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NOM DE L'AUTEUR / NAME OF AUTHOR:	RHI, Seok-Ho
ADRESSE POSTALE / MAILING ADDRESS:	707-88 Somerset Street Ottawa, Ontario K2P 0H6
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(AUTEUR)

SIGNATURE

(AUTHOR)



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**An Experimental and Analytical (Simulation) Study on Two-Phase Loop
Thermosyphons; Very Small to Very Large Systems**

by

Seok-Ho Rhi

A thesis submitted to the Faculty of Graduate
and Postdoctoral Studies in partial fulfilment of
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Faculty of Engineering
University of Ottawa



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ABSTRACT

An experimental and analytical (numerical) study was carried out on the fluid flow, heat transfer and instabilities for five (5) small, medium and large scale two-phase loop thermosyphons. The five two-phase closed loop thermosyphons (TLTs) specially designed and constructed for the present study are one small scale loop (SSL: 5.4 mm ID & 1.2 m $\times$ 0.5 m; 150 W), two medium scale loops (MSL I: 0.095 mm ID & 0.423 m in width; 60 W and MSL II: 0.0202 m ID & 3 m $\times$ 2 m; 1,500 W) and two large scale loops (LSL I: 0.0202 m ID & 1.8 m $\times$ 2 m; 7,500 W and LSL II: 0.0202 m ID & 2.208 m $\times$ 10 m; 100,000 W).

Experimentally, various parameters which would affect the operation of a TLT such as the total system temperature difference, saturation temperature of the working fluid in the system, coolant velocity over the condenser section, coolant temperature, the size of the condenser section, the amount of a working fluid in the system, the choice of working fluids, pressure drop, effectiveness and instability were investigated.

There are many difficulties to simulate numerically a TLT system. One difficulty encountered is that some of the correlations for various heat transfer coefficients available has to be used for the conditions beyond the range of variables over which they are established. For an example, even if there are numerous investigations on heat transfer coefficients such as boiling, condensation and

forced convection over finned surfaces, most, if not all, of them are empirical. Because of the great variety of effective parameters and the complexity of the phenomena, purely theoretical analysis cannot provide a general equation. It is now well established that the large difference among the empirical correlations for the supposedly same condition has been reported by a large number of different investigators. This implies that even though a computer simulation may predict the most of the parameters involved, provided that correct empirical correlations are used, to do so, the interior temperature distribution must be verified by experiment.

The second difficulty is the determination of the best method to handle the transition regions between the various flow regimes occurred in the evaporator section. Various parameters such as velocity, void fraction and quality are to be used to determine the transition regions.

The third difficulty is that the pressure drop must be analyzed to converge the entire simulation program. The similar difficulties encountered for the heat transfer coefficients also exist for the equations to be used in the simulation.

In the present study, two simulation methods based on thermal resistance net work, *lumped* and *sectorial*, are presented. In the simplified model called here as the *Lumped method*, the evaporator section is dealt as one lumped boiling section, i.e, one empirical equation for a forced convective boiling. However, this model could not simulate the TLTs adequately because of its simplistic approach to cover all the heat transfer modes of the different flow regimes involved in the system by one equation.

In the *Sectorial method*, all possible phenomena which would occur in the evaporator section due to the two-phase boiling process are considered in detail. Flow regimes (bubbly, slug, churn, annular and annular mist etc.), the flow transition between flow regimes and other two-phase

parameters involved in two-phase flows are carefully analyzed. In the present study, the results of two different simulation methods are compared with experimental results and the limitation of the computer simulation for such two-phase heat transfer systems as the TTLs studied is critically discussed. The study also showed that the simulation results by the Lumped method and by the Sectorial method did not show any partiality for the method used for the simulation.

To complement the simulation study for five different scale of TTLs, an analysis was also made on the instability of the SSL as a separate study.

