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PREFACE

Using the boiling of liquids in cooling systems is one of the most effective ways to remove heat in modern equipment and plants. The boiling process is used to cool nuclear reactors, individual elements of spacecrafts, and modern computers, as well as in refrigeration, cryogenics, chemical-engineering, and oil refining plants, heat pumps, etc. This requires using a huge variety of heat carriers and process parameters, such as pressure, temperature, liquid and vapor flow velocities, as well as the materials of cooled surface and their geometric configuration.

The boiling heat transfer process turned out to be such a complex phenomenon that despite the desperate efforts of scientists, no mathematical model has yet been created that would satisfactorily describe the results of experiments performed even for the simplest case of nucleate pool boiling of a liquid.

To date, it has been proved that boiling heat transfer is a problem with conjugate boundary conditions. The intensity of boiling heat transfer depends both on physical properties of the boiling liquid and physical properties of the material and geometric parameters of the heat-realizing surface. Dozens of monographs, subject collections, and many thousands of papers published in various journals deal with the study of boiling heat transfer. This diversity of literary sources testifies to just one thing—the boiling heat transfer process appeared to be much more complex than it was imagined at the beginning of the last century when these studies just began.

The objective that the author of the present monograph has set is to describe the fundamental achievements in the boiling heat transfer studies, carried out in recent decades by various authors. The monograph deliberately does not consider certain research areas on boiling, such as film boiling, boiling of liquid metals, low-pressure boiling, boiling heat transfer in non-stationary processes, etc.

The flow boiling heat transfer in tubes is described very briefly. Each of the boiling modes is so specific and complex that their description requires devoting a specific monograph.

Thus, in the present monograph, the main focus was made on the study of heat transfer in nucleate pool boiling under free convection conditions.

It should be noted that a significant number of repeatedly cited experimental studies were carried out with methodological errors, which the authors of publications usually did not suspect. The choice of the most reliable studies, where the results and methods of conducting experiments are presented in detail, led to the fact that the monograph contains a significant number of tables, presenting the main parameters of the conducted experiments. It is these data from various sources, as well as experiments performed at the S.S. Kutateladze Institute of Thermophysics of the Siberian Branch of the Russian Academy of Sciences involving the author of the present monograph, that served the basis for obtaining correlations given in a dimensionless form, and determining the effect of a particular parameter.

The author expresses deep gratitude to his colleagues with whom he cooperated in different years: Academician S. S. Kutateladze, PhD G. I. Bobrovich, PhD N. N. Mamontova, Professor A. R. Dorokhov, engineers A. E. Silkachev and I. N. Svorkova, PhD N. V. Valunina, engineers V. N. Bochagov, A. I. Kataev, I. B. Mironova, V. A. Gavrilov, Professor A. M. Sukhotin, PhD I. A. Semirikova, PhD V. I. Sosunov, PhD Yu. M. Petin, Academicians A. K. Rebrov and S. V. Alekseenko, engineer Yu. M. Pshenichnikov and Professor S. V. Stankus.

Valuable comments and criticism will be appreciated.

INTRODUCTION

Continuous technological advancement in various branches of engineering poses increasingly complicated problems for cooling components of various devices which generate a huge amount of heat both in stationary and non-stationary conditions. It is known that cooling the surface with boiling liquid is one of the most effective ways to reduce its temperature or maintain it at a given level. Therefore, boiling heat transfer is currently studied in all industrially advanced countries. Thus, at the Fifteenth International Heat Transfer Conference held in 2014 in Kyoto, 34 papers from different countries of the world were devoted to the study of pool boiling heat transfer. Generally, in boiling heat transfer studies, the determining parameters are the properties of a heat carrier, flow velocity, pressure, geometric and physical parameters of the cooled surface, stationary and non-stationary heat release processes, etc. In experimental studies of recent years, the most modern devices and the latest technologies are used to obtain fundamentally new information about boiling heat transfer mechanisms.

The following fundamentally important results were obtained in previous years: measurements of surface temperature fluctuations under a growing vapor bubble (Moore and Mesler 1961, 620–624; Hsu and Schmidt 1961, 254–260), the effect of physical properties of a heat-releasing surface on boiling heat transfer (Grigoriev, Pavlov and Ametistov 1977, 289; Ametistov, Grigoriev and Pavlov 19793, 908–910), the effect of tube wall thickness based on the concept of cooling depth (Grigoriev, Pavlov and Ametistov 1977, 289; Klimenko, 1975, 32), the effect of surface roughness on boiling heat transfer (Danilova and Belsky 1970, 24–28; Kurihara and Myers 1960, 23–31; Vachon, Tanger, Davis and Nix 1968, 52–61; Berenson 1962, 985–999), the formation of dry patches and liquid-wetted spots in a thin liquid film washing a surface in the pre-crisis mode (van Ouwerkerk 1972, 25–34), and recording of the local temperature pulsations of a surface in pre-crisis mode (Efferson 1969, 1995–2000; Ishigai and Kuno 1966, 361–368).

The fundamental monographs devoted to the research on boiling heat transfer (Kutateladze 1979, 415; Grigoriev, Pavlov and Ametistov 1977, 289; Galin and Kirillov 1987, 375; Dwyer 1980, 516; Prisnyakov 1988, 240; Kutepov, Sterman and Styushin 1986, 447) describe principal achievements in boiling heat transfer and present the results of original research conducted by various authors. Systematization and generalization of the results published in domestic and foreign literature allowed us to highlight the achievements in the concerned area and identify the unsolved problems. A huge and carefully thought-out array of experimental data and theoretical achievements provided for drawing an unambiguous conclusion about the significant effect of physical and geometric characteristics of a heat-releasing surface on boiling heat transfer. At present, it is quite obvious that boiling heat transfer is described by the Navier-Stokes equations for vapor and liquid phases solved together with the energy equation and the non-stationary heat conduction equation written for a heat-releasing wall, provided conjugation of these equations. Yet this problem has not been solved completely, however, it is quite obvious that it is overdue and waiting for its solution.

A special chapter of the monograph presents a review of works on methods aimed at enhancing boiling heat transfer. In vapor generators where one heat carrier has a high heat transfer coefficient, while the other —a low one, the problem of heat transfer enhancement becomes especially relevant since only by ensuring the same heat transfer coefficients on both sides of the wall separating the heat carriers one can minimize dimensions and weight of the heat exchanger. *This condition can be achieved by using finned walls, increasing their porosity, changing the wetting angle of the process fluid, etc., which complicate the already complex boiling process and open up an unlimited field of activity for researchers.*

The danger of emergencies caused by achieving critical heat flux in boiling has given rise to a huge number of experimental and theoretical works devoted to the study of specific conditions under which the nucleate boiling mode is replaced by the film boiling. However, only the hydrodynamic theory of crises developed by S. S. Kutateladze has perfectly described the obtained experimental results. The boiling crisis can be considered as a kind of limiting case in nucleate boiling.

In Chapter 5, it is shown that in general the stability criterion characterizing the occurrence of boiling crisis is not a constant value, but depends on physical properties of the liquid and dimensionless geometric parameters of a heat-releasing wall. The developed map of stability criterion variations depending on a dimensionless geometric parameter provides for generalizing a huge array of experimental data on boiling of a saturated and subcooled liquid on the outer surface of tubes with different diameters varying within the range of $5 \cdot 10^{-2} \leq \bar{D} \leq 60$ (Gogonin and Kutateladze 1977, 802–806; Bobrovich, Gogonin, Kutateladze and Moskvicheva 1962, 108–111; Bobrovich, Gogonin and Kutateladze 1964, 137–138; Kutateladze, Valukina and Gogonin 1967, 569–575; Gogonin 1970, 24–28). Thus, in the case of a thin-walled heater, the stability criterion can be reduced many times compared to that for thick-walled heaters, which is most clearly shown in the works of L. A. Bernath (1960, 95–116) or F. Tachibana et al. (Tachibana, Akiyama and Kawamura 1967, 121–130), and presented in generalized coordinates in Gogonin (2009, 1152–1159; 2010, 84–95).

It is known that boiling processes are widely used in thin-film heat exchangers. One of the advantages of a thin-film heat exchanger is multiple reductions in the amount of working fluid compared to immersion heat exchangers. The widespread use of thin-film heat exchangers in engineering is largely hindered by the lack of well-proven calculation methods for determining heat transfer coefficients in film evaporation and boiling, as well as calculating critical heat fluxes at the film boiling mode. When studying film flow, it is necessary to distinguish the laminar, wavy, and turbulent flow of a liquid film. Under the wavy flow regime ($100 \leq Re \leq 1{,}000$), heat transfer remains approximately constant and independent of the irrigation density. When irrigating an array of horizontal tubes located one under the other, it is impossible to obtain a laminar film flow due to the appearance of dry patches. Besides, the initial section of a thermal boundary layer is formed on each tube of the array, which role in the film evaporation can be either decisive or negligible. Practically, film boiling heat transfer depends neither on the irrigation density nor velocity of the concurrent vapor flow. Heat transfer in film boiling is always accompanied by the heat transfer caused by evaporation (Nusselt 1916, 541).

The peculiarities of heat transfer on a finned tube array are related to the fact that, in addition to the gravity and the viscosity forces, film flow hydrodynamics significantly depends on the surface tension forces. Therefore, the fin height becomes a characteristic linear dimension. Ultimately, this leads to multiple enhancements of heat transfer in film boiling and evaporation (Fujita and Tsutsui 1994, 175–180). Finning of the tube array ensures its uniform and complete irrigation and stabilizes the film flow.

The boiling characteristics of binary mixtures and corresponding critical heat fluxes are largely determined by mass transfer processes at the phase interface. In this case, boiling heat transfer can be reduced several times compared to that in single-component liquids, while the critical heat flux at certain concentrations of mixtures may increase.

Immersion-type vapor generators are still widely used in various industries. At that, they are subject to increased requirements when used in the cycles of binary power plants and heat pumps. From the generator, the vapor is directed through a superheater to a turbine or compressor; this means that vapor must be dry and contain a limited amount of moisture. In the last chapter, the dependencies known from the literature are presented, which allows us to estimate the limit vapor velocities at the outlet of the inter-tube space of a vapor generator. Only by maintaining the vapor velocity below the limit value one can ensure the required vapor moisture.

The monograph presents calculation dependencies, as well as it describes the algorithms for calculating heat transfer in the studied processes. The presented dependencies are proved by satisfactory coincidence with the available experimental data of different researchers.

CHAPTER 1

ANALYSIS OF TYPICAL METHODOLOGICAL ERRORS IN EXPERIMENTAL STUDIES OF POOL BOILING HEAT TRANSFER

INTRODUCTION

Kutateladze (1979, 415), Grigoriev et al. (Grigoriev, Pavlov and Ametistov 1977, 289), Galin and Kirillov (1987, 375), Dwyer (1980, 516), Prisnyakov (1988, 240), Kutepov et al. (Kutepov, Sterman and Styushin 1986, 447), and Labuntsov (2000, 386) show that at present there are no universal formulas that take into account the numerous features of boiling heat transfer observed in experiments. This is due to certain circumstances, of which the following are the most important.

A. The heat transfer process from the heated surface to the boiling liquid, which still has not been strictly described mathematically, without significant simplifications and assumptions, is very complex. This leads to the fact that even with processing data in criterial form, several defining dimensionless parameters that characterize the heat-transfer surface are excluded from consideration. As shown in Grigoriev et al. (Grigoriev, Pavlov and Ametistov 1977, 289), Prisnyakov (1988, 240), and Labuntsov (2000, 386), the problem of heat transfer at boiling must be considered as a problem with conjugate boundary conditions. The heat transfer intensity during boiling depends both on the physical properties of the boiling liquid and on the thermophysical properties of the heated surface.

B. Many publications lack a complete description of the experimental conditions. Even the most important parameters of the test section, such as the material, heated wall thickness, roughness, tube diameter, and length of the test section, are rarely given in publications completely. It is not always mentioned about the chemical purity of the heat transfer fluid and the methods of its degassing before the experiment.

C. A significant number of works were performed with methodological errors. Some of them are so obvious that they can be detected by careful study of the publication. Others remain hidden from the reader due to the brief description of the experimental conditions.

The latter two reasons lead to the fact that when comparing the data presented by different authors, one cannot be sure that the experiments were performed under the same conditions, and thus, comparing theoretically defined parameters with experimental data turns out to be meaningless.

1.1. EXPERIMENTAL FACILITIES

Since pool boiling occurs under free convection conditions, it is necessary to provide strict conditions ensuring free convection in the bulk where the test section is located. In this context, various researchers have used three types of experimental facilities to study the pool boiling heat transfer.

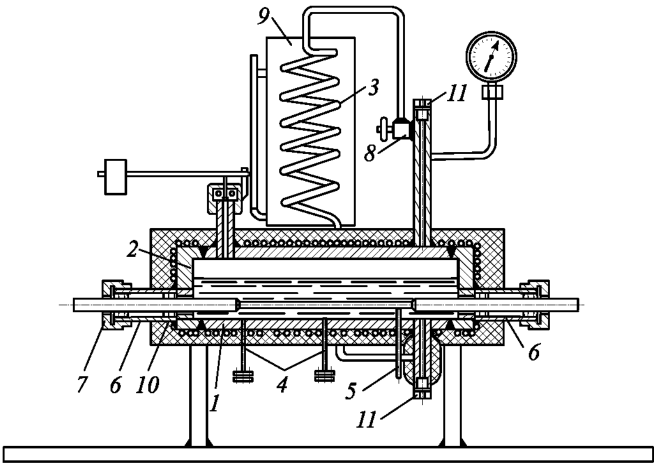

Figure 1-1 ***A*****.** Schematic diagram of the test bench (Borishansky, Kozyrev and Svetlova 1964, 71–103).
1—working vessel; *2*—flange (welded or removable); *3*—condenser coil; *4*—fitting for thermocouple insertion; *5*—thermocouple sleeve; *6*—gland; *7*—textolite washers; *8*—valve; *9*—condenser casing; *10*—thermal insulation with a guarding heater; *11*—safety valve.

The first and most common type of test bench, where boiling heat transfer is studied using liquids with a saturation temperature exceeding room temperature at atmospheric pressure, is shown in Figure 1-1 A

(Borishansky, Kozyrev and Svetlova 1964, 71–103; Mesler and Banchero 1958, 102–113; Kurihara and Myers 1960, 83–91; Borishansky, Bobrovich and Minchenko 1961, 75–93; Tolubinsky and Ostrovsky 1965, 39–46; Webb and Pais 1992, 1893–1904; Ametistov, Grigoryev and Pavlov 1973, 908–910).

When studying boiling heat transfer under free convection conditions, each experimenter designs an individual test bench, which somehow differs from other test benches. However, all the test benches available in published so far can be divided into three categories. The test benches used in Borishansky et al. (Borishansky, Kozyrev and Svetlova 1964, 71–103), Mesler and Banchero (1958, 102–113), Kurihara and Myers (1960, 83–91), Borishansky et al. (Borishansky, Bobrovich and Minchenko 1961, 75–93), Tolubinsky and Ostrovsky (1965, 39–46), Webb and Pais (1992, 1893–1904), and Ametistov et al. (Ametistov, Grigoryev and Pavlov 1973, 908–910) differ in many features from the schematic diagram shown in Figure1-1, (A), but these are secondary differences. When studying pool boiling, the main and indispensable condition is strict compliance with the requirements of free convection (large vessel, or pool), as usually stated in the title of the paper or the introduction. All the main units of the test bench are equipped with the necessary number of thermocouples which allow controlling the temperature of the liquid, steam, the test section walls, etc. All vessels are equipped with removable or welded flanges, where the test sections are installed. Optical glass windows are provided for visual observations, making photos, and high-speed shooting of the boiling process in the working vessel. The condenser is designed to maintain a given pressure at a given saturation temperature. Compensation of heat losses in the working vessel is made employing a guarding heater and external thermal insulation.

The main disadvantage of this design of the test bench is the inability to control the heat flux through the walls of the vessel with the boiling fluid. At $q_w > 0$, an excessive amount of heat that is not controlled precisely is supplied through the walls of the working vessel. At $T_w > T_s$, additional convection occurs. At $q_w < 0$, a certain energy is lost through the walls of the vessel to the environment, which results in wall temperature decrease below saturation temperature $T_w < T_s$. This condition is also accompanied by additional convection of the liquid in the working vessel. In any of these cases, the conditions of free convection are not met. At heat fluxes at the test section surface, corresponding to the transition region from convection to boiling, the heat transfer from test section wall to the liquid can be several times higher than that obtained by careful thermostatic control of the working vessel, all other conditions being equal (Figure 1-3).

Unfortunately, the vast majority of experiments on the boiling of water, alcohols, and many organic liquids with T_S>20° C, published in the literature, were performed using this type of test benches, that is, with a methodological error.

The schematic diagram of the second type of test benches is shown in Figure 1-1 B. As a rule, this type of installation is made of glass and designed to visualize the boiling process.

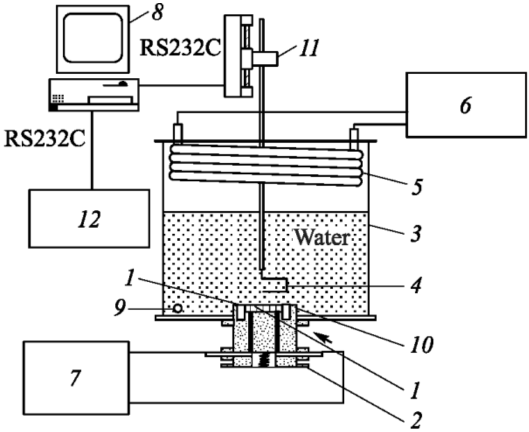

Figure 1-1 *B*. Schematic diagram of the test bench (Saito, Tanaka and Mishima 2001, 173–178). *1*—test section (liquid metal); *2*—resistive heater; *3*—working vessel; *4*—movable thermocouple; *5*—condenser; *6*—temperature control system; *7*—energy source; *8*—experimental data collecting and processing system; *9*—auxiliary heater.

This type of installation is not fundamentally different from the test bench shown in Figure 1-1 A. Very often, the upper half of the working vessel is not equipped with any guarding heaters at all and is not thermally insulated. Here, the provision of free convection conditions is not even attempted. Heat losses through the vessel wall, as well as the inclusion of an auxiliary heater located inside the boiling liquid, create additional convection on the heat-releasing surface. The heat transfer in the transition region from convection to boiling is repeatedly overestimated comparing to conditions where free convection is strictly observed. Examples of such test benches can be those described in Vachon et al. (Vachon, Tanger and

Davis 1968, 52–61), Gaertner (1965, 20–35), Adelman and Nagarajan 1970, 39–46), Saito et al. (Saito, Tanaka and Mishima 2001, 173–178).

The third type of test benches is usually used to study boiling heat transfer in cryogenic liquids or refrigerating agents operating at temperatures below room temperature. The fundamental difference of these experiments is the requirement for careful thermostatic control of the working vessel (Nishikawa, Kusuda and Yamasaki 1967, 328–338; Kirichenko, Tsybulsky and Kostromeev 1971, 271–282; Ayub and Bergles 1990, 249–255; Gogonin 1971, 349; Gorenflo 1968, 757–762). The schematic diagram of such a test bench is shown in Figure 1-1 C (Kirichenko, Tsybulsky and Kostromeev 1971, 271–282).

The test liquid in the working vessel has the same temperature as the liquid in the thermostat, in which the working vessel is placed. Heat losses through the walls of the vessel, where the test section is located are either completely absent or reduced to a minimum. Careful thermal control of the working vessel allows asserting that the free convection conditions are strictly fulfilled which means that the experimental section is the only heat source in this vessel.

In the author's opinion, only experiments performed using this type of installations can be considered reliable over the entire range of heat fluxes. The ideal situation is, where experiments on boiling heat transfer are preceded by measurements of heat transfer at natural convection. The latter data should be compared with the results of the calculation based on the classical criterial dependencies describing natural convective heat transfer (Mikheev, 1956, 392).

Measurements of boiling heat transfer at low heat fluxes can be considered reliable only if the free convection conditions are met carefully. When performing experiments with boiling water, alcohols, and other organic liquids, it is not necessary to place the experimental vessel in two or three thermostats, as is common when studying boiling of cryogenic liquids. Just one thermostat is quite enough to fulfill the requirement of zero heat flux through the walls of the experimental vessel.

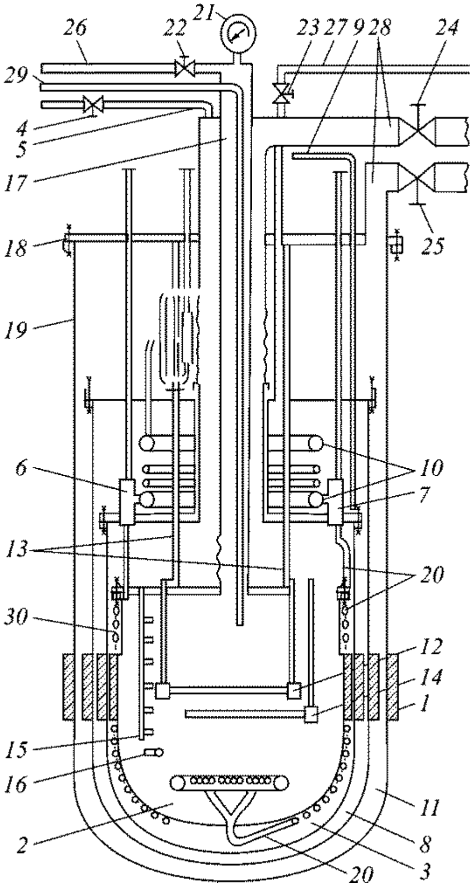

Figure 1-1 ***C.*** Schematic diagram of the test bench (Kirichenko, Tsybulsky, Kostromeev, 1971, 271–282).
1—optical windows; *2*—working vessel; *3*—the external casing of the Dewar's vessel; *4*, *6*, *7*, *19*, *22—25*—valves; *5*, *9*, *17—20*, *26—29*—pipelines; *8*—the Dewar vessel volume filled with liquid nitrogen; *10*—condenser; *11*—volume where a high vacuum is maintained; *12*, *14*—test sections; *15*—thermometers; *16*—resistive thermometer; *21*—standard pressure gage.

1.2. EXPERIMENTAL PROCEDURE TO DETERMINE THE HEAT TRANSFER COEFFICIENT AT BOILING UNDER FREE CONVECTION CONDITIONS

1.2.1. General provisions

To determine the heat transfer coefficient of a saturated liquid, it is necessary to know the following parameters: pressure (saturation temperature), specific heat flux; the wall surface temperature, geometric parameters of the test section, test section material, surface finish characteristics, the chemical purity of the liquid, and the liquid and vapor temperature. Experiments should be performed under the conditions $q_W = const$ or $T_W = const$, depending on the test section surface heating method.

The experimental vessel should be sufficiently large comparing with the test section dimensions. The following conditions must be met for cylindrical test sections: $D/D_0 \geq 10$; $L/D_0 \geq 10$, where D is the diameter of the vessel; D_0 is the diameter of the cylindrical test section; L is the length of the cylindrical test section. The vessel where the boiling occurs must be carefully insulated from the environment so that the heat fluxes through the vessel walls are zero. The working vessel should provide all the conditions of free convection heat transfer. It is advisable to start the experiments by determining the heat transfer coefficient at free convection to compare measured values with calculations made using the known criterion relations describing heat transfer at free convection. Such experiments should be considered control tests. They can confirm whether free convection conditions are carefully met, as well as are an indirect confirmation of the thorough calibration of the instruments and sensors used in the experiments.

As shown below, the vast majority of publications where boiling heat transfer was studied, were performed under gross infringement of the free convection conditions. The experimental procedure methodology in many publications is described so briefly that it is impossible to learn from the paper even all the necessary characteristics of the experimental test section (metal purity and grade, length and thickness of the test section wall, surface finish characteristics, etc.).

1.2.2. Method for determining the temperature of the heat-releasing surface

When determining the surface temperature of a thin wire (with a diameter of 10–200 microns), the latter is used as a resistive thermometer. As a rule, platinum wires are used. The temperature coefficient for each wire is determined individually by measuring the resistance of the section placed in the thermostat. The wire itself is part of a single or double bridge circuit (Kutateladze, Valukina and Gogonin 1967, 569–575). The average surface temperature of a cylindrical heater, as a rule, is measured using a differential thermocouple. One of the hot junctions is inserted into the inner cavity of the cylinder, while the other hot junction is placed in the saturated liquid vapors under strict *P(T)* dependence for a given liquid (Borishansky, Kozyrev and Svetlova 1964, 71–103; Borishansky, Bobrovich and Minchenko 1961, 75–93; Tolubinsky and Ostrovsky 1965, 39–46).

When using a thick-walled tube, the thermocouples are inserted into a special hole drilled in the tube endface, as it was done in Ayub and Bergles (1990, 249–255). In this work, the copper tube had an outer diameter of 25.4 mm and a wall thickness of δ_n = 7.5 mm. Thermocouples were inserted into 1.4 mm holes to a depth of 40 mm along the tube perimeter at an angular distance of 45 degrees. The average temperature of the tube wall was determined by averaging the readings of all thermocouples.

The heat flux and wall temperature can be determined by the well-known Fourier law. In this case, the end surface of the vertical rod serves as a heat-releasing wall. The lower end of such a rod is finned. An electric heater, installed in the cavities between fins, generates set heat flux. The side surfaces of the cylinder are carefully insulated. At least three thermocouples are mounted along the height of the rod at certain distances from the cooled endface. By measuring the temperature gradient and knowing the distance between the thermocouples, it is easy to calculate the heat flux and determine the wall temperature at the solid-liquid interface. At that, the thermal conductivity of the used material and its temperature dependence can be taken from Mesler and Banchero (1958, 102–113), Kurihara and Myers (1960, 83–91), Webb and Pais (1992, 1893–1904), Ametistov et al. (Ametistov, Grigoryev and Pavlov 1973, 908–910). In some experiments, for example in Ivanov (1965, 32) and Perkins and Westwater (1956, 471), a horizontal experimental tube was used as a resistive thermometer to determine the average wall temperature. However, this method of determining the temperature leads to a systematic error and an underestimation of the average wall temperature. The reason for the error is the different conditions under which the tube is calibrated as a resistive thermometer and used in boiling experiments. Calibration takes place in conditions where there is no temperature gradient along the tube perimeter, while temperature gradient necessarily appears when the liquid boils on the tube surface even at the high thermal conductivity of the wall material. In the latter case, the tube must be considered as consisting of several segments along its perimeter, of which each has its temperature and, therefore, certain resistance. In the measuring circuit, these resistances are connected in parallel, and as a result, their total resistance will be lower than the lowest. Besides, to increase the absolute value of the resistance, the cylinder is usually made thin-walled. As will be shown below, this also leads to a distortion of the classical q—ΔT dependence at boiling and a significant decrease in the heat transfer intensity.

1.2.3. Insufficient volume of the experimental vessel

In case of insufficient volume of the experimental vessel, boiling occurs in a limited volume or a gap between the walls of the vessel and the walls of the experimental area, rather than in a large vessel, which corresponds to pool boiling. For example, in Webb and Pais (1992, 1893–1904), the diameter of the horizontal vessel in which freons boiling was studied, was 76.2 mm, while the diameters of the experimental sections varied from 17.5 to 19.1 mm. The ratio of the diameter of the vessel to the diameter of the test section ranged from 4 to 4.3. Thus, it cannot be assumed that the experiments in Kurihara and Myers (1960, 83–91), Gaertner (1965, 20–35), and Adelman and Nagarajan (1970, 39–46) corresponded to pool boiling. In Kurihara and Myers (1960, 83–91) the diameter of the vessel was D = 203 mm, while the endface of the horizontal rod, which served the heat-releasing surface, was D_0 = 76.2 mm. In Gaertner (1965, 20–35), D = 142 mm, and D_0 = 50 mm, while in Adelman and Nagarajan (1970, 39–46), D = 228 mm, and D_0 = 152 mm. In all these works, boiling occurred in a constrained environment and thus cannot be considered as traditional pool boiling. The temperature head corresponding to the transition from convection to nucleate boiling in a limited volume can significantly differ from the temperature head at free convection, which occurs in a large volume. For this reason, the measurement results given in these publications at low heat fluxes (undeveloped boiling mode) cannot be considered reliable.

1.3. THERMOSTATIC CONTROL OF THE WORKING VESSEL

It was already noted that another reason for violation of the free convection conditions is that the wall temperature of the working vessel is not equal to the saturation temperature of the liquid. An excess amount of heat ($q_W > 0$) can be supplied through the wall of the experimental vessel (using an electric heater outside or inside the vessel). It is quite possible that the wall can be poorly insulated or not insulated at all, which results in heat loss to the environment ($q_W < 0$). In both cases this leads to additional convective flows in the working vessel. An example of such violation of free convection conditions are the results of Japanese researchers published in Saito et al. (Saito, Tanaka and Mishima 2001, 173–178). Experiments on water boiling were performed in non-thermostated vessels with significantly different volumes. Series A was performed in a vessel with glass bottom, D = 100 mm, completely covered with molten Wood alloy, while the heat-generating copper block under the molten metal had a diameter of D_0 = 80 mm. Series B was performed in a 220 × 220 mm square vessel, while the heat-generating copper block located in the center of the bottom of this vessel had a disc diameter of D_0 = 40 mm. Figure 1-2 shows the results of these experiments.

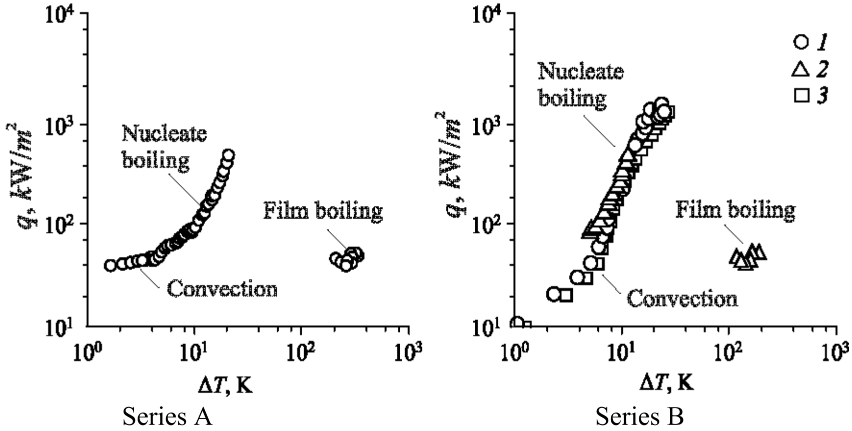

Figure 1-2. Heat transfer during boiling of water on the surface of molten metal. P = 1 bar. T_S = 100 °C (Saito, Tanaka and Mishima 2001, 173–178). Series A: D = 100 mm, D_0 = 80.0 mm. Series B: Square vessel 203 × 203 mm, D_0 = 40.0 mm.

It is clearly seen that the experimental data on developed nucleate boiling in series *A* and *B* practically do not differ from each other. However, the area of free convection and the transition from convection to nucleate boiling significantly differ in the compared experiments. Series *A* was performed in a limited space and the experimental value of the convective heat transfer coefficient is 8–14 times higher than the value calculated for heat transfer in natural convection. At boiling, there is an area of parameters where the contribution to the total heat flux by boiling and convection processes is almost commensurable $\alpha \sim q^{0.5}(q \leq 10^5 \mathrm{W/m^2})$.

When boiling in a large vessel (series *B*), there is no such clearly defined area. However, due to the heat loss of the working vessel, experimental heat transfer coefficients exceed the calculated values by up to four times during convection. The experiments have shown that the convective flow velocity in a limited volume (series *A*) was many times higher than that of the liquid in a large volume (series *B*).

The assumption of the authors of this work about the reasons for the onset of nucleate boiling on an ideally smooth surface of molten metal is, in all likelihood, not the only one. Significant heat loss through the wall of the working vessels and the resulted additional convection could fundamentally change the transition point from convection to boiling and cause convective flows in the molten metal. Convective flows in the metal form a cellular structure on its surface. Such surface cannot be considered perfectly smooth.

The author of the present monograph has conducted special studies on the effect made by heat fluxes through the wall of the working vessel on heat transfer during convection and the transition from convection to nucleate boiling. The results of these experiments are published in Gogonin (1971, 349; 2008, 413–420). The experiments of liquid boiling on horizontal cylinders with D = 2.48 and D = 3.0 mm were carried out using freon R21 and water.

The experimental results on boiling heat transfer of water are shown in Figure 1-3. The experiments on boiling were preceded by measurements of heat transfer during natural convection.

The experiments were deliberately carried out according to two methodologies, differed from each other by heating mode of the working vessel. In the first case, the working vessel was heated by a liquid in a thermostat which temperature was 10–12 °C higher than the saturation temperature of the liquid in the working vessel. In this case, individual vapor bubbles were formed on the inner wall of the working vessel. This is how the heating of the working vessel was modeled by an electric heater located outside the working vessel that is widely used in many experiments, for example, Borishansky et al. (Borishansky, Kozyrev and Svetlova 1964, 71–103), Mesler and Banchero (1958, 102–113), and Borishansky et al. (Borishansky, Bobrovich and Minchenko 1961, 75–93). The experiments in Figure 1-3 are shown by black triangles (*2*). They practically coincide with similar data obtained in Borishansky et al. (Borishansky, Bobrovich and Minchenko 1961, 75–93) at the boiling of water. At this method of heating the working vessel, the heat transfer coefficient for free convection exceeded the values calculated according to the well-known formula describing the convective heat transfer coefficient by 5–6 times, since in this case conditions for free convection were not met.

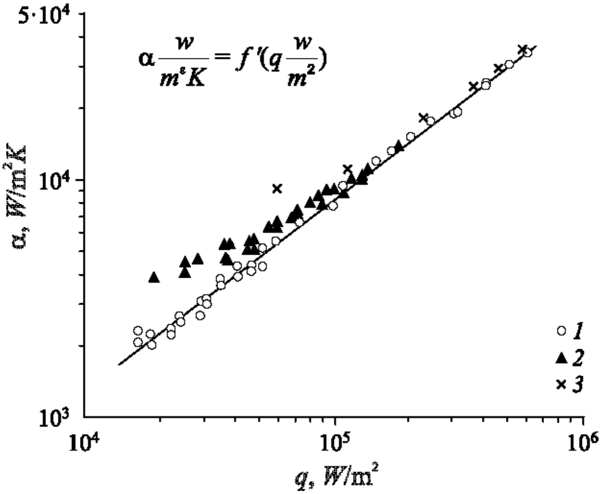

Figure 1-3. Heat transfer coefficients at water boiling (Gogonin 1971, 349). P = 1 bar; T_S = 100 °C; D_0 = 3.0 mm; δ_W = 0.5 mm; stainless steel; *1*—thermostated vessel, *2*—non-thermostated vessel, *3*—data obtained in Borishansky, Bobrovich and Minchenko (1961, 75–93).

When conducting experiments using the second method, the working vessel was placed in a thermostat which temperature corresponded to the saturation temperature of the water in the vessel. In Figure 1-3, the obtained data is indicated by light circles (*1*). In this case, the natural convection experiments corresponded to the calculations according to the corresponding criterion dependence describing the heat transfer in natural convection.

From the data shown in Figure1-3, it follows that for heat fluxes corresponding to the transition from convection to boiling, the boiling heat transfer coefficient in a non-thermostated vessel was twice the value obtained in a vessel where the free convection conditions were strictly observed.

Experiments on boiling heat transfer are usually described by dependencies written in a form of

$$\alpha \sim Aq^n \qquad (1.1).$$

Based on the experiments in a non-thermostated vessel shown in Figure 1-3, one can state that $n = 2/3$, while if strictly observing the free convection conditions, $n = 0.8$.

Visual observations and high-speed filming show that when the free convection conditions are not observed, the number of active vaporization centers at low heat fluxes increases, and boiling occurs much more intensively. The exponent in the dependence of the form (1.1) can be taken from 0.6 to 0.8 depending on the heat flux variation range, even in the experiments of the same series. It should be noted that in the experimental works by Kirichenko et al. (Kirichenko, Tsybulsky and Kostromeev 1971, 271–282), Ayub and Bergles (1990, 249–255), Gogonin (1971, 349), and Gorenflo (1968, 757–762) in which the free convection conditions in the experimental vessel were strictly observed, the exponent in correlation (1.1) equals $n = 0.75$–0.8 which follows from the obtained experimental data. Note also that in Gorenflo (1968, 757–762) N. G. Styushin's formula describing boiling heat transfer is given, and in Grigoriev et al. (Grigoriev, Pavlov and Ametistov 1977, 289) there is Yu. A. Kirichenko's formula which suggests that

$$Nu^* \sim Re_*^{0.75} \qquad (1.2).$$

1.4. THE RELATIVE LENGTH OF THE TEST SECTION

The correct determination of the boiling heat transfer coefficient assumes meeting the boundary conditions $T_W = const$ or $q_W = const$. These conditions are met only when conducting experiments on relatively long test sections where losses from the endfaces can be neglected. However, the latter condition is not always met. Figure 1-4, retrieved from Chaika (1996, 212), shows the design of the experimental test section used in experiments. The experiments were carried out using copper tubes with diameter varying from 10 to 70 mm. The table below shows some of the geometric dimensions of these tubes. Here L is the length of the tube heated section.

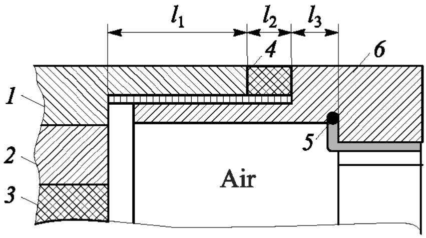

Figure 1-4. Schematic diagram of the test section in experiments in Chaika (1996, 212). *1*—heat-releasing copper cylinder, *2*—bushing, *3*—heat-generating element, *4*—solder, *5*—thermocouple.

Table 1-1. Geometric parameters of the test sections in Chaika, (1996, 212)

d, mm	L, mm	L/D_0	δ_W (Chaika 1996, 212)	δ_W (Chaika 1996, 212)	$0.5L/\delta_W$
10	60	6.0	2	2	15
70	100	1.4	8	6.0	6.5, 8.3

As follows from Figure 1-4 and the data shown in Table1-1, it is impossible to satisfy the condition q_W = *const*. One can claim that in the unheated area l_1 and at high heat fluxes—in areas l_2 and $l3$—boiling will occur. The l_1 section operates as a fin with temperature gradient and specific heat flux gradient along its length. The intense heat sink through the fin during boiling will necessarily result in distortion of the wall temperature field along the length of the short experimental cylinder. As a result, it will be impossible to correctly determine the average heat flux. The correct calculation of heat losses from the endface assumes measuring the temperature field near it along the length of the experimental section (Petukhov 1952, 343). Chaika (1996, 212) did not measure the temperature field along the length of the cylinder. When calculating heat losses, he considered the losses only from the endface presuming that heat is removed from there due to convection. The unique result on the effect of the cylinder diameter on the intensity of boiling heat transfer obtained in Chaika (1996, 212), in our opinion, is associated only with methodological errors and incorrect comparison of the data obtained under different conditions while measuring heat transfer on cylinders of different diameters. Besides, when boiling on a cylinder with D_0 = 70 mm, the experiments were performed in a limited vessel, since D/D_0 = 3.57. As shown in Kutateladze (1979, 415), boiling heat transfer does not depend