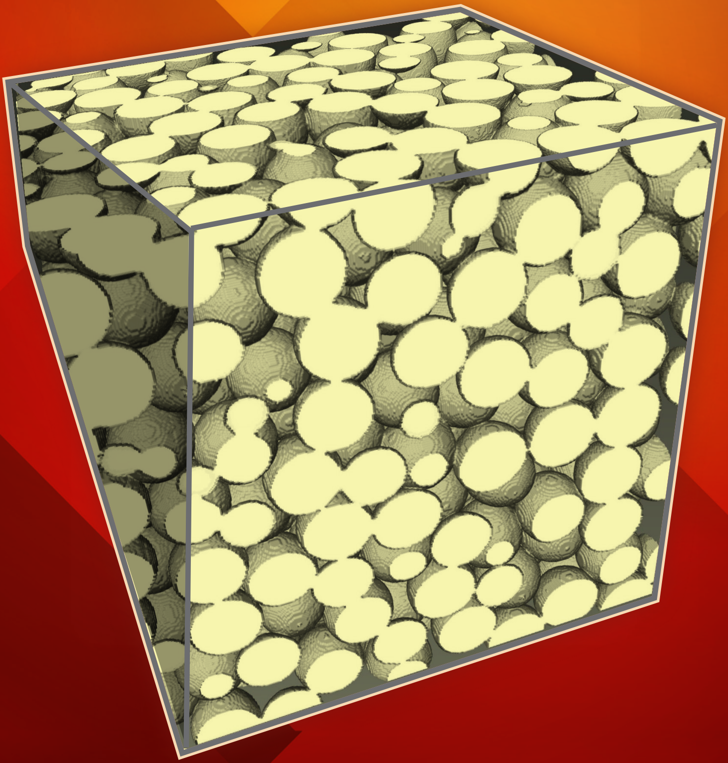
HEAT PIPES and SOLID SORPTION TRANSFORMATIONS

Fundamentals and Practical Applications

EDITED BY

L.L. Vasiliev • Sadik Kakaç



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Preface

This book covers the state of the art of adsorption research and technologies for relevant applications based on the use of the efficient heat transfer devices—heat pipes and two-phase thermosyphons—with the objectives of energy efficiency and sustainability. The severities of energy crisis and environmental problems have been calling for rapid developments in Freon-free air conditioning and heat pump technologies, the heat exchangers, which are considered as the components of prime importance. The concerns of energy consumption and environmental pollution urge researchers to work on the development of clean energy and the utilization of waste energy. From this viewpoint, interest in fuel cells and thermally activated (heat pipe heat exchangers) adsorption systems using natural refrigerants and/or alternative to hydrofluorocarbon-based refrigerants has increased significantly. The quest to accomplish a safe and comfortable environment has always been one of the main preoccupations of the sustainability of human life. Accordingly, during the past few decades, research aimed at the development of thermally powered adsorption cooling technologies has intensified. They offer two main benefits: (1) reduction in energy consumption and (2) adoption of environmentally benign adsorbent/refrigerant pairs, without compromising the desired level of comfort conditions.

The efficiency of new power sources (co-generation, tri-generation systems, fuel cells, photovoltaic systems) can be increased with the help of heat pipe heat exchangers, solid sorption heat pumps, refrigerators, accumulators of heat and cold, heat transformers, and fuel gas (natural gas and hydrogen) storage systems. Low-temperature power systems are generally significantly less expensive to build than high-temperature ones. Since a major barrier to acceptance, this is a main concern for fuel cell technology, in general. All these arguments are considered as the fundament to launch this book. Finally, the heat pipe thermal control of the spacecraft is also considered in detail in some chapters of the book.

The analysis of transport phenomena (including heating and cooling) in porous media using heat pipes and thermosyphons technology is considered in detail in Chapters 1, 5, 6, 10, and 11 of this book. Against the fact that technology of two-phase cooling systems such as heat pipes is well developed and known, some aspects of the fundamental phenomena are still not analysed in detail. The latest development in the modelling of heat and mass transfers in porous wicks in relation to the study of conventional heat pipes, loop heat pipes, and two-phase thermosyphons are is welcome. The main numerical approaches are analysed, and the advantages and drawbacks of direct methods, semi-direct methods (pore network models), and mean-field classical methods (Darcy's scale) are considered and discussed.

The global warming crisis is real and it is our responsibility to put forth an effort to reduce and prevent further damage to the global environment. One of the efficient tools to do this is heat pipe and thermosyphons application in different branches of industry. Chapter 5 contains a detail review of numerous successful heat pipe applications ranging from computer electronics to renewable energy. Heat pipes, vapour chambers, and thermosyphons have emerged as the most significant technology and cost-effective thermal solution owing to their excellent heat transfer capabilities, high efficiency, and structure simplicity.

Chapter 6 deals with some aspects of the heat pipe–based thermal control of fuel cells, solid sorption transformers, and electronic components and air-condition devices in a renewable energy context. Firstly, it briefly relates heat pipe design for fuel cells application, adsorption cooling and heating (including solar cooling), snow melting, and ground heating and cooling.

Finally, it outlines different heat pipe designs based on nanofluid and nanocoating application. The combination of fuel cell and solid sorption transformer is also discussed in this chapter based on heat pipe technology application.

The basic principles of adsorption cooling in terms of adsorption equilibrium, kinetics, and heat of adsorption are discussed in Chapters 2, 3, 7, and 8. Some thermally activated adsorption cooling cycles are overviewed. These systems have the following advantages: firstly, exploiting renewable energy or waste heat of temperature below 100°C, and secondly, using very low electricity for the circulation of heat transfer fluids (hot, cold, and chilled water). Finally, a three-bed dual evaporator-type advanced adsorption cooling cum desalination cycle is shown. The evaporators work at two different pressure levels and produce simultaneously cooling effects and potable water from a single low-grade energy input. Silica/water-based direct contact condensation and evaporation cycles can significantly improve heat and mass transfer. Advances in understanding and trends in studying the adsorption dynamics (Chapter 3) are the subject of prime interest as the booming progress in the materials science offers a huge choice of novel porous solids, which may be used for adsorption transformation of low-temperature heat. An overview of original and literature data on several classes of materials potentially promising for some very important applications, namely, metalaluminophosphates metal–organic frameworks, ordered porous solids, and various composites a useful for further its application in modern designs of adsorption coolers and heat pumps using heat pipe thermal management to increase the efficiency of sorption cycles. A mathematical model of coupled heat and mass transfer during the isobaric adsorption process has been used for detailed dynamical analysis and for extraction of the heat and mass transfer coefficients. The most important findings and general regularities that have been revealed for systematic studying of the adsorption dynamics of water, methanol, and ammonia in adsorption heat transformer systems are summarised, illustrated, and discussed.

The fundamental mechanisms of intensive heat transfer, which are observed not only in nucleate boiling but also in transition and film boiling (Chapter 4), are typical not only for heat pipes and thermosyphons but also for other two-phase heat transfer devices, including the subcooled liquid boiling. The availability of three-phase boundaries is considered as a peculiarity of nuclear boiling, which distinguishes it from other modes of convective heat transfer. It is important to note that the only mechanism that can explain the extremely high heat transfer intensity of boiling is evaporation in the vicinity of interlines, that is, the boundaries of contact of three phases. Chapter 7 covers some complex models of sorption and chemisorptions heat pumps and refrigerators. These models have been elaborated in Chapters 2, 3, 7, and 8 on the basis of the present state of the art in the field of transport phenomena in porous media. The models describe thermodynamics and kinetics of sorption processes as well as transport of single- and two-phase sorbate fluid in adsorbers and its phase transitions (condensation and evaporation). A set of computer programs for numerical modelling and optimization of sorption heat converters has been developed with the use of elaborated models.

The programs allow investigation of various heat conversion devices designed as a combination of one or several reactors with condensers and evaporators. The programs have the options for varying reactor parameters, describing sorption processes and heat and mass transport phenomena in it, temperatures of heat carrier thermal reservoirs, resistance of control valves connecting the reactors, and time of their opening and closing. As a result of modelling, the programs help to calculate main characteristics of thermal energy conversion such as the coefficient of performance, the specific heat production (specific cold production), and the maximal temperature difference between the working fluid and adsorbent during the process. Besides, the time and space distribution of temperatures and concentrations over the reactors is available in tables and graphical representation. The concept of flow boiling and flow condensation heat transfer in microchannels is analysed in detail in Chapter 12, which guarantees the novel cooling strategies in the areas requiring successful thermal management. The influence of capillary forces, disjoining pressure, wall roughness, and vapour shear stress on configuration of the microscale interface and the mathematical model of the annular flow in a rectangular microchannel is considered. The stability of meniscus in the channel corners and establishment of limit radius of meniscus curvature at high vapour velocities seem to be the limiting factors for the accumulation of liquid in the corners. A pressure drop model was proposed for rectangular microchannel which accounts the capillary pressure using statistical parameters of a gas–liquid flow. Some features of evaporating and condensing heat transfer in a small-sized channel at low heat fluxes are considered for closed and open microchannels. Experimental data and numerical modelling of evaporation and condensation inside the rectangular channel show the significant effect of capillary forces on film surface curvature and heat transfer.

We are very grateful to all the authors, Sassi Ben Nasrallah, Marc Prat, B.B. Saha, I.I. El-Sharkawy, Yuri I. Aristov, Victor V. Yagov, Masataka Mochizuki, Aliakbar Akbarzadeh, Thang Nguyen, O.S. Rabinovich, N.V. Pavlyukevich, M.Yu. Liakh, Larisa E. Kanonchik, Mikhail K. Bezrodny, V.M. Baturkin, Marcia Barbosa Henriques Mantelli, Vladimir V. Kuznetsov, and S.A. Safonov, who provided the substance of this book and its full success.

During the preparation of this book, many colleagues at the A.V. Luikov Heat and Mass Transfer Institute were involved in the fruitful collaboration. We are thankful to the members of the Porous Media Laboratory and A.V. Luikov Heat and Mass Transfer Institute. Our heartful thanks are extended to Drs. A.A. Antukh, Grakovich, A.G. Kulakov, D.A. Mishkinis, M.I. Rabetsky, and A.S. Zhuravlyov.

We also thank Laurie Schlags, project coordinator, Editorial Project Development, Jonathan W. Plant, executive editor, and other individuals at CRC Press for their close collaboration and guidance during the preparation of this book.

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Editors

L. L. Vasiliev is the president of the New Independent States (NIS) Scientific Association 'Heat Pipes' and a general leading scientist of Porous Media Laboratory, A.V. Luikov Heat and Mass Transfer Institute, National Academy of Sciences, Belarus. He was born on 3 January 1937 in Simferopol, Crimea, the USSR.