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NOMENCLATURE

A	area (m^2)
A_f	flow area occupied by liquid phase (m^2)
A_g	flow area occupied by vapour phase (m^2)
a	evaporator thickness (m)
$A_{air, cr}$	cross sectional area for free pass of air through condenser (m^2)
$A_{v, cr}$	cross sectional area for free pass of vapor (m^2)
A_x	cross sectional area (m^2)
b	evaporator width (m)
Bo	boiling number ($q / \dot{m}\Delta h$)
c	wave velocity (m/s)
Cc	contraction coefficient
C_D	drag coefficient
Co	parameter given in Shah's equation in Appendix C
c_p	specific heat (J/kgK)
C_{SF}	a constant on Rohsenow's pool boiling correlation
D	outer diameter of tube (m)

D_b	outer diameter of tube at the bended area (m)
d	inner diameter (m)
D_{fin}	outer diameter of fin, circular fin (m)
D_h	hydraulic diameter (m)
d_{insert}	insert diameter (m)
D^*	critical diameter (m)
E	transfer function, or liquid entrainment
Ep	physical property group
Exp	experiment
F	frictional parameter, factor appearing in Chen correlation
Fr	Froude number (u_{lo}^2/gD)
f	friction factor
G	mass flux (kg/m^2s)
g	acceleration due to gravity (m^2/s)
G_G	mas flux of gas (kg/m^2s)
G_{GM}	mas flux of gas at onset of maximum entrainment (kg/m^2s)
Gr	Grashof number ($(\beta g \Delta t D^3 \rho^2) / \mu^2$)
H	enthalpy (J/kg)
h	heat transfer coefficient ($W/m^2 K$)
h_c	heat transfer coefficient appearing in Shah correlation ($W/m^2 K$)
h_{cond}	condensation heat transfer coefficient ($W/m^2 K$)
$h_{cond,f}$	condensation heat transfer coefficient in horizontal finned surface ($W/m^2 K$)

h_{conv}	convective heat transfer coefficient ($W/m^2 K$)
h_f	saturated liquid enthalpy (J/kg)
h_{fg}	latent heat of evaporation (J/kg)
$I.D.$	inner diameter
j	superficial velocity (m/s)
j_g	superficial velocity of vapor phase (m/s)
j_g^*	dimensionless vapor volumetric flux
j_l	superficial velocity of liquid phase (m/s)
j_l^*	dimensionless liquid volumetric flux
k	thermal conductivity ($W/m K$), wave number (m^{-1})
K	ratio of phase velocities
K_{TPC}	parameter used in Geiger and Rohrer's correlation in Appendix F
l	length (m)
L	length (m)
L	L_c / L_{ev}
ΔL_{ec}	vertical distance between the evaporator and the condenser (m)
LSL	large scale loop thermosyphon
$\dot{m}$	film thickness (m), mass flow rate (kg/s)
$\overline{m}$	mean film thickness (m)
$\hat{m}$	amplitude of wave on the interface (m)
$\dot{m}$	mass flow rate (m)
MSL	medium scale loop thermosyphon

N_{fin}	number of fins
Nu	Nusselt number (hD/k)
OD	outer diameter (m)
Δp_a	accelerational pressure drop (Pa)
Δp_f	frictional pressure drop (Pa)
Δp_h	hydraulic pressure drop (Pa)
P	pressure (Pa)
Pe	Peclet number (GDc_p/k)
Pe^*	modified Peclet number ($qb/r\rho\alpha_l$)
(dp/dy)	pressure gradient (Pa/m)
Pr	Prandtl number ($\mu c_p/k$)
P_r	reduced pressure (P/P_{crit})
Q	heat transfer rate (W)
q	heat flux (W/cm^2)
R	resistance (K/W), radius (m), hold-up (liquid or vapor)
r	radius (m)
Ra	Rayleigh number ($g\mu c_p \beta \gamma^3 \Delta t/k\nu$)
Re	Reynolds number ($\rho uD/\mu$)
R_p	surface roughness (μm)
R_{po}	standard surface roughness (μm)
S	slip ratio, suppression factor used in Chen correlation
S_m	length of the convex condensate film surface measured from the fin tip

