МВт*	18,05	19,63
3.	Heat consumption per turbine, *МВт*	1116,36	1129,82
4.	Thermal power of the boiler, *МВт*	1153,92	1167,74
5.	Conditional fuel consumption per unit, *кг/с*	38,39	38,85
6.	Electrical efficiency of the installation	0,434	0,446
7.	Effective efficiency of the power supply unit	0,366	0,387
8.	Specific heat consumption for electricity generation, $г.у.т/_{кВт \cdot ч}$	283,41	275,53
9.	Specific consumption of conventional fuel per unit, , $г.у.т/_{кВт \cdot ч}$	336,07	317,53
10.	Fuel costs per year, thousand rubles.	20018,3	18913,676

10. Elaboration of other ways to increase profitability

In addition to improving the efficiency of the unit by installing an additional high-pressure heater, there are other ways to improve the technical and economic performance of the unit, such as:

Perfection of the turbine installation and its equipment:

- *Improved design of the condenser and the circulation system for the vacuum deepening;*
- *application of no-deaerator circuits of the condensate-feed pipeline;*
- *Improvement of the design of the turbine in order to increase its efficiency;*
- *Improved design of heat exchangers, pipelines and fittings to reduce pressure loss;*
- *Improvement of the drive and condensate pumps to reduce costs for own needs.*

Improving boiler economics:

- *Reduction of the coefficients of excess air, suctions;*
- *Reduction of the hydraulic resistance of the boiler and steam lines of fresh steam.*

11. Automatic level control in high-pressure heater

11.1 Determination of the scope of control and automation of the technological control object

At present, power industry is based on power units operating at high and supercritical steam parameters, with turbo generator unit power of a 1200 MW. To operate such units under normal operation conditions, it is necessary to monitor continuously or periodically up to 2000 process parameters. In this regard, even the full mechanization of around-the-clock energy equipment does not save personnel from tedious and monotonous work to manage the main and auxiliary installations of TPPs, and what is most important, does not guarantee their reliable economic work even with high skill of operating personnel. This led to a great development of automation in modern power industry.

According to the "Guidelines for the volume of technological measurements, signaling, automatic regulation and technological protection at thermal power plants" (16), it is necessary to install automatic level systems for high-pressure heaters in order to maintain a constant level of condensate in the heater to ensure reliable and economical operation of the equipment. Overfilling of the high-pressure heater enclosure with water to the level of the tie-in of the heating steam pipeline is unacceptable because of the danger of throwing water into the steam line.

11.2. Selection of principal schemes for monitoring and automation of technological subjects of control

The structural diagram of the automatic control system of the condensate level in the heater is shown in Fig. 11.1.

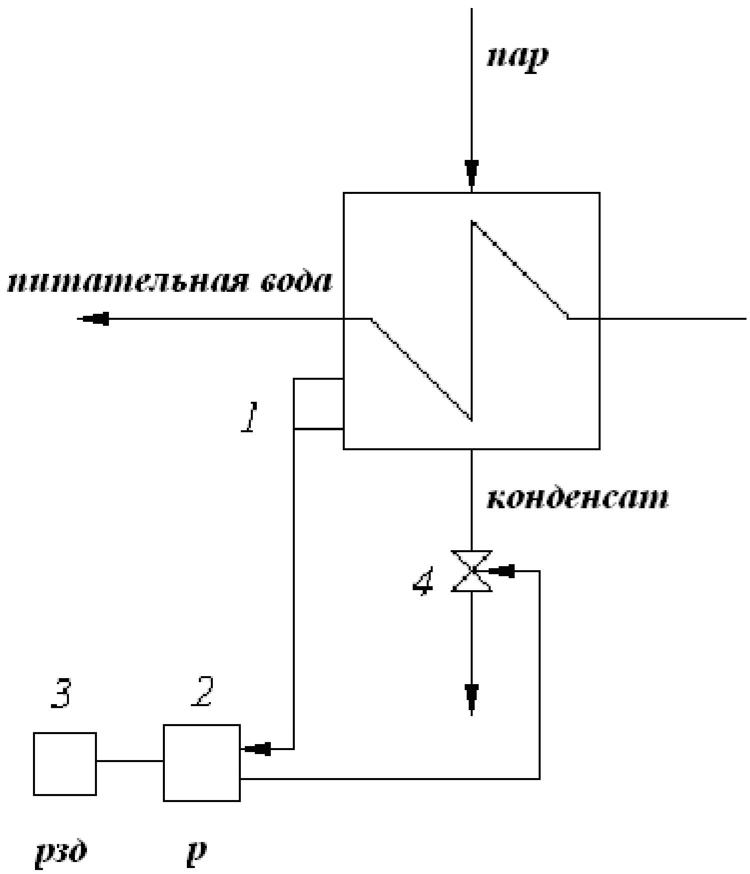


Fig. 11.1 Principle single-pulse scheme of level automatic control system

The scheme for condensate level adjusting in the high-pressure heater shown in Figure 11.1 works as follows: the level 2 regulator receives the signal from the summer housekeeper of level 1, compares it with the set signal received from the setpoint 3. In the case of different signals, the signal goes from regulator 2 to regulating Valve 4. Thus, the level control is performed by controlling the condensate flow rate. This control circuit is simple in design, reliable and economical in operation, but has its disadvantage, which is the small number of pulses arriving at the regulator 2.

We choose this variant of regulation, since it is the most optimal for the following series of reasons: 1) reliable; 2) it is economical; 3) ease of execution.

11.2.1. Review of existing control equipment and selection of equipment

Currently, all automation devices are manufactured in accordance with the the State system of industrial automation equipment. By the type of energy consumed, the regulators of indirect action are divided into three large classes: electric, pneumatic and hydraulic.

Electrical automatic regulators are used for regulation on non-explosive objects, at large distances from the control point to the control object. They do not require a special power source and allow to be installed over long distances from the control point to the control object.

In the automation of technological processes, a wide range of applications are widely used the complexes of electrical control means: aggregated complex of electrical control means, Cascade and others.

Modern electrical means of automatic control are developed on an aggregated-block principle. Blocks form an aggregated complex of technical means, with the help of which, for a particular technological object, by means of a set of specific automatic control system blocks with the required statistical and dynamic properties.

The aggregated complex of electrical control means (ACSEC) works with sensors and converters having a unified output signal. Aggregated complex of electrical control means is characterized by the use of integrated microcircuits for general and special purposes and increased functional saturation of individual units, the possibility of discrete or analog change in the parameters of dynamic adjustment of control units. This feature, in combination with the functional blocks, allows the control units to communicate with a centralized control computing system.

Experience in the operation of the above-mentioned control systems showed that the optimal auxiliary regulation equipment is the aggregated complex of electrical control means, which showed high reliability and quality of control of technological parameters, which has the general and special purpose of integrated circuits. Increased functional saturation of individual units, the possibility of a discrete or analog change in the parameters of dynamic adjustment of control units. This feature, in combination with functional blocks, makes it possible to connect the control units with a centralized control computer complex and as a consequence of the above, it can be said that the functional abilities of aggregated complex of electrical control means exceed the CASCADE control system.

11.3. The choice of technical means for the implementation of control systems and automation systems

The functional diagram of automatic control system of condensate level in a high-pressure heater is shown in Figure 11.2.

The functional scheme is made by the first method with the image of devices in accordance with ГОСТ 21.404-85. In the diagram three rectangles are taken for:

- local instruments;
- operator shield;
- operator panel.

Lines of communication between the sensors installed on the process equipment and devices and installed in place on the board and on the remote control are made with breaks.

All sets of control and automation equipment have digital designations.

Control of the actuating mechanism is carried out at the signal of the pulsed RBI regulating unit.

The signal from the level 1 sensor, which has passed the conductive separator, is fed to the input of the RBI block adder. The reference signal is generated by the manual master of the RCD and also arrives at the input of the RBI block adder, where an error signal is generated together with the feedback signal. The object can be managed either automatically or remotely. The choice of the control mode is carried out with the key of the control unit BRU-U. In automatic mode, the error signal through the key of the manual control unit BRU-U, set to position "A" (automatic), is fed to the FGP amplifier, which controls the actuator.

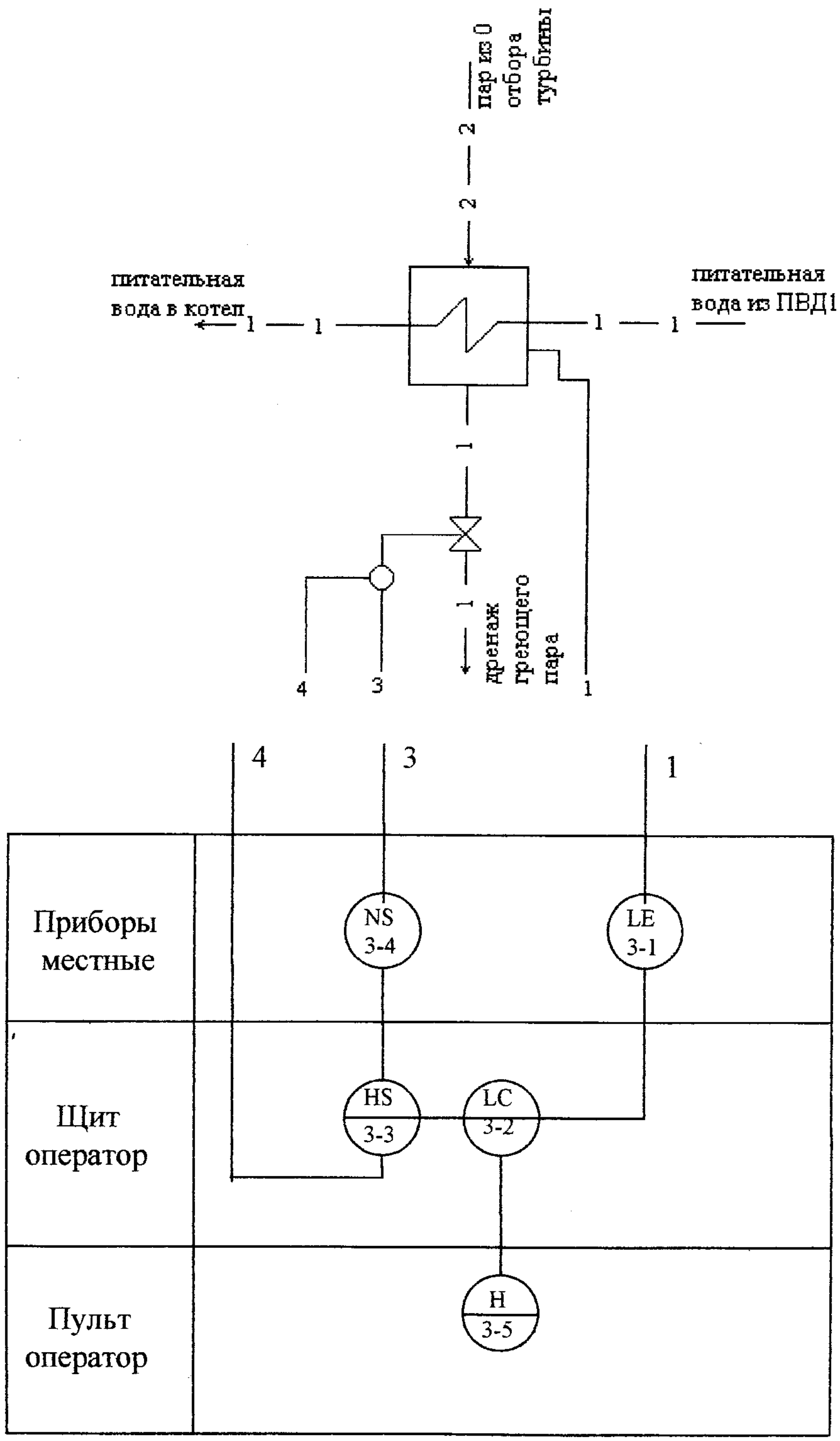


Figure 11.2 - Functional diagram of the automatic control system level.

11.3.1. Customized specification for technical means of control and automation of technological object management

A customized hardware specification is provided in the appendix.

11.3.2. Conclusion

As a result of the work done, we chose automatic condensate level control system in the heater that satisfies the technical requirements for the operation of the equipment, namely the insuring of reliable and economical operation of the automatic control system of condensate level in the high pressure heater was carried out on the devices of the aggregated complex of electrical control means system, a functional diagram and a custom specification were compiled.

12. Life safety and labor protection issues

Life safety and labor protection of employees while servicing stationary heating plants.

12.1. General safety requirements

Stationary heating plants should be operated in accordance with:

- Safety regulations for the operation of thermal mechanical equipment of power plants and heating networks;

- Rules of State Mining and Safety Organization of the Russian Federation;

- Rules for the design and safe operation of pressure vessels;

- Instruction for the operation of heat networks;

- Rules of technical operation of power plants and networks of the Russian Federation;

- SNiP 2.04.07-86 Thermal networks

In accordance with the above mentioned guidance materials, the project provides:

12.2. Premises and workplaces

12.2.1. Floors, intermediate floors should be fireproof, waterproof, smooth, non-slip and easily cleaned of dirt.

12.2.2. Drains, drainage and other channels are overlapped flush with the floor.

12.2.3. Passages and driveways, entrances and exits are illuminated, free and safe for maintenance personnel.

12.2.4. The project provides stairs, platforms, crossings and handrails. Lattices and steel sheets should be strengthened, the railing after repair must be restored.

12.3. Boiler installation

12.3.1. The boiler system must have the number of the corresponding unit, (turbine), from the selection of which receives the heating steam.

12.3.2. The project provides for safety thermal insulation of pipelines of network water, heating steam and the boiler itself. All pipelines and boilers are covered with sheet aluminum or galvanized iron and, distinctive strips are applied accordingly.

12.3.3. Shut-off valves for network water and heating steam should be located in such a way that there is the possibility of manual (non-electrical) control. According to the project, shut-off valves have an electric drive.

12.3.4. Boiler maintenance sites are equipped with expanded steel or corrugated iron.

12.3.5. Each boiler should have a table with the dates of internal inspection and hydraulic testing, which is carried out according to the schedule by the operating organization. On the pipelines "inlet" - "outlet" of the network water the arrows are put.

12.3.6. A group of boilers according to the project has two condensate pump boilers, which ensure a constant level of condensation in the boiler.

12.3.7. The boiler group is equipped in accordance with the "Norms", protections against pressure, condensate level, temperature.

12.4. Installation of network pumps

12.4.1. The mains pumps are arranged in such a way that passages are available during maintenance of the unit.

12.4.2. Each network pump has an emergency disconnection button on the installation site.

12.4.3. The flanges connecting the pipelines to the pumps must have a casing to avoid damage to the equipment by delivery water when the gaskets break through.

12.4.4. The control of the network pumps and their oil pumps must be carried out from the group (block) shield of the respective units.

12.4.5. Piping of delivery water pumps must have thermal insulation.

12.5. General safety requirements for Stationary heating plants.

12.5.1. On the installation site of boilers and network pumps under the project provides repair lighting 12V, preparation of compressed air for pneumatic tools, emergency and working lights.

12.5.2. The work of heat exchangers (boilers) should be prohibited if the dates of the next survey have expired or the defects that threaten their reliable and safe operation have been identified.

12.5.3. It is prohibited during the operation of boilers to carry out repair or work related to the elimination of a leak in the seals of the connections of individual elements under pressure.

12.5.4. The boiler must be switched off immediately when:

● the pressure increases above the permitted level;

● safety valves fail;

● holes, cracks, bulges in the welds of the enclosure are found.

12.5.5. When putting out the boiler for the repair on the shut-off grid for the water, steam and condensate posters with the text "Do not open - people work" and "Do not close - people work" on the drainage valves are hung.

A poster "Work here" is hung on the site of the work.

All the valves of the gate valves are locked on the chain.

12.5.6. For the hydraulic testing of the boiler core the project provides the location of the stand (one stand is for all boilers).

The stand must have a device for draining with hydraulics into the sewage system; to drain acid during the washing of the cores pumping for neutralization id provided.

12.5.7. When replacing the brass tubing of the boilers in the repair site rollers for turning the core are provided.

12.5.8. When removing the boiler core the opening of the boiler housing is covered with a portable wooden shield in order to avoid falling.

12.5.9. When installing or repairing all welded joints must be checked for welding quality (in order to avoid tearing, that is very important for maintenance personnel).

12.6. Service staff

12.6.2. The service personnel of the stationary heating plant are allowed to work after the passing of the technical instruction for operation, examining of the Safety Rules and passing of the exams with the certificate obtaining. Exams should be conducted annually.

12.6.3. At the workplace of the attendants there should be:

- schematic diagram of the heat supply system for network water and steam;
- job description for the personnel;
- instruction for the elimination of accidents in the heating plant.

Conclusion

As a result of the completed thesis it was revealed that the installation of an additional fourth high-pressure heater in partial mode gives:

- fuel economy of $18{,}54\,г.у.т./кВт \cdot ч$, due to stronger heating of feed water;
- increase of the unit efficiency of 2.1%.

After the calculation of the technical and economic performance of the unit, it became clear that the costs associated with the installation of an additional high-pressure heater are compensated in 4.9 months, providing the unit operation in partial mode of 60% from the nominal during all this time.

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Appendix A

Specification for equipment to functional scheme of automatic level control in HPH.

Поз.	Pos. Name, technical characteristics of instruments and means of automation, Manufacturer	Type and brand of the device	Quantity
3-1	Differential Pressure Transmitter	Sapphire 22 DD	1 pc.
3-2	Pulse regulating unit	РБИ	1 pc.
3-3	Control unit with integrated position indicator	БРУ-У	1 pc.
3-4	Contactless reversing contactor	ПБР-3М-1	1 pc.
3-5	Manual operation switch	РЗД	1 pc.
3-6	Electric single-turn actuator	МЭО-63-0.63	1 pc.