He started his scientific career with studies of thermal properties (thermal conductivity, heat capacity and thermal diffusivity) of solid materials at cryogenic temperatures and developed a new non-stationary method of their measurements (1960–1964) at A.V. Luikov Heat and Mass Transfer Institute under the supervision of Professor Alexis Luikov. He received his first doctoral degree (Candidate of Sciences) in 1964 in Minsk with the thesis 'Thermal Properties of Solid Materials at Cryogenic Temperatures'. In 1972, he got his second doctoral degree (Doctor of Science) in thermal physics from A.V. Luikov Heat and Mass Transfer Institute with the thesis presentation 'Heat and Mass Transfer in Capillary Porous Media Based on Heat Pipe Phenomena'. In 1978, he was elected as a professor of mechanical engineering in Belarusian Academy of Sciences. From 1967 to 1986, he was the head of Cryogenic Laboratory of A.V. Luikov Heat and Mass Transfer Institute. During the same period, he gave lectures on two-phase heat and mass transfer in capillary porous media in Belarusian State University and Belarusian Polytechnic University. In 1986, Cryogenic Laboratory of A.V. Luikov Heat and Mass Transfer Institute was renamed as Porous Media Laboratory, and Professor Vasiliev remains as the head of this laboratory.

Twenty-seven doctor/engineers can call him a supervisor; three were habilitated under his supervision. Three of his disciples are now the head of the laboratory in A.V. Luikov Heat and Mass Transfer Institute.

Professor Vasiliev was frequently invited to visit as a guest professor or leading scientist in scientific institutions all over the world. In 1967, he had his stage at the Aerothermodynamics Laboratory in Meudon, France, and spent 10 months there. During 1969–1982 period, he was regularly invited as visiting scientist to the Institute of Thermomechanics, Academy of Sciences of the Czech Republic, Prague, and to the State Institute of Machinery, Behovize, Czechoslovakia. Many times he was invited as a visiting professor to some institutions in Poland, Bulgaria, and Baltic countries.

He contributed his knowledge and scientific experience to many national and international scientific associations and committees.

From 1967 to 1975, he was a member of the editorial board of *Journal of Engineering Physics and Thermophysics*—National Academy of Sciences of Belarus (translated to English by Springer Inc.). Since 1981, he has been a member of advisory board of *Journal of Applied Thermal Engineering* (in the past, *Journal of Heat Recovery Systems*), Elsevier. Since 1997 he has been an associate editor of *International Journal of Thermal Sciences*, Elsevier; in January 2006, he was elected as a member of editorial board of *International Journal of Low Carbon Technologies*, Manchester University Press.

In 1976, he was elected as the member of the International Scientific Committee of the organizing conference on Heat Pipes. In 1991, he was elected as the president of the NIS Scientific Association Heat Pipes and enjoying his position till now. He was the editor of the Proceedings of the Seventh International Heat Pipe Conference (two volumes) (Begell House Inc., 1993), co-editor of *Microscale Heat Transfer—Fundamentals and Applications* (Springer, 2005), and co-editor of *Mini-Micro Fuel Cells—Fundamental and Applications* (Springer, 2008).

In 1993, he was the initiator of the series of international conferences 'Minsk Seminars—Heat Pipes, Heat Pumps, Refrigerators, Power Sources' and the editor of the eight Proceedings of these Conferences held in Minsk. The last one, 'Heat Pipes, Heat Pumps, Refrigerators, Power Sources', was held on 12–15 September 2011.

He was elected as scientific committee member of international conferences in Russia, Belarus, China, India, Poland, Canada, France, England, Brazil, Korea, Australia, and Italy several times.

Now he is a member of the Scientific Council of the International Centre for Heat and Mass Transfer, which demonstrates his esteem and respect in the scientific world.

In recognition of his national and international scientific reputation, Professor Vasiliev obtained the Arcot Ramachandran Professorship in Madras Institute of Technology, India, in 2005, and numerous prestigious awards such as Belarusian State Award (1981), USSR Council of Ministers Award (1983), and Czechoslovak Silver Medal (1986). He was awarded one gold, two silver, and one bronze medals of USSR Exhibitions on Advances of Industrial and Agricultural Products in Moscow, USSR. In 1988, he was awarded the silver medal of USSR Exhibitions for his book with co-authors *Hydrodynamic and Heat Transfer in Porous Components of Aircraft Design* (Mashinostroenie, 1988) in Moscow.

In 2005, he was given the Luikov Award and Medal of National Academy of Science of Belarus. In 2011, he was given the George Grover Distinguish Scientist Award and Medal as an appreciation for his outstanding contribution to the development of the science and technology of heat pipes (Committee on International Heat Pipe Conferences).

He authored or co-authored about 340 publications in journals and conference proceedings. He is the author and co-author of 14 books, has 218 Soviet author certificates on invention, and owns 12 patents.

His main fields of activity were and still are heat and mass transfer in capillary porous media with phase change, thermophysical properties of solid materials at low temperatures, heat pipes, heat pumps, heat exchangers, solid sorption machines, solar collectors, energy recovery systems, thermal and gas storage systems, electronic components cooling, and mini/microscale heat transfer with phase change.

In addition to the scientific excellence he demonstrated throughout his career, we want to emphasize the important role he played during his guidance for 45 years in the Porous Media Laboratory, A.V. Luikov Heat and Mass Transfer Institute, National Academy of Sciences. The practical importance of the efficiency of Porous Media Laboratory is the fact that his colleagues were awarded 34 medals for different merits in their scientific activity.

Sadık Kakaç was born on 1 October 1932 in Çorum, Turkey. He received his degree of Dip.-Ing (1955) from the Department of Mechanical Engineering at the Technical University of Istanbul (ITU).

Professor Kakaç joined the chair of Heat Technique as a research/teaching assistant (1955–1958) in the Department of Mechanical Engineering at ITU, and then he went to Massachusetts Institute of Technology (MIT) under a scholarship to study nuclear engineering; he received his MS in mechanical engineering in 1959 and his MS in nuclear engineering in 1960, both from MIT. Receiving a scholarship from the UNESCO in 1963, he went to UK, and in 1965, he received his PhD from the Victoria University of Manchester, UK.

He started his academic life in the Department of Mechanical Engineering at the Middle East Technical University (METU) (1960) as an instructor. He then became associate professor (1965) and professor (1970); then he also served different governmental and research positions; he was elected as a member of the Turkish Scientific and Technological Research Council (1972–1980) and appointed as the secretary general of the Turkish Atomic Energy Commission (1978–1980), representing Turkey in a number of scientific endeavours abroad as a member of NATO Science Committee (1979–1980), the OECD NEA Steering Committee (1978–1980), the Cento Scientific Coordinating Board (1972–1974). He served as the chairman of the Department of Mechanical Engineering at METU (1976–1978), and then he was invited as a visiting professor to the Department of Mechanical Engineering at the University of Miami (1980–1982), and in 1982, he was appointed as a full-time professor of the Department of Mechanical Engineering at the University of Miami with tenure. He served as the chairman of the department during 1990–1998.

He has maintained his high quality of research for more than 50 years. He has authored or co-authored, publishing over 200 scientific papers on transient and steady-state laminar, turbulent forced convection, two-phase flow instabilities, fuel-cells modelling, micro heat transfer with sleep flow, and recently, he has been concentrated in research on convective heat transfer enhancement with nanofluids in single-phase and two-phase flow conditions. He performed very valuable contributions in all these areas, and he has been known as one of the most recognized scientists in the field of heat transfer.

He is highly respected by his peers. He has received many international recognition: Alexander von Humboldt Senior Distinguished U.S. Scientist Award for his outstanding contributions on heat transfer and two-phase flow was bestowed to him in 1989; Science Award by the Association of Turkish–American Scientists in 1994; American Society of Mechanical Engineers (ASME) Heat Transfer Memorial Award in 1997. Because of his contributions to research and education through his research work and books, he received distinguished service awards from the Middle East Technical University in 1998 and from the Turkish Scientific and Technical Research Council in 2000. He has received a 'lifetime achievement' award from the TOBB University of Economics and Technology, Ankara (2010) and iNEER Leader Award for his contribution to International Network for Engineering Education and Research (2012).

He received the Doctor Honoris Causa from the University of Ovidius, Romania (1998), the University of Reims, France (1999), and Odessa State Academy of Refrigeration (2007). He is a member of the Turkish Academy of Sciences (1999), and a foreign member of the Academy of Sciences of the Republic of Bashkortostan of Russian Federation (1998), and the Brazilian Academy of Sciences (2012). He is an honorary professor of Shanghai Institute of Electrical Power (1986), Xi'an Jiaotong University, China (1988), and Gandhi Institute of Technology and Management College of Engineering, India (1992); he is a fellow member of ASME.

He is a member of the scientific council and executive committee, and fellow of the International Centre of Heat and Mass Transfer.

He has organized and directed well-known NATO Advanced Study Institutes on various topics of thermal and fluid sciences for the last 35 years bringing the eminent scientists as lecturers and the young scientists together from NATO and non-NATO countries for exchange of ideas, educating young scientists, and dissemination of information on emerging topics providing very valuable service to the heat transfer community.

He has been frequently invited as lecturer and speaker by various institutions in the United States, Europe, China, Malaysia, Singapore, Brazil, and other countries; he organized short courses on the thermal design of heat exchangers in Taiwan, Singapore, Thailand, France, and Turkey.

He has been serving as an editor in editorial board of major journals on heat and mass transfer, and energy. He is the author or co-author of very popular textbooks of *Convective Heat Transfer* (CRC Press, 1995), *Thermal Design of Heat Exchangers* (CRC Press, 1997), and *Heat Conduction* (CRC Press, 2008); he edited 14 volumes in the field of thermal sciences and heat exchanger fundamentals and design, including *Handbook of Single-Phase Convective Heat Transfer* (John Wiley, 1987) and *Boilers, Evaporators, and Condensers* (John Wiley, 1991), which all became permanent reference books in the field.

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1

A Review of Modelling Approaches to Heat and Mass Transfers in Porous Wicks

Sassi Ben Nasrallah and Marc Prat

CONTENTS

1.1 Introduction

Although the technology of two-phase cooling systems, such as conventional heat pipes or loop heat pipes (LHPs) or capillary pumped loops (CPLs), is well developed and of widespread use (Peterson 1994, Faghri 1995), many aspects of it are not well understood (Smirnov 2010). Furthermore, the improved design of such systems still often requires costly series of trial and error experimental campaigns. It is, therefore, desirable to develop numerical tools aiming at both improving our understanding of systems and helping the engineer in designing improved systems with a limited number of experimental tests. The increasing demand for miniaturised devices is also a factor for developing advanced numerical simulations. Developing numerical simulations can be interesting at various scales: at the scale of the system (a whole LHP, e.g. as in Launay et al. [2007] and Kaya et al. [2008]), a component of the system (e.g. the evaporator or the condenser of an LHP), a subelement of a component (the porous wick of a heat pipe or an LHP) or even at smaller scales (e.g. the cross section of a single groove of a heat pipe; see Stephan and Busse 1992). In this chapter, we focus on the numerical simulation tools at the porous wick scale. The porous wick is a key component of the evaporator in an LHP and is often encountered in conventional heat pipes as well as in many other devices (thermal spreaders, etc.). The coupled heat and mass transfers with liquid–vapour phase change occurring in an LHP evaporator porous wick are encountered in many other domains of engineering, such as nuclear engineering (Lipinski 1984, Fichot et al., 2006), geothermal engineering (Woods 1999), and drying (Perre et al., 1993). Although our interest is mainly cooling devices, the models presented in this chapter are clearly of broader interest.

As sketched in Figure 1.1, simulations and the associated models can be developed at various scales. The microscopic approaches refer to *ab initio* simulations, molecular dynamics and associated theoretical approaches, such as the density functional theory. These approaches are useful for the fundamental study of the phenomena at the scale of a single pore (typically of nanometric size) over a very small time scale. To the best of our knowledge, this type of approach has not yet been applied in relation with the study of transfer in the porous wick of two-phase cooling devices, and therefore is not discussed in this chapter. The next scale is the pore scale (also referred to sometimes as 'mesoscale'), in which direct approaches can be developed. The governing equations are solved at the pore scale over digital porous structures. On account of computational costs, the direct simulations are generally limited to domains containing only a few pores. By contrast, the pore-network simulations allow much larger domains containing up to 10^6 pores, but with important simplifications in the modelling of transport phenomena between two adjacent pores. The last scale considered

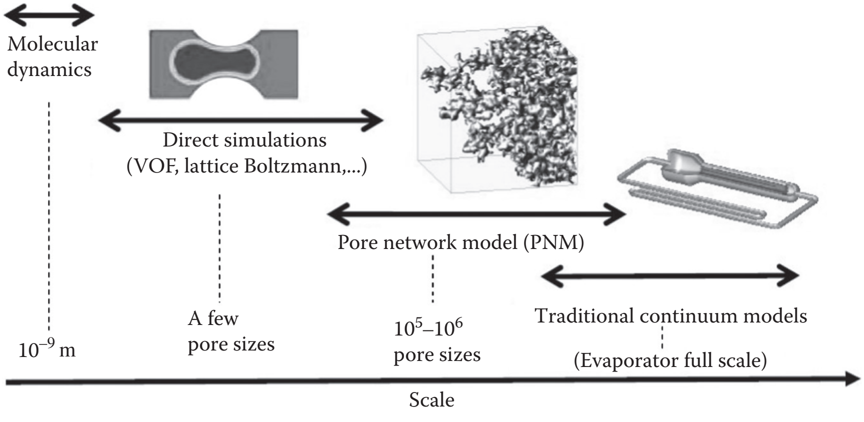

FIGURE 1.1
Various methods for the numerical simulation of transport phenomena in porous media.

in this chapter is Darcy's scale. This scale corresponds to the most classical approach, in which the porous medium is viewed as a fictitious continuum or a set of interacting fictitious continua.

In this chapter, we give an overview of these various approaches (with the exception of microscopic approaches), with a special focus on pore-network and continuum approaches.

1.2 Digital Porous Media

Before presenting the various simulation approaches, it is worth mentioning the concept of digital porous media. Digital porous media refer to numerical porous structures. These numerical porous structures can be used as input data for the approaches described in Sections 1.3 through 1.5, especially with regard to the direct and the pore-network approaches. Three-dimensional (3D) numerical porous structures can be obtained, for example, from direct imaging of real porous media using micro x-ray computerised tomography (Coker et al., 1996, Manke et al., 2011), nuclear magnetic resonance imaging (Baldwin et al., 1996) or focused ion beam scanning electron microscopy (Karwacki et al., 2011). As an example, Figure 1.2 shows the reconstructed

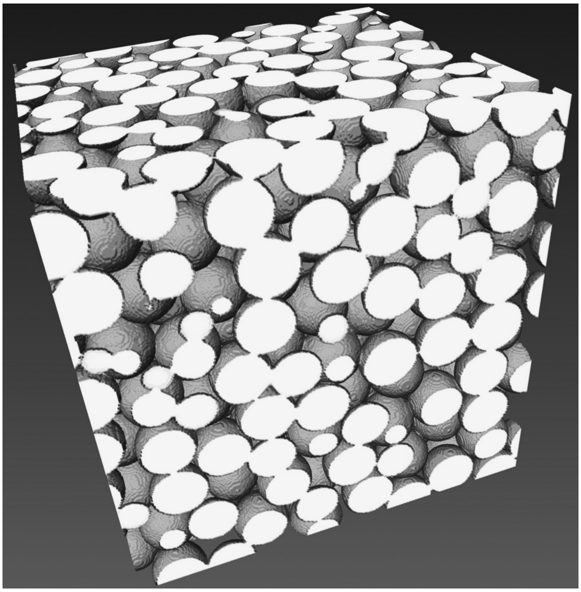

FIGURE 1.2
Digital random packing of spherical particles obtained from a real random packing using micro x-ray computerised tomography at IFPEN. (From Horgue, P., *Ph.D. dissertation*, University of Toulouse, 2012.)

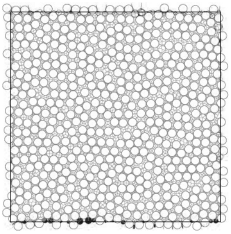

FIGURE 1.3
Example of numerically generated porous matrix.

image of a real random packing of spherical particles obtained using micro x-ray computerised tomography (Horgue 2012).

A popular alternative is to generate the microstructures numerically. This can be done starting from some statistical properties of a real porous structure (Liang et al., 1998) or purely numerical (Torquato 2002), a typical example being the numerical generation of a random packing of spherical particles (Soppe 1990).

Figure 1.3 shows a simple example in which the porous matrix is formed by a two-dimensional (2D) array of randomly distributed discs.

Once a numerical microstructure is obtained, the next step is to compute the transport phenomena of interest over this microstructure, that is, in the pore space (fluid flows, for instance) together possibly with the transport in the solid phase (in the case of thermal transfer, for instance).

1.3 Direct Approaches

In direct approaches, the transport problem of interest is directly solved at the pore scales. Increasingly popular examples in this category are the Lattice Boltzmann method (LBM) (Chen and Doolen 1998) and the direct simulations with diffuse interface models (Anderson et al., 1998). Whereas the latter are based on the discretisation of the Navier–Stokes equations, the former treats a fluid as an ensemble of artificial particles. Although the LBM method is perhaps more popular in the porous media literature (Olson and Rothman 1997, Porter et al., 2009), notably because complex geometry can be handled relatively simply with this method, diffuse interface models are also well adapted to direct simulations in microstructures of complex shapes. It is therefore anticipated that these methods will also be frequently used in future works. As an example, Figure 1.4 shows the simulation of an immiscible two-phase flow in a 2D array of discs performed using the so-called volume of fluid (VOF) method (Hirt and Nichols 1981), implemented in the commercial code Fluent™ (Horgue 2012).

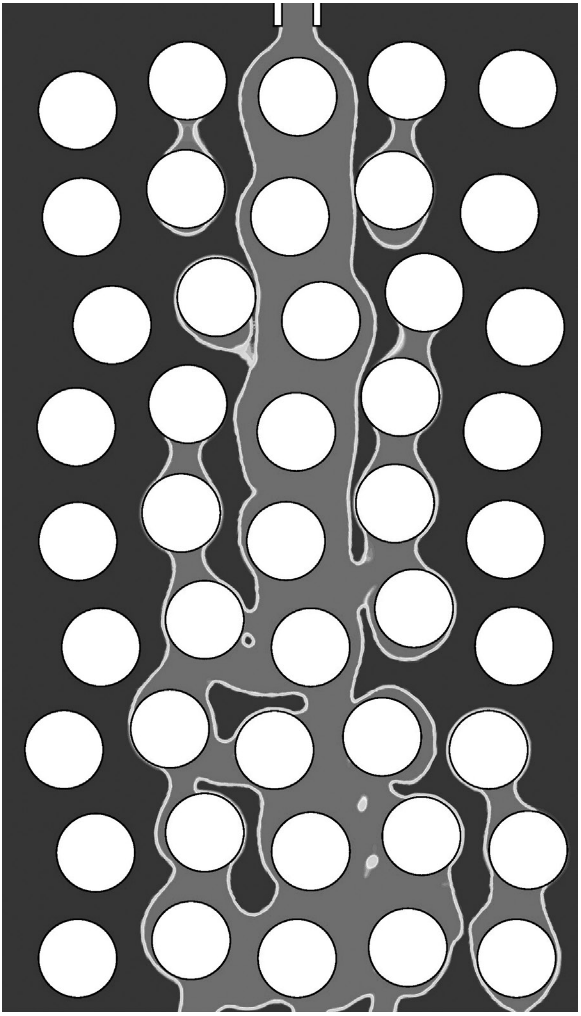

FIGURE 1.4
Example of direct simulation of two-phase flow. Liquid (in light grey) is injected from the top and flows through the random array of discs (in white) under the action of gravity and viscous effects. Note the gas (in dark grey) entrainment induced by this gravity-driven liquid flow. (From Horgue, P., *Ph.D. dissertation*, University of Toulouse, 2012.)

Although these methods seem very attractive since they allow studying the process at the pore scale without any further modelling or simplifications, they have serious computational limitations. As discussed in more depth in Joekar-Niasar et al. (2010), the direct methods are presently too memory and computational-time demanding for simulating transport phenomena over a representative elementary volume (REV) of a microstructure. The REV size being itself generally small compared to the size of a porous domain in most engineering applications, it is clear that the direct simulation of transport phenomena over the whole wick of an LHP evaporator is completely out of reach for the moment. This also explains why, despite the apparent attractiveness of direct methods, little results of practical interest have been obtained so far using this type of method (as far as two-phase flows or even more complex processes are concerned). Although most of the works with direct methods in relation with porous media can be essentially categorised as illustrative or methodological, it is surmised that this type of method could be advantageously used in conjunction with other methods, such as the pore-network methods described in Section 1.4.

As a result, it is not surprising that this type of method has practically not been used so far for the study of transfer phenomena in evaporator porous wick. The only work of which we are aware (Xuan et al., 2011) can be seen as a good example of the computational limitation of the direct method. The computational domain is 2D and of unrealistic high porosity, much too small compared with practical sizes as well as in number of pores considered (only a few pores), the solid phase is formed of isolated particles (the solid phase is thus non-percolating), the wick is fully liquid saturated and so on.

A partial conclusion is that direct methods can be very useful for the careful study of phenomena at the scale of one or a few pores, so as to provide guidance or reference solutions, for instance, for modelling adapted to larger scales. When the objective is the simulation at the scale of an REV or a whole porous domain, direct methods are of no practical interest because of computational limitations. In this case, it is much more appropriate to rely on the pore network models (PNMs) or continuum models described now. Naturally, it can be expected that significant increases in computer performances might change this situation in the future.

1.4 Pore-Network Approach

Initially developed mostly in relation with oil recovery problems, pore scale models have been successfully applied to many other domains, including the study of heat and mass transfers with vaporisation in the porous wick of LHPs (Figus et al., 1999, Prat 2010, Louriou and Prat 2012). As given in Figure 1.5, PNMs are based on the representation of pore space in terms of a network of pores (or sites) connected by throats (or bonds). The 'pores' correspond roughly to the larger voids whereas the throats connecting the pores correspond to the constrictions of the pore space. As discussed in Prat (2010), the pore network can be constructed from the digitally reconstructed porous structure (see Section 1.3). This leads to 'morphological' or

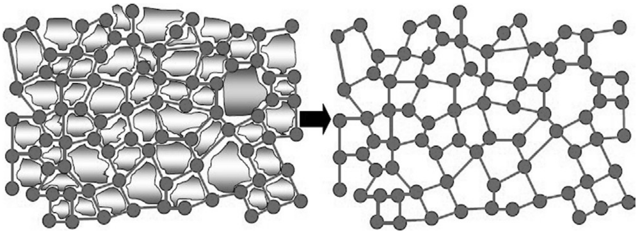

FIGURE 1.5
Schematic of a pore network. The pore space is represented as a network of sites (~ intersections of the pore space) connected by bonds (corresponding to the constrictions of the pore space).

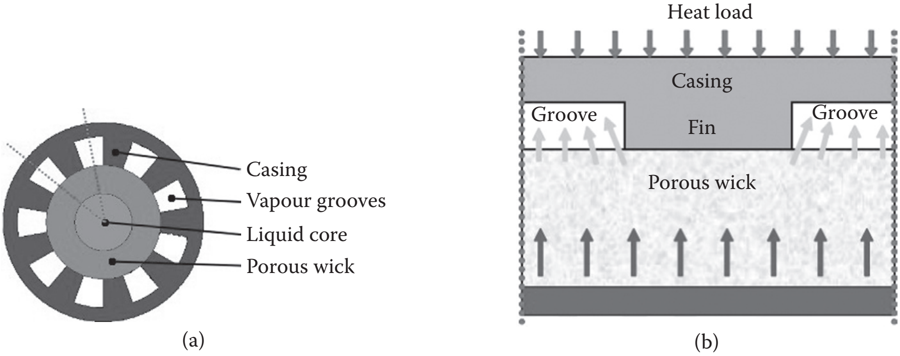

FIGURE 1.6
(a) Cross section of a capillary cylindrical evaporator (b) two-dimensional unit cell of a capillary evaporator.

'unstructured' pore networks. Although using morphological pore networks is certainly the most attractive approach, using simpler pore networks can be very instructive.

In fact, morphological pore networks have not yet been used in relation with LHP-related problems. The available studies (Prat 2010 and references therein) are based on simple 2D square networks. Computations are easier than for a morphological network, and it is still possible to incorporate information from the 'real' microstructure such as throat and pore size distributions. Also, to understand some effects, a simple network is generally sufficient. However, it can be anticipated that 3D morphological networks will be used in future works. Accordingly with the above description, only simple networks are considered in what follows.

Since a recent review is available (Prat 2010), only a short overview of the technique and main results is proposed in this chapter. As presented in many previous studies, a typical LHP consists of an evaporator, a reservoir (usually called a compensation chamber), vapour and liquid transport lines and a condenser. As illustrated in Figure 1.6, the evaporator consists of a liquid-passage core, vapour-evacuation grooves, an outer casing and a capillary porous wick. Heat applied to the outer casing leads to the vaporisation of the liquid inside the wick or at the wick/vapour groove interface. The produced vapour is collected in the vapour grooves and flows through the vapour transport line towards the condenser. The menisci formed at the wick/grooves interface or inside the wick adjust themselves to establish a capillary suction that balances the total pressure drop in the device. Pore network (PN) simulations are developed for analysing the transfers within the evaporator, which clearly represents a crucial component of an LHP.

Figure 1.6 shows a sketch of the cross section of a cylindrical evaporator. Although real systems are often more complicated and may include, for example, a secondary wick structure and a bayonet, only the simpler situation depicted in Figure 1.6 has been studied so far using PNM. Owing to the spatial periodicity of the evaporator structure, computations are restricted to a unit cell of the structure, such as the one shown in Figure 1.6. PNMs have been used to study steady-state regimes (Figus et al., 1999, Coquard 2006, Coquard et al., 2007), as well as transient regimes (Prat 2010, Louriou and Prat 2012). For the sake of brevity, we consider some results only for the steady-state regimes in this chapter. The main results obtained for transient regimes are summarised at the end of this section.

As discussed, for example, in Figus et al. (1999) and sketched in Figures 1.6b and 1.7, two main regimes are distinguished regarding the phase distribution within the wick. For low-to-moderate heat loads (Figure 1.6b), the wick is assumed to be fully saturated (this situation is referred to as the all-liquid wick) and vaporisation takes place at the wick/groove interface. Above a certain heat load, bubble nucleation occurs within the wick, leading to the formation of an internal vaporisation front, as illustrated in Figure 1.7. The

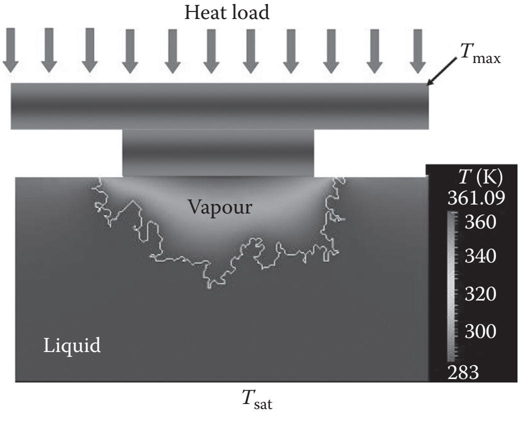

FIGURE 1.7
Steady-state regime with well-developed vapour pocket.

situation with the vapour pocket is referred to as the vapour–liquid wick. To describe the transition from the all-liquid wick to the vapour–liquid wick, the simplest approach is to assume that the transition occurs when a given boiling incipient superheat value is reached somewhere under the fin, which is the hottest region within the wick. As reported in Prat (2010), visualisation experiments in a model porous medium have confirmed the development of the vapour pocket inside the porous medium.

The PNM summarised in what follows is the one developed in Coquard (2006). Compared to a previous model presented in Figus et al. (1999), several new features were introduced: the conduction heat transfer in the external casing is computed, convection heat transfer is taken into account within the wick in addition to heat conduction and phase change, variations in density in the vapour phase with pressure and temperature are taken into account assuming an ideal gas behaviour and heat loss due to the vapour convective flow in the vapour grooves is taken into account. To facilitate the parametric studies, the model developed in Coquard (2006) also presents some significant differences in terms of model formulation. In the classical PNM, the disorder of the porous structure is taken into account through the random distribution of pore volumes and throat sizes. The disorder affects the transport since the throat conductance (see Equation 1.2) varies randomly in the hydraulic network and the local thermal conductance of a bond in the thermal network varies randomly as well (Figus et al., 1999). The randomness of the wick microstructure also affects the capillary effects since the capillary invasion threshold associated with a throat (of radius r) of the hydraulic network is defined as (assuming a perfectly wetting liquid)

$$p_{\mathrm{cth}} = \frac{2\sigma}{r} \tag{1.1}$$

This local capillary pressure, Equation 1.1, represents the pressure difference across a meniscus in a throat that must be overcome for the throat to be invaded. Since r is a random variable, p_{cth} varies randomly in the hydraulic network.

In this classical conception of PNMs, the pore/throat size distribution is imposed and the permeability of the network must be computed afterwards (Prat 2010). Similarly, the effective thermal conductivity is computed once a local thermal conductance is assigned to each bond of the thermal network. This is not convenient in the prospect of a parametric study on the effect of the wick macroscopic properties (permeability, effective thermal conductivity, etc.). It is much more convenient to impose macroscopic transport properties directly as input parameters.

Since the pore size distribution in an evaporator wick is expected to be rather narrow, a simple solution is to neglect the effect of disorder on the transport while considering its effect on capillarity. (We know from previous studies that the shape of the liquid–vapour front and associated capillary fingering directly depend on the capillary effects associated with porous medium disorder.) Such a PNM in which the disorder of the porous medium is fully taken into account regarding capillarity but considered in an average sense regarding the computation of the transport phenomena is called a mixed PNM. For example, if the desired permeability of the porous medium is k, the hydraulic conductance of each bond in the square network is given by $g = k\,e$, where e is the 2D network thickness. Similarly, the thermal conductance of a bond of the thermal network is specified as a function of the desired thermal effective conductivity, which typically takes two values depending on the fluid, liquid or vapour, occupying the pore space. An additional advantage of the mixed PNM is that the nodes of the thermal and hydraulic networks representing the wick are the same. Hence, contrary to a classical PNM (Figus et al., 1999), it is not necessary to use a finer network for the thermal network.

Under these circumstances and under the assumptions of a steady-state process, homogeneous and isotropic capillary structure, negligible gravitational and radiative effects and local thermal equilibrium between the porous structure and the working fluid, the problem to be solved within the framework of the mixed PNM can be expressed as follows (for the most complicated case, i.e. in the presence of the vapour pocket, the all-liquid wick situation is a simpler subcase):

Heat conduction in casing:

$$\nabla \cdot \left(\lambda_a \nabla T\right) = 0 \tag{1.2}$$

Heat transfer and flow in porous wick:

$$(\rho C_p)_i \mathbf{u}_i \cdot \nabla T = \nabla \cdot (\lambda_{\text{effi}} \nabla T) \tag{1.3}$$

$$\nabla \cdot \rho_i \mathbf{u}_i = 0 \tag{1.4}$$

$$\mathbf{u}_i = -\frac{k}{\mu_i} \nabla p_i \tag{1.5}$$

with $i = \ell$ in the wick liquid region (where $\rho_\ell = \text{cst}$) and $i = v$ in the wick vapour region (where $\rho_v = p_v M/RT$). The Clausius–Clapeyron relationship:[2]

$$p_s(T_{\text{sat}}) = p_{\text{ref}} \exp\left(-\frac{h_{vl} M}{R}\left[\frac{1}{T_{\text{sat}}} - \frac{1}{T_{\text{ref}}}\right]\right) \tag{1.6}$$

Continuity and energy conservation conditions at the vapour/liquid interface within the wick:

$$T_\ell = T_{\text{v}} = T_{\text{sat}}(p_{\text{v}}) \tag{1.7}$$

$$\rho_{\mathrm{v}}\mathbf{u}_{\mathrm{v}} \cdot n \;=\; \rho_{\ell}\mathbf{u}_{\ell} \cdot n \tag{1.8}$$

$$\left[-\lambda_{\mathrm{effl}}\nabla T\right]\cdot n = \left[-\lambda_{\mathrm{effv}} \cdot \nabla T\right]\cdot n + h_{\mathrm{vl}}\,\dot{m} \tag{1.9}$$

Boundary condition and energy conservation conditions at the wick/casing interface:

$$\nabla p \cdot n = \; 0 \tag{1.10}$$

$$\left[-\lambda_{\mathrm{effi}}\nabla T\right]\cdot n = \left[-\lambda_{a} \cdot \nabla T\right]\cdot n \quad i = l\,\mathrm{or}\,v \tag{1.11}$$

Boundary conditions at the porous wick/groove interface:

$$-\lambda_{\mathrm{effl}}\nabla T \cdot n = h_{\mathrm{vl}}\,\dot{m} \quad \text{or} \quad \lambda_{\mathrm{effl}}\nabla T \cdot n = h_{\mathrm{c}}(T - T_{\mathrm{gr}}) \tag{1.12}$$

where the convection heat transfer coefficient h_{c} is specified as in Kaya and Goldak (2006).

Boundary condition at the casing/groove interface:

$$\lambda_{a}\nabla T \cdot n = h_{\mathrm{c}}(T - T_{\mathrm{gr}}) \tag{1.13}$$

Boundary condition at the entrance of the wick (wick/liquid–passage core interface):

$$T \;=\; T_{\mathrm{sat}} - \Delta T_{o}, p = p_{\mathrm{s}}(T_{\mathrm{sat}}) \tag{1.14}$$

Boundary condition at the casing external boundary (heat flux):

$$\lambda_{a}\nabla T \cdot n = Q \tag{1.15}$$

Spatial periodicity conditions are imposed on the lateral sides of the computational domain.

The method of solution is similar to that presented in Figus et al. (1999). Once discretised on the network, Equations 1.2 through 1.15 lead to systems of equations quite similar to the ones obtained using standard finite difference or finite volume techniques, the mesh in the wick being formed by the sites of the pore network. Note that computational nodes are also placed in the metallic casing so as to compute the heat transfer within the casing and the fin.

Hereafter, we summarise the procedure when a vapour pocket exists within the wick. The all-liquid wick situation is easier to compute. Initially, the first row of pores under the fin is assumed to be occupied by vapour (experimental visualisations have revealed a quick vapour invasion along the fin). Equations 1.2 through 1.15 are solved using an iterative method as described in Coquard (2006). This gives, in particular, the pressure field in the vapour and liquid regions within the wick. Then we compute the pressure jump $\delta p = p_v - p_\ell$ over each meniscus present within the wick or at the wick/groove interface. Each meniscus for which the computed pressure jump is greater than its capillary pressure threshold (see Equation 1.1), that is, $\delta p > (2\sigma/r)$, is moved into the adjacent pore. This gives a new phase (liquid/vapour) distribution within the wick. Equations 1.2 through 1.15 are then solved again until convergence for this new phase distribution and stability of menisci is tested as explained before, possibly leading to new pore invasions. This procedure is then repeated until the criterion of meniscus stability, that is $\delta p \leq (2\sigma/r)$, is fulfilled for each meniscus present in the wick.

The interesting feature of the model is that it permits us to track explicitly the position of the liquid/vapour interface within the wick. This is illustrated in Figure 1.7, which shows a typical example of phase distribution and temperature field obtained with this model. More details (mesh size, computational domain size, fluid properties, etc.) on this computation can be found in Coquard et al. (2007). Note the irregular (fractal) internal vaporisation front typical of this type of invasion process controlled by the competition between viscous and capillary effects (Prat and Bouleux 1999).

As exemplified in Coquard et al. (2007), this model is well-adapted to explore the impact of the wick transport properties on evaporator performance. To illustrate this, the casing overheat is defined as

$$\Delta T_{\max} = T_{\max} - T_{\text{sat}} \tag{1.16}$$

where $T_{\max}$ is the maximum temperature of the casing and T_{sat} the saturation temperature. The temperature of the casing cannot exceed a certain temperature, and this defines a casing overheat operating limit.

The heat flux balance over the computational domain can be expressed as follows:

$$Q = Q_{\text{p}} + Q_{\text{c}} + Q_{\text{l}} + Q_{v} + Q_{\text{evap}} \tag{1.17}$$

where Q is the applied heat load, Q_p the parasitic heat flux (which is the heat flux lost by conduction at the entrance of the wick), Q_c the heat flux lost by convection in the groove, Q_l the heat flux to heat up the liquid, Q_v the heat flux to heat up the vapour and Q_{evap} the heat flux used to vaporise the liquid. Of special interest is the parasitic heat flux, which can possibly contribute to heat up the compensation chamber and therefore modify the operating thermodynamic condition in the loop. It is therefore desirable to limit the parasitic heat flux.

Figure 1.8a shows the variation in the casing overheat as a function of the applied heat load for different values of the thermal conductivity λ_m of the porous matrix. It shows that a high thermal conductivity leads to a much greater casing overheat operating limit compared with a wick of low thermal conductivity. However, as illustrated in Figure 1.8b, a high

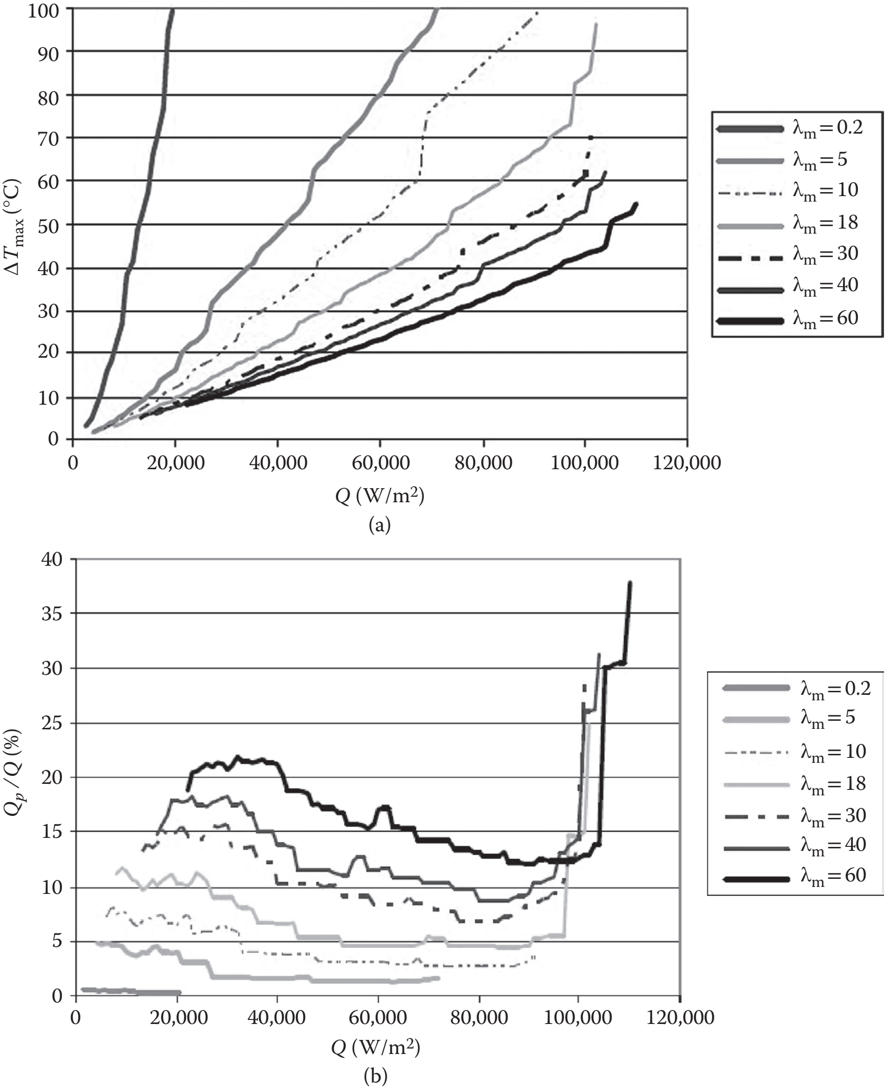

FIGURE 1.8
(a) Evolution of casing overheat as a function of applied heat load for different values of porous matrix thermal conductivity (b) evolution of parasitic flux as a function of applied heat load for different values of porous matrix thermal conductivity.

conductivity increases the parasitic heat flux. Together with other results concerning the capillary operating limit (not shown here), this suggests that a two-layer wick should lead to improved performance. The layer in contact with the casing should be of relatively high thermal conductivity (so as to limit the casing overheat) whereas the layer in contact with the liquid artery should be of lower thermal conductivity (so as to limit the parasitic heat losses) and with smaller pores (so as to avoid the vapour breakthrough across the wick). As shown in Coquard et al. (2007), this two-layer wick leads to better performance, at least according to this type of simulation.

As mentioned earlier, transient regimes have been studied as well (Louriou 2010, Prat 2010). The numerous results obtained include the identification of a vapour pocket pressurisation scenario leading to possible premature vapour breakthrough across the wick, oscillation phenomenon due to a succession of liquid reinvasions/vapour breakthroughs in the groove, hysteretic effects induced by non-monotonous changes in the applied heat load and a heat pipe effect within the wick (Louriou and Prat 2012).

1.5 Models Based on the Continuum Approach

1.5.1 Introduction

Within the framework of the continuum approach to porous media, the porous material is seen as a fictitious continuum medium or as a set of several interacting continua in which three phases – namely the liquid (l), solid (s) and vapour (v) – can be present. The porous structure is assumed to be rigid and inert. The heat and mass transfer governing equations in this system can be, in principle, obtained using an upscaling technique (Whitaker 1999), allowing one to obtain the governing equations of interest from those expressed at the pore scale through a series of mathematical operations. A popular approach in this context relies on the concept of REV. Such an elementary or averaging volume V is associated with every point in space. Then, the macroscopic transport equations are obtained by averaging the pore scale equations over this volume and by using closure hypotheses (Whitaker 1999). For some variable p_β associated with the β-phase, the pressure in the β-phase for example, the method consists of defining the corresponding variable at the scale of the fictitious continuum as $\langle p_\beta \rangle^\beta$ with

$$\langle p_\beta \rangle^\beta = \frac{1}{V_\beta} \int_{V_\beta} p_\beta \mathrm{d}V \tag{1.18}$$

where V_β represents the volume of the β-phase contained within the averaging volume V. However, the upscaling techniques are difficult to use in the presence of moving menisci at the pore scales. As a result, it is more exact to consider the equations presented in this section as mostly empirical and phenomenological.

The macroscopic equations governing the simultaneous heat and mass transfer in the different regions are as follows:

In the single-phase regions:

Mass balance equations:

$$\frac{\partial \varepsilon \rho_l}{\partial t} + \nabla \cdot (\rho_l \mathbf{u}_l) = 0 \tag{1.19}$$

$$\frac{\partial \varepsilon \rho_v}{\partial t} + \nabla \cdot (\rho_v \mathbf{u}_v) = 0 \tag{1.20}$$

where ρ_l and ρ_v are the liquid and vapour densities, $\mathbf{u}_l$ and $\mathbf{u}_v$ are the average liquid and vapour flow velocities and ε is the porosity.

Momentum equations:

$$\mathbf{u}_l = -\frac{k}{\mu_l}\left[\nabla p_l - \rho_l \mathbf{g}\right] \tag{1.21}$$

$$\mathbf{u}_v = -\frac{k}{\mu_v}\left[\nabla p_v - \rho_v \mathbf{g}\right] \tag{1.22}$$

where k is the absolute permeability, μ_l and μ_v are the liquid and vapour viscosities, $\mathbf{g}$ is the gravity vector and p_l and p_v are the liquid and vapour pressures.

In the two-phase regions:

Mass balance equations:

Assuming that the liquid and vapour densities do not vary significantly within the averaging volume, the macroscopic mass balance equations are expressed as

$$\frac{\partial \varepsilon s \rho_l}{\partial t} + \nabla \cdot (\rho_l \mathbf{u}_l) = \dot{m} \tag{1.23}$$

$$\frac{\partial \varepsilon (1-s) \rho_v}{\partial t} + \nabla \cdot (\rho_v \mathbf{u}_v) = -\dot{m} \tag{1.24}$$

where s is the liquid saturation ($s = \varepsilon_l/\varepsilon$) and $\dot{m}$ is the vaporisation rate.

Momentum equations:

Using the generalised Darcy laws, the liquid and vapour flow velocities $\mathbf{u}_\mathrm{l}$ and $\mathbf{u}_\mathrm{v}$ are given by

$$\mathbf{u}_\mathrm{l} = -\frac{kk_\mathrm{rl}}{\mu_\mathrm{l}}\left[\nabla p_\mathrm{l} - \rho_\mathrm{l} g\right] \tag{1.25}$$

$$\mathbf{u}_\mathrm{v} = -\frac{kk_\mathrm{rv}}{\mu_\mathrm{v}}\left[\nabla p_\mathrm{v} - \rho_\mathrm{v} g\right] \tag{1.26}$$

where k_rl and k_rv are the relative permeabilities of liquid and vapour.

Capillary pressure:

The macroscopic capillary pressure, which results from the presence of curved menisci within the averaging volume, is given by

$$p_\mathrm{c} = p_\mathrm{v} - p_\mathrm{l} \tag{1.27}$$

The capillary pressure depends on the pore geometry, fluid physical properties and phase saturation. For a given porous medium and a given fluid, the capillary pressure and the relative permeabilities are generally assumed to depend only on saturation s.

Energy equations: The macroscopic description of heat transfer in a porous medium subject to a two-phase flow with phase change is often investigated using a single-temperature equation model. This model is based on the local thermal equilibrium assumption that means that the macroscopic (averaged) temperatures of the three phases are close enough so that a single temperature suffices to describe the heat transport process. The concept of thermal equilibrium involves the volume-averaged temperatures and should therefore not be confused with the classical assumption of local thermodynamic equilibrium. The local thermal equilibrium is no longer valid when the particles or pores are not small enough, when the phase thermal properties differ widely, when convective transport is important and when there is a significant internal heat generation (Duval et al., 2004). Under these conditions, separate transport equations for each phase are required leading to non-equilibrium models.

1.5.2 Non-Equilibrium Models

On the basis of a steady-state closure at the microscopic scale, a comprehensive generalised three-temperature model taking into account the vaporisation in the porous medium was presented in Duval et al. (2004). When the non-traditional convective terms are negligible and the three macroscopic temperature gradients are sufficiently close to each other, the thermal energy equations of this model can be expressed in the two-phase region as follows:

In the vapour phase:

$$\begin{aligned}&\rho_{\mathrm{v}}\varepsilon(1-s)C_{p\mathrm{v}}\frac{\partial T_{\mathrm{v}}}{\partial t}+\rho_{\mathrm{v}}C_{p\mathrm{v}}\mathbf{u}_{\mathrm{v}}\cdot\nabla T_{\mathrm{v}}-\dot{m}C_{p\mathrm{v}}(T_{\mathrm{sat}}-T_{\mathrm{v}})\\&\quad=\nabla\cdot\left(\lambda_{\mathrm{effv}}\nabla T_{\mathrm{v}}\right)-H_{\mathrm{vv}}(T_{\mathrm{v}}-T_{\mathrm{sat}})-H_{\mathrm{vl}}(T_{\mathrm{l}}-T_{\mathrm{sat}})-H_{\mathrm{vs}}(T_{\mathrm{s}}-T_{\mathrm{sat}})\end{aligned}\tag{1.28}$$

In the liquid phase:

$$\begin{aligned}&\rho_{\mathrm{l}}\varepsilon sC_{p\mathrm{l}}\frac{\partial T_{\mathrm{l}}}{\partial t}+\rho_{\mathrm{l}}C_{p\mathrm{l}}\mathbf{u}_{\mathrm{l}}\cdot\nabla T_{\mathrm{l}}+\dot{m}C_{p\mathrm{l}}(T_{\mathrm{sat}}-T_{\mathrm{l}})\\&\quad=\nabla\cdot\left(\lambda_{\mathrm{effl}}\nabla T_{\mathrm{l}}\right)-H_{\mathrm{ll}}(T_{\mathrm{l}}-T_{\mathrm{sat}})-H_{\mathrm{lv}}(T_{\mathrm{v}}-T_{\mathrm{sat}})-H_{\mathrm{ls}}(T_{\mathrm{s}}-T_{\mathrm{sat}})\end{aligned}\tag{1.29}$$

In the solid phase:

$$\rho_{\mathrm{l}}(1-\varepsilon)C_{p\mathrm{s}}\frac{\partial T_{\mathrm{s}}}{\partial t}=\nabla\cdot\left(\lambda_{\mathrm{effs}}\nabla T_{\mathrm{s}}\right)-H_{\mathrm{ss}}(T_{\mathrm{s}}-T_{\mathrm{sat}})-H_{\mathrm{sv}}(T_{\mathrm{v}}-T_{\mathrm{sat}})-H_{\mathrm{sl}}(T_{\mathrm{s}}-T_{\mathrm{sat}})\tag{1.30}$$

where T_{s}, T_{l} and T_{v} are the volume-averaged temperatures of the solid, liquid and vapour phases, respectively. T_{sat} is the saturation temperature. C_{ps}, $C_{p\mathrm{l}}$ and $C_{p\mathrm{v}}$ are the heat capacity of the solid, liquid and vapour, respectively. λ_{effs}, λ_{effl} and λ_{effv} are the effective thermal conductivity of the solid, liquid and vapour, respectively. $H_{\alpha\beta}$ (with α = v, l, s and β = v, l, s) is the effective heat transfer coefficient.

Mass rate of vaporisation: When the volume fraction gradients and the pseudo-convective contributions are negligible, the mass rate of vaporisation (Duval et al., 2004) is given by

$$\dot{m}h_{\mathrm{vl}}=H_{\mathrm{l}}(T_{\mathrm{l}}-T_{\mathrm{sat}})+H_{\mathrm{v}}(T_{\mathrm{v}}-T_{\mathrm{sat}})-H_{\mathrm{s}}(T_{\mathrm{s}}-T_{\mathrm{sat}})\tag{1.31}$$

where $h_{\mathrm{vl}}=h_{\mathrm{vsat}}-h_{\mathrm{lsat}}$ represents the heat of vaporisation.

In the single-phase regions, there is no phase change and the thermal energy equations for the two-temperature model can be expressed as follows:

In the vapour region:

$$\rho_{\mathrm{v}}\varepsilon C_{p\mathrm{v}}\frac{\partial T_{\mathrm{v}}}{\partial t}+\rho_{\mathrm{v}}C_{p\mathrm{v}}\mathbf{u}_{\mathrm{v}}\cdot\nabla T_{\mathrm{v}}=\nabla\cdot\left(\lambda_{\mathrm{effv}}\nabla T_{\mathrm{v}}\right)-H_{\mathrm{vs}}(T_{\mathrm{v}}-T_{\mathrm{s}})\tag{1.32}$$

$$\rho_{\mathrm{s}}(1-\varepsilon)C_{p\mathrm{s}}\frac{\partial T_{\mathrm{s}}}{\partial t}=\nabla\cdot\left(\lambda_{\mathrm{effs}}\nabla T_{\mathrm{s}}\right)-H_{\mathrm{vs}}(T_{\mathrm{s}}-T_{\mathrm{v}})\tag{1.33}$$

In the liquid region:

$$\rho_{\mathrm{l}}\varepsilon C_{p\mathrm{l}}\frac{\partial T_{\mathrm{l}}}{\partial t}+\rho_{\mathrm{l}}C_{p\mathrm{l}}\mathbf{u}_{\mathrm{l}}\cdot\nabla T_{\mathrm{l}}=\nabla\cdot\left(\lambda_{\mathrm{effl}}\nabla T_{\mathrm{l}}\right)-H_{\mathrm{ls}}(T_{\mathrm{l}}-T_{\mathrm{s}})\tag{1.34}$$

$$\rho_s(1-\varepsilon)C_{ps}\frac{\partial T_s}{\partial t} = \nabla\cdot\left(\lambda_{\text{effs}}\nabla T_s\right) - H_{\text{ls}}(T_s - T_l) \tag{1.35}$$

As mentioned before, the use of thermal non-equilibrium models is desirable when the conditions are such that a significant difference can exist between the averaged temperatures associated with each phase. In the case of the two-phase cooling devices, it is widely considered that the local thermal equilibrium assumption represents a valid approximation. In fact, it is even often assumed that the liquid–vapour–wick region is isothermal without much loss of accuracy. This is especially true for the single-component liquid–vapour system because the local thermodynamic equilibrium assumption (again not to be confused with the local thermal equilibrium assumption) implies that the liquid–vapour region temperature in the wick is nearly equal to equilibrium saturation temperature T_{sat}.

1.5.3 Local Thermal Equilibrium Models

When local thermal equilibrium is assumed to be valid, the single phase-wick regions can be described by a single spatial average temperature T and the following approximation can be used: $T_s \approx T_l \approx T$ in the liquid–wick region; $T_s \approx T_l \approx T$ in the vapour–wick region. The macroscopic thermal energy equations in the vapour–wick and liquid–wick regions are given by

$$\left(\rho_v\varepsilon C_{pv} + (1-\varepsilon)\rho_s\varepsilon C_{ps}\right)\frac{\partial T}{\partial t} + \rho_v C_{pv}\mathbf{u}_v\cdot\nabla T = \nabla\cdot\left(\lambda_{\text{effv}}\nabla T\right) \tag{1.36}$$

$$\left(\rho_l\varepsilon C_{pl} + (1-\varepsilon)\rho_s\varepsilon C_{ps}\right)\frac{\partial T}{\partial t} + \rho_l C_{pl}\mathbf{u}_l.\nabla T = \nabla\cdot\left(\lambda_{\text{effl}}\nabla T\right) \tag{1.37}$$

where λ_{effv} and λ_{effl} are the effective thermal conductivity of the vapour–wick and the liquid–wick regions, respectively. The modelling of the liquid–vapour region is discussed in Sections 1.5.4 and 1.5.5.

1.5.4 Local Thermal Equilibrium Models with Explicit Tracking of Macroscopic Interfaces

The governing equations are coupled by internal boundary conditions at moving and irregular interfaces separating the various regions. The locations of theses interfaces are not known *a priori* and depend on the coupled flow and heat and mass transfer in each region. A numerical procedure for such a multi-region problem needs to explicitly track the moving interface, thus calling for complex coordinate mapping or numerical remeshing. For a boiling problem in porous media with two regions (liquid–wick and liquid–vapour–wick regions), Ramesh et al. (1990a,b) use a coordinate transformation so as to deal with an immobilised interface. The transformation serves two purposes: first, it fixes the domain in the transformed space; and, second, it explicitly introduces the non-linearity due to the moving interface into the equations. However, the resulting equations are naturally more complicated than the original equations, leading to serious numerical difficulties.

1.5.5 Multi-Phase Mixture Model

To overcome these difficulties, Wang and Beckermann (1993a,b) proposed, when the thermal equilibrium is valid, a two-phase mixture model in which the liquid and vapour phases are regarded as constituents of a multi-phase mixture. This model has several advantages. First, significantly fewer governing equations are to be solved. Second, the model leads to a single domain formulation (there is no need to distinguish explicitly the various regions: liquid, vapour, liquid–vapour). Hence there is no need to track the interfaces explicitly. The set of conservation equations of heat, mass and momentum for such a mixture is deduced from the macroscopic governing equations of each phase in each region (Wang and Beckermann 1993a,b, 1994, Wang 1997, Najjari and Ben Nasrallah 2002, 2003, 2005). By assuming that radiative heat transfer is negligible the mixture governing equations can be written as follows:

Continuity equation for the two-phase mixture:

$$\varepsilon\frac{\partial\rho}{\partial t}+\nabla\cdot(\rho\mathbf{u})=0 \tag{1.38}$$

where ρ and $\mathbf{u}$ are the density and velocity of the mixture given by

$$\rho\mathbf{u}=\rho_l\mathbf{u}_l+\rho_v\mathbf{u}_v \tag{1.39}$$

$$\rho=\rho_l s+\rho_v(1-s) \tag{1.40}$$

Momentum equation for the two-phase mixture:

$$\mathbf{u}=-\frac{k}{\mu}(\nabla p-\rho_k g) \tag{1.41}$$

where $\rho_k=\rho_l[1-\beta_l(T-T_0)]\lambda_l+\rho_v[1-\beta_v(T-T_{\text{sat}})]\lambda_v$,

$$\mu=\frac{[\rho_l s+\rho_v(1-s)]}{(k_{rl}/\nu_l)+(k_{rv}/\nu_v)};\quad \nu=\frac{\mu}{\rho},\ \nu_1=\frac{\mu_1}{\rho_1};\quad \lambda_1=\frac{\nu}{\nu_1}k_{rl}$$

$$\nabla p=\lambda_l\nabla p_l+(1-\lambda_l)\nabla p_v \tag{1.42}$$

By replacing $\mathbf{u}$ by its expression in Equation (1.43), the following pressure equation is obtained:

$$\varepsilon\frac{\partial\rho}{\partial t}+\frac{k}{\nu}\nabla^2 p-\nabla p\cdot\nabla\left(\frac{k}{\nu}\right)+\nabla\left(\frac{k}{\nu}\rho_k g\right)=0 \tag{1.43}$$

Energy equation: A unified form of the energy conservation equations for solid, liquid and vapour phases is given by the volumetric enthalpy H equation:

$$\Omega\frac{\partial H}{\partial t}+\nabla(\gamma_h uH)=\nabla\left(\frac{\Gamma_h}{\rho}\nabla H\right)+\nabla\left(f(s)\frac{k\Delta\rho h_{\mathrm{vl}}}{\nu_{\mathrm{v}}}g\right) \tag{1.44}$$

where $H=\rho(h-2h_{\mathrm{vsat}})$ and $\rho h=\rho_{\mathrm{l}}sh_{\mathrm{l}}+\rho_{\mathrm{v}}(1-s)h_{\mathrm{v}}$. h_{l} and h_{v} are the enthalpies of the liquid and vapour phases; they are related to temperature by the relationships

$$h_{\mathrm{s}}=c_{\mathrm{s}}T+h_{\mathrm{s}}^{0} \tag{1.45}$$

$$h_{\mathrm{l}}=c_{\mathrm{l}}T \tag{1.46}$$

$$h_{\mathrm{v}}=c_{\mathrm{v}}T+\left[(c_{\mathrm{l}}-c_{\mathrm{v}})T_{\mathrm{sat}}+h_{fg}\right] \tag{1.47}$$

whereas the coefficients Ω, γ_h, Γ_h/ρ and $f(s)$ are given by

$$\Omega=\varepsilon+\rho_{\mathrm{s}}c_{\mathrm{s}}(1-\varepsilon)\frac{\mathrm{d}T}{\mathrm{d}H};\quad \gamma_h=\frac{[s\rho_{\mathrm{l}}+\rho_{\mathrm{v}}(1-s)][h_{\mathrm{vsat}}(1+\lambda_{\mathrm{l}})-h_{\mathrm{lsat}}\lambda_{\mathrm{l}}]}{(2h_{\mathrm{vsat}}-h_{\mathrm{lsat}})s\rho_{\mathrm{l}}+\rho_{\mathrm{v}}(1-s)h_{\mathrm{vsat}}},$$

$$\frac{\Gamma_h}{\rho}=\frac{\rho_{\mathrm{l}}h_{fg}}{\rho_{\mathrm{l}}h_{fg}+(\rho_{\mathrm{l}}-\rho_{\mathrm{v}})h_{\mathrm{vsat}}}D+\lambda_{\mathrm{eff}}\frac{\mathrm{d}T}{\mathrm{d}H},$$

$$D=\frac{\sqrt{\varepsilon k}\sigma}{\mu_{\mathrm{l}}}\frac{k_{\mathrm{rl}}k_{\mathrm{rv}}}{(\nu_{\mathrm{v}}/\nu_{\mathrm{l}})k_{\mathrm{rl}}+k_{\mathrm{rv}}}(-J'(s));\, f(s)=\frac{k_{\mathrm{rl}}k_{\mathrm{rv}}/\nu_{\mathrm{l}}}{k_{\mathrm{rl}}/\nu_{\mathrm{l}}+k_{\mathrm{rv}}/\nu_{\mathrm{v}}};\, \Delta\rho=\rho_{\mathrm{l}}-\rho_{\mathrm{v}}$$

The temperature and liquid saturation can be deduced from the volumetric enthalpy H by

$$T=\begin{cases}\dfrac{H+2\rho_{\mathrm{l}}h_{\mathrm{vsat}}}{\rho_{\mathrm{l}}c_{\mathrm{l}}} & H\le-\rho_{\mathrm{l}}(2h_{\mathrm{vsat}}-h_{\mathrm{lsat}})\\ T_{\mathrm{sat}} & -\rho_{\mathrm{l}}(2h_{\mathrm{vsat}}-h_{\mathrm{lsat}})<H\le-\rho_{\mathrm{v}}h_{\mathrm{vsat}}\\ T_{\mathrm{sat}}+\dfrac{H+\rho_{\mathrm{v}}h_{\mathrm{vsat}}}{\rho_{\mathrm{v}}c_{\mathrm{v}}} & -\rho_{\mathrm{v}}h_{\mathrm{vsat}}<H\end{cases} \tag{1.48}$$

$$s=\begin{cases}1 & H\le-\rho_{\mathrm{l}}(2h_{\mathrm{vsat}}-h_{\mathrm{lsat}})\\ -\dfrac{H+\rho_{\mathrm{v}}h_{\mathrm{vsat}}}{\rho_{\mathrm{l}}h_{fg}+(\rho_{\mathrm{l}}-\rho_{\mathrm{v}})h_{\mathrm{vsat}}} & -\rho_{\mathrm{l}}(2h_{\mathrm{vsat}}-h_{\mathrm{lsat}})<H\le-\rho_{\mathrm{v}}h_{\mathrm{vsat}}\\ 0 & -\rho_{\mathrm{v}}h_{\mathrm{vsat}}<H\end{cases} \tag{1.49}$$

The expression of $\mathrm{d}T/\mathrm{d}H$ is different in each zone:

In the liquid zone: $\left(H \le -\rho_l \left(2h_{\mathrm{vsat}} - h_{\mathrm{lsat}}\right)\right), \dfrac{\mathrm{d}T}{\mathrm{d}H} = \dfrac{1}{\rho_l c_l}$

In the vapour zone: $\left(-\rho_v h_{\mathrm{vsat}} < H\right), \dfrac{\mathrm{d}T}{\mathrm{d}H} = \dfrac{1}{\rho_v c_v}$

In the two-phase zone: $\left(-\rho_l (2h_{\mathrm{vsat}} - h_{\mathrm{lsat}}) < H \le -\rho_v h_{\mathrm{vsat}}\right), \dfrac{\mathrm{d}T}{\mathrm{d}H} = 0$

The velocities of individual phases can also be computed as follows:

$$\rho_l u_l = \lambda_l \rho u + j \tag{1.50}$$

$$\rho_v u_v = (1 - \lambda_l)\rho u - j \tag{1.51}$$

where j is a mass diffusion flux:

$$j = -\rho_l D \nabla(s) + f(s)\frac{k\Delta\rho}{\nu_v} g \tag{1.52}$$

The capabilities of the two-phase mixture model have notably been shown through application to boiling with thermal convection in a porous layer heated from below (Wang et al., 1994, Najjari and Ben Nasrallah 2005), to boiling with thermal mixed convection in an inclined porous layer discretely heated (Najjari and Ben Nasrallah 2003) and to pressure-driven boiling flow adjacent to a vertical heated plate inside a porous medium (Najjari and Ben Nasrallah 2002).

1.5.6 Application to Evaporators

There have been several works based on the continuum approach to porous media devoted to the study of transfers within a subregion of a capillary evaporator, such as the one shown in Figure 1.6b. In Cao and Faghri (1994a,b) solutions were developed assuming a completely liquid-saturated wick. A qualitative discussion of the boiling limit in a capillary structure was provided. A conclusion was that reasonably accurate results can be obtained by a 2D model especially when the vapour velocities are small for certain working fluids such as Freon-11 and ammonia. The vapour pocket within the wick was first considered in Demidov and Yatsenko (1994) and then in Figus et al. (1999). Among other things, these works show that the size of the vapour pocket increases with the applied heat load. The comparison with

PN simulations reported in Figus et al. (1999) indicates that the continuum model used in Figus et al. (1999) and Demidov and Yatsenko (1994) was consistent only with an extremely narrow pore size distribution (so as to have a sharp transition between the liquid zone and the vapour zone). In other terms, the existence of a two-phase zone within the wick was not taken into account. A similar assumption was made in Kaya and Goldak (2006), where the numerical simulations help explain the robustness of LHPs to the boiling limit.

A two-dimensional numerical model based on the two-phase mixture model (see Section 1.5.5) was used in Yan and Ochterbeck (2003) to study the behaviour of a cylindrical CPL evaporator under steady-state operation. In contrast with the simulations reported in Figus et al. (1999), Demidov and Yatsenko (1994) and Kaya and Goldak (2006), the approach leads to significant variations in the saturation within the wick and thus to the existence of an important two-phase liquid–vapour region. The effects of heat load, liquid sub-cooling and effective thermal conductivity of the wick structure on the evaporator performance were studied.

The sharp interface assumption (no two-phase zone within the wick) was again made in a series of similar papers by the same group (Wan et al., 2007, 2009, 2001). As in Figus et al. (1999) or Demidov and Yatsenko (1994), remeshing is needed to track the liquid–vapour interface within the wick. One interesting feature of the model was to use a single set of equations, based on the so-called Brinkman–Darcy–Forchheimer model, within the porous structure and the vapour grooves. Also, the 2D computational domain contains several grooves and this permits us to investigate the evolution of the vapour–liquid interface within the wick over a significantly larger domain than in previous works.

Three-dimensional numerical models have also been proposed (Li and Peterson 2011, Chernysheva and Maydanik 2012). The model presented in Li and Peterson (2011) is, however, restricted to a saturated wick (no vapour pocket) whereas that presented in Chernysheva and Maydanik (2012) is apparently essentially based on the solution of the thermal energy equation without the consideration of partially invaded wick (the wick is either locally saturated or fully dry). Also, convective heat transfer within the wick is ignored. A nice aspect of the model presented in Chernysheva and Maydanik (2012) is the scale of investigation, namely the whole evaporator, as opposed to the small subregion (as shown in Figures 1.6b and 1.9) generally considered in most of the other studies.

It should be pointed out that only steady-state regimes have been numerically simulated in the works mentioned in this section. This is in contrast with the PN simulations, which have been also used to investigate some aspects of transient regimes (Prat 2010, Louriou and Prat 2012).

In summary, this review of the literature on the application of continuum models to the study of heat and mass transfers in capillary evaporators reveals serious and interesting drawbacks in the development of numerical

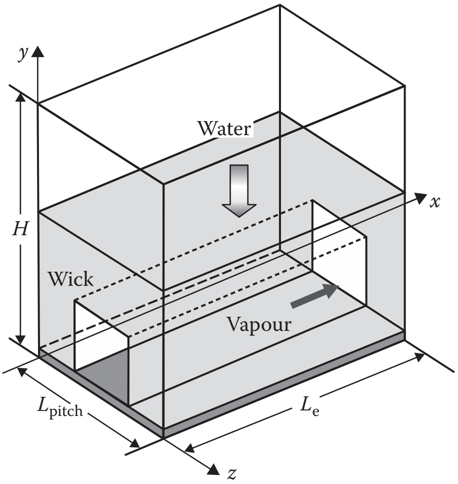

FIGURE 1.9
Three-dimensional computational domain considered in Li and Peterson (2011).

simulations. First, the often-made assumption of sharp interface between the liquid zone and the vapour zone within the wick (no two-phase zone) is questionable. This assumption is neither consistent with the PN simulations nor consistent with the simulations based on the mixture model. Three-dimensional simulations are scarce and based on fairly restrictive assumptions (e.g. saturated wick). Simulation of transient regimes is an open area. This is somewhat surprising owing to the problems often encountered and often not well understood during the start up of LHP. Comprehensive transient 3D models including a proper description of the two-phase zone within the wick with adequate coupling with the transfers in the other parts of the evaporator (vapour grooves, liquid channel, compensation chamber, metallic casing, etc.) are yet to be developed.

1.6 Conclusion

As in other fields involving transport phenomena in porous media, simulations of heat and mass transfers in porous wicks can be performed according to a variety of techniques. These include the direct simulations at the pore scale, PNMs and continuum (Darcy's scale) models. Direct simulations are still little developed in relation with capillary evaporators owing to their very high computational cost. Pore network simulations can be used even in the absence of length scale separation, that is when the traditional continuum approaches are likely to lead to poor results. Furthermore, the relevant transport equations are not solved directly in the pore space and this permits considering systems containing a relatively large number of pores, a distinguishing advantage compared to the direct simulations. Pore network simulations are particularly well adapted to simulate transfers in thin systems (e.g. porous coatings). Also, pore network simulations can be used to evaluate the parameters of the continuum models from 3D images of porous microstructures (notion of digital porous media). Although much less than for the direct simulations, the computational costs for the PN simulations are generally higher than those for simulations based on continuum models.

As a result, there is room for the development of further macroscopic simulations in relation with the study of capillary evaporators or other devices involving porous wicks. Three-dimensional simulations including a proper modelling of two-phase regions are still an interesting open challenge for steady-state as well as for transient regimes.

Nomenclature

C_p	Specific heat capacity (J/(kg-K))	T_{ref}	Reference temperature (K)
$\mathbf{G}$	Gravitational acceleration (m/s^2)	T_{sat}	Saturation temperature (K)
h_{vl}	Latent heat of vaporisation (J/kg)	$\mathbf{u}_i$	Filtration velocity (m/s)
H	Effective heat transfer coefficient (W/m^3/K)	V	Averaging volume (m^3)
k	Permeability (m^2)	$V\beta$	Volume of β-phase (V, m^3)
k_{r}	Relative permeability	**Greek letters**	
M	Molecular mass (kg/mol)	ε	Porosity
$\dot{m}$	Mass flux (kg/m^2/s)	λ	Thermal conductivity (W/(m-K))
n	Unit normal vector	M	Dynamic viscosity, Pl
p	Pressure (Pa)	$(\rho C_p)_i$	Thermal conductivity
p_{c}	Capillary pressure (Pa)	ρ_i	Fluid density (kg/m^3)
p_{ref}	Reference pressure (Pa)	Σ	Surface tension (N/m)
p_{sat}	Saturation pressure (Pa)	**Subscripts and abbreviations**	
Q	Heat flux (W/m^2)	eff	Effective
r	Pore size (m)	ℓ, 1	Liquid
R	Universal gas constant (J/mol/K)	S	Solid
r	Throat radius (m)	Sat	Saturation
s	Saturation	V	Vapour
t	Time (s)		
T	Temperature (K,°C)		
T_{max}	Casing maximum temperature (K)		

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2

Thermally Powered Adsorption Cooling: Recent Trends and Applications

B. B. Saha and I. I. El-Sharkawy

CONTENTS

2.1 Introduction

The quest for a safe and comfortable environment has always been one of the main preoccupations of the sustainability of human life. Accordingly, research aimed at the development of thermally powered adsorption cooling technologies has intensified in the past few decades. These technologies offer double benefits of reductions in energy consumption, peak electrical demand in tandem with adoption of environmentally benign adsorbent/refrigerant pairs without compromising the desired level of comfort conditions. Alternative adsorption cooling technologies are being developed, which can be applied to buildings [1–4]. These systems are relatively simple to construct as they have no major moving parts. In addition, there is only marginal electricity usage, which might be needed for the pumping of heat transfer fluids. The heat source temperature can be as low as 50°C if a multistage regeneration scheme is implemented [5,6]. However, since the system is driven by low-temperature waste heat, the coefficient of performance (COP) of thermally activated adsorption systems is normally poor [7]. A recent study shows that the cooling capacity of the two-stage silica gel/water refrigeration cycle can be improved significantly when a reheat scheme is used [8]. In the first part of this chapter, several advanced thermally activated adsorption cooling cycles are overviewed. Finally, a three-bed dual evaporator–type advanced adsorption cooling cum desalination (AACD) cycle is introduced in which the evaporators work at two different pressure levels and produce cooling effects and potable water from saline or brackish water. The performance results of the AACD cycle are presented.

2.2 Adsorbent/Refrigerant Pairs for Adsorption Cooling and Desalination Applications

2.2.1 Silica Gel/Water Pair

In the 1980s, silica gel/water systems were investigated mainly in Japan. This pair got attention when Sakoda and Suzuki [9] proposed a transient simulation model of a solar-powered adsorption cooling cycle. The authors [10] also analysed solar-driven adsorption refrigeration system. They reported that a solar COP of 0.2 can be achieved by using a solar collector with dimensions of 500 × 500 × 50 mm and 1 kg silica gel. In 1995, Saha et al. [1] analytically investigated the effects of operating conditions on the cooling capacity and COP of a silica gel/water-based adsorption chiller. The same system was analysed experimentally by Boelman et al. [2]. A lumped transient model and a distributed transient model of a two-bed adsorption cooling system were investigated by Chua et al. [11,12]. Saha et al. [13,14] developed and analysed various multi-stage adsorption cooling cycles using silica gel/water as the adsorbent/refrigerant pair.

2.2.2 Zeolite/Water Pair

Nearly four decades back, Tchernev [15] investigated a zeolite/water system for solar air-conditioning and refrigeration applications. Alefeld et al. [16] used the same pair for heat pump and heat storage systems. Tatlıer and Erdem-Senatalar [17] and Tatlıer et al. [18] proposed an arrangement involving zeolite synthesised on metal wire gauzes to eliminate the limitations caused by insufficient heat transfer within the solar collector. Two types of zeolites – namely zeolite 4A and zeolite 13X – were investigated by the authors. Zhang [19] presented a prototype of an adsorption cooling system driven by waste heat from a diesel engine, using a zeolite/water pair. It is reported that the proposed system can achieve a specific cooling power (SCP) and a COP of 25.7 W/kg and 0.38 respectively. Sward et al. [20] presented a model for a thermal wave adsorption heat pump cycle using NaX zeolite/water as the adsorbent/adsorbate pair. The proposed model is used to examine the cycle performance. It is reported that the COP was as high as 1.2 at the heat source, condenser and evaporator temperatures of 120°C, 30°C, and −15°C respectively. A different cycle pattern-periodic reversal forced convective cycle has been studied theoretically by Lai [21] using zeolite 13X/water as the adsorbent/refrigerant pair. Wang et al. [22] presented a novel design of an adsorption air conditioner system that can supply 8°C–12°C chilled water. The system consists of an adsorber, a condenser and an evaporator; all the system components are housed in the same adsorption/desorption chamber. The predicted cooling capacity and COP are 5 kW and 0.25, respectively.

2.2.3 Zeolite Composites/Water Pair

Composite adsorbents made from high conductivity carbon (graphite) and metallic foam with zeolite have been investigated by Guilleminot et al. [23]. The compositions were typically 65% zeolite + 35% metallic foam, and 70% zeolite + 30% graphite. The proposed adsorbents have high thermal conductivity, and hence the system performance was improved. Pons et al. [24] experimentally investigated the performance of a regenerative thermal wave cycle by using an innovative composite adsorbent composed of zeolite and expanded natural graphite. It is reported that the cooling COP improved to 0.9. However, the specific cooling capacity was only 35 W/kg.

2.2.4 Activated Carbon/Methanol Pair

Activated carbon (AC)/ethanol systems have been investigated mainly in Europe since 1980. Pons and Guilleminot [25] studied an AC/methanol system for ice production using renewable energy. Douss and Meunier [26] investigated the possibility of using active carbon of type AC135 along with methanol as a working fluid for refrigeration application. It is reported that for an evaporating temperature lift equal to 25°C, the experimental COP of the intermediate cycle reached 0.5 with regeneration temperature difference (the difference between regeneration and ambient temperatures) equal to 65°C. Critoph [27] studied the possibility of using activated charcoal with different types of refrigerants working at sub-atmospheric and supra-atmospheric pressures. Refrigerants that are sub-atmospheric include methanol, acetonitrite, methyl amine and NO_2. However, ammonia, formaldehyde and SO_2 represent the supra-atmospheric group in the study. Douss and Meunier [28] proposed and analysed a cascading adsorption cycle in which an active carbon/methanol cycle is topped by a zeolite/water cycle. The cycle COP exceeds 1 with a driving heat source of 250°C. Anyanwu et al. [29], Anyanwu and Ezekwe [30] and Wang et al. [31] also conducted studies on various adsorption systems using the same adsorbent/adsorbate pair. El-Sharkawy et al. [32] investigated the isothermal characteristics of methanol on two specimens of ACs: Maxsorb III and Tsurumi activated charcoal. The Dubinin–Radushkevich (D-R) equation is used to correlate the adsorption isotherms of the two assorted adsorbent/adsorbate pairs. Experimental measurements show the superiority of Maxsorb III/methanol pair over the Tsurumi activated charcoal/methanol pair.

2.2.5 Activated Carbon/Ammonia Pair

The possibility of using an AC/ammonia pair for adsorption cooling applications has been investigated by Critoph [27]. He also proposed a novel cycle in which adsorbent heat transfer can be enhanced by forced convection [33]. The adsorption cycle is modelled for AC/ammonia, and results show that a power density of 1–3 kW/kg of adsorbent can be achieved. The heating COP of the adsorption cycle was 1.3 when heat was pumped from 0°C to 50°C. Critoph [34] also used the same pair to build a 10 kW air conditioner. Miles and Shelton [35] experimentally tested a two-bed system using AC/ammonia as the adsorbent/adsorbate pair. An analytical methodology for the prediction of system performance and optimisation has been presented. They reported that through the use of a thermal wave regeneration concept the cycle efficiency can be significantly increased.

2.2.6 Monolithic Carbon/Ammonia Pair

Tamainot-Telto and Critoph [36,37] presented a laboratory prototype of an adsorption refrigeration system using a monolithic carbon/ammonia pair. Critoph and Metcalf [38] suggested a conceptual model of an adsorption refrigeration system based on a plate-type sorption generator using the same adsorbent/adsorbate pair. The results showed that the system could reach a COP of about 0.3 and a specific cooling capacity value above 2 kW/kg of adsorbent.