<i>SSL</i>	small scale loop thermosyphon
<i>t</i>	temperature (°C)
<i>t_{air}</i>	surrounding air temperature (°C)
<i>t_c</i>	coolant temperature (°C)
<i>t_v</i>	vapor temperature (°C)
<i>Δt_{h-c}</i>	temperature difference between the heater and coolant (°C)
<i>TCT</i>	two-phase closed thermosyphon
<i>TLT</i>	two-phase loop thermosyphon
<i>u</i>	velocity (m/s)
$\overline{u}$	mean velocity (m/s)
<i>u_h</i>	hot fluid flow velocity in the evaporator section (m/s)
<i>u_c</i>	cold fluid flow velocity in the condenser section (m/s)
<i>u_{RJ}</i>	weighted mean drift velocity (m/s)
<i>u_{max}</i>	maximum flow velocity between fins (m/s)
<i>U_T</i>	overall heat transfer coefficient (W/m ² K)
<i>v</i>	liquid or vapour flow velocity in TLT (m/s)
<i>V</i>	volume (m ³)
<i>V_{ev}</i>	evaporator volume (m ³)
<i>V_l</i>	liquid pool volume (m ³)
<i>V</i>	dimensionless working fluid volume, <i>V_l/V_{ev}</i>
<i>V_{WF}</i>	amount of working fluid (ml or gram)
<i>V_{WFR}</i>	volume ratio of charged working fluid to the loop volume (%)

W	mass rate of flow (kg/s)
We	Weber number ($\rho u^2 l / \sigma$)
X	Lockhart-Martinelli parameter
X_M	quality at onset of maximum entrainment
x	vapor quality, x axis
y	y axis

Greek Letter Symbols

α	void fraction
β	volumetric quality
Γ^*	physical property parameter ($\Delta p_{gd} / \Delta p_{lo}$)
γ	inclination angle (<i>degree</i>)
δ	thickness (m), perturbation
ζ	loss coefficient, fin surface profile diameter
ε	effectiveness
η	efficiency
θ	bended angle (<i>degree</i>)
θ_m	angle through which a normal to the fin surface rotates in going from the tip to the end of the convex surface (<i>degree</i>)
λ	wavelength (m)
π_{ng}	normal stress exerted by the gas phase on the interface ($kg/m^2 s$)
π_{nl}	normal stress exerted by the liquid phase on the interface ($kg/m^2 s$)

μ	viscosity (<i>kg/ms</i>)
σ	surface tension (<i>N/m</i>)
ψ	stream function, parameter used in Shah correlation of Appendix C
ϕ	diameter, parameter used in Steiner and Taborek's equation in Appendix C
ϕ_f	flooding angle

Subscribpts

<i>a</i>	acceleration in pressure gradient and pressure drop.
<i>AD</i>	annular flow to drop flow transition
<i>air</i>	surrounding air
<i>ann</i>	annular flow
<i>av</i>	average
<i>b</i>	bulk, boiling, bottom, before in Eq. (4.28), and bended section in Appendix G
<i>BC</i>	single phase flow to bubbly flow transition
<i>bor</i>	before orifice
<i>BS</i>	bubbly flow to slug flow transition
<i>bub</i>	bubbly flow
<i>c</i>	cold section, coolant
<i>cond</i>	condensation, condenser section
<i>conv</i>	convection
<i>crit</i>	critical
<i>cr sect</i>	cross sectional

<i>ct</i>	condenser tube
<i>dis</i>	disturbance
<i>do</i>	dry-out region
<i>drop</i>	drop flow
<i>e</i>	evaporation, evaporator
<i>ef</i>	effective finned surface
<i>eq</i>	equivalent
<i>ev</i>	evaporation, evaporator
<i>ea</i>	average evaporator
<i>eb</i>	average bulk
<i>ew</i>	average evaporator wall
<i>ex</i>	expansion
<i>f</i>	fluid, liquid, <i>f</i> is frictional for pressure drop term, flooding
<i>FC</i>	forced convective boiling
<i>f, tp</i>	two-phase frictional pressure drop
<i>fil</i>	filler
<i>fs</i>	fin sides
<i>ft</i>	fin tips
<i>g</i>	gas, vapor state
<i>gc</i>	gas core
<i>go</i>	vapor only
<i>H</i>	homogeneous model

<i>h</i>	heater, hot for Re_{scb} , hydraulic for pressure drop
<i>hot</i>	heating section
<i>i</i>	inner
<i>ib</i>	incipient boiling condition
<i>l</i>	liquid
<i>lo</i>	liquid only
<i>loop</i>	thermosyphon loop
<i>lp</i>	liquid-phase
<i>m</i>	average value
<i>max</i>	macro
<i>max</i>	maximum
<i>mic</i>	micro
<i>mis</i>	miscellaneous
<i>n</i>	number, row number of LSL II
<i>NB</i>	nucleate boiling
<i>nf</i>	heat flux correction term
<i>NS</i>	translational velocity
<i>o</i>	outer, orifice
<i>p</i>	plug, pool boiling
<i>pl</i>	evaporator plate
<i>r</i>	root
<i>s</i>	slug, slip flow model, separated flow model

<i>SA</i>	slug flow to annular flow transition
<i>sat</i>	saturation
<i>sb</i>	slug bubble
<i>scb</i>	subcooled boiling
<i>sl</i>	single phase liquid
<i>slug</i>	slug flow
<i>sp</i>	single phase flow
<i>T</i>	total
<i>t</i>	tube
<i>tb</i>	taylor bubble
<i>tp</i>	two-phase
<i>tph</i>	two-phase hydraulic
<i>tr</i>	transporting section
<i>u</i>	unflooded
<i>v</i>	vapour
<i>w</i>	wall
<i>WF</i>	working fluid
∞	surrounding air

CHAPTER 1

INTRODUCTION

Two-phase loop thermosyphons (TLT) are the systems in which the flow is driven by thermally generated density gradients so that pumping is not required. These are sometimes called as natural circulation loops, natural convection loops or separate loop heat pipes. These various names are based on the concept that two-phase loop thermosyphons are working in the natural convection mode without any external power supply.

The two-phase loop thermosyphons studied in the present thesis are to be used in MCM (multichip module) cooling system, refrigerator and waste heat recovery systems. These systems as given in Table 1.1 are referred as the small scale loop (SSL - 160 W), the medium scale loops (MSL - 60W) I and MSL II - 1500W) and the large scale loops (LSL I - 7500 W and LSL II - 100000 W) based on heat transfer surface area of TLTs. In industry, two-phase loop thermosyphons have been found in many practical engineering applications. Applications appear in turbine blade cooling [Japikse, 1973], internal combustion engines [Japikse, 1973], electronic element cooling [Polášek, 1994], geothermal energy [Kreitlow, 1978], cold climate applications [Reid, 1975] etc.

An extensive literature survey shows that there is no universally accepted method for predicting the pressure drop and heat transfer in any two-phase flow systems. Therefore, it is

impossible presently to obtain a purely analytical solution for Two-Phase Loop Thermosyphon (TLT) systems.

Many in the past have attempted to simplify the two-phase flow and heat transfer processes in TLT systems, to avoid the complexity and uncertainty of the flow patterns of the two-phase flow involved. In general, the heat transfer of a two-phase flow depends on a great number of different factors: heat flux, pressure, mass flow rate, quality, void fraction, thermal properties of liquid and wall material, surrounding physical geometry and others. This fact, together with a great number of different two-phase flow patterns, hinders the construction of an adequate physical model of the process. Also to simulate a two-phase flow/heat transfer system, such as TLTs, a large number of empirical correlations are needed which present additional complications in the effort to simulate the two-phase system.

The first is that some of the empirical correlations available have to be used for the conditions beyond the range of variables over which they are established. Even if there are numerous investigations on heat transfer coefficients for such as boiling, condensation and forced convection over finned surfaces, most, if not all, of them are empirical. Because of the great variety of affecting parameters and the complexity of the phenomena as mentioned above, purely theoretical analysis could not provide a general equation. A large difference among the empirical correlations for the supposedly the same condition has been reported by a large number of different investigators.

The second is the determination of the best method to handle the transition regions between the various flow regimes in the evaporator section. Various parameters such as velocity, void fraction, quality etc., are needed to determine the transition regions. Accordingly, extending the flow regimes correlations available in the literature, although it may be able to represent the essential

features of the two-phase flow in the system, can not be accepted.

The third is that the pressure drop must be analyzed to converge the entire simulation program. The equations which are used in the calculation have the same difficulties related to the first difficulty discussed above.

In the present thesis, the simulation study on five very different TLT systems are compared with the experimental results and the limitation of the computer simulation for such two-phase heat transfer systems is critically discussed. Therefore, the main objective of the present study is to show that a computer simulation alone could not, at the moment, give any meaningful quantitative results unless it is accompanied with some experimental results for that particular system, implying that no computer simulation code should be developed for a two-phase heat transfer system such as proposed without a benchmark experimental verification.

In addition, an analysis was also made on the instability of the SSL as a separate study to supplement the simulation and experimental investigations of TLTs.

1.1 Two-Phase Loop Thermosyphons

As shown in Fig. 1.1, Two-Phase Loop Thermosyphons (TLT) transfer energy from a heat source to a heat sink by the natural circulation of a working fluid between two regions. As an example, the fluid inside a loop is heated and cooled at certain locations. This creates a temperature difference and consequently a density gradient along the loop. The gravitational force field activates the buoyancy forces that drive the working fluid circulation. Also, a TLT utilizes the difference in the saturation pressure between an evaporator and a condenser to cause the vapor to flow from the hot evaporator to the cold condenser. The geometric configuration must be such that the condensate

can flow back to the bottom of the evaporator by the gravity. Under a steady-state condition, this buoyancy force is equalized by the pressure losses along the pipe. The main advantage of a TLT is that it can transport heat from a source to a sink without a mechanical pump.

The proper design of any engineering system requires a modeling. For example, in the case of a nuclear reactor, it is crucial to be able to predict the flow rate and the core temperature distribution that would occur if an emergency cooling procedure using natural circulation was implemented. One may also want to know what heat exchanger length and wall temperature are required to maintain the absolute temperature inside the loop below a certain level. This thesis is the first detailed study investigating a wide range of TLTs for various applications.

TLTs offer a lot of significant advantages over other types of heat exchangers. A few remarkable advantages can be summarized as;

1. TLTs do not need any external power for the circulation of the working fluid.
2. There are no moving parts in the system. Only the working fluid circulates in the loop.
3. A TLT system can operate indefinitely within the design temperature range, provided that the system is perfectly sealed.
4. Heat can be transferred in either direction.
5. TLT systems are only partially filled with working liquids and hence are lighter than single-phase liquid circulation loops.

1.1.1 Interrelationship of Flow Patterns and Heat Transfer

A two-phase flow in a TLT can be described by several flow patterns, such as bubbles, slugs, or film flows. When the flow pattern changes from one to another, large variations in heat transfer

can occur . Various flow patterns and hence various heat transfer regions are encountered in the evaporator section of a TLT. These flow patterns and heat transfer are interrelated and affect the hydrodynamics of the fluid flow. In the evaporator section of a TLT, the working fluid reaches the entrance of the evaporator, in general, as a liquid in a subcooled state and enters the evaporator section from the header or the transport section. In the entrance region, both the thermal and the momentum boundary layers develop as the working fluid flows along the evaporator section. Initially, both the bulk fluid temperature and the wall temperature remain below that necessary for nucleation. The heat transfer process is single-phase convective heat transfer. At some point along the TLT, nucleation takes place in the subcooled liquid. This is known as subcooled nucleate boiling. The bubbles produced in the superheated wall region condense as they move away into the subcooled liquid. In this region, the wall temperature is higher than the saturation temperature of the fluid but the bulk fluid temperature is still below the saturation temperature. The transition between the subcooled nucleate boiling and the next flow region is the point at which the bulk fluid temperature equals the saturation temperature. Saturated nucleate boiling (or bulk boiling) starts at this stage. As bubbles accumulate and coalesce, the flow patterns change from bubbly to slug to annular flow. With the increase of the quality and the vapor fraction, the resulting higher vapor velocities and higher two-phase heat transfer coefficients cause nucleate boiling to be replaced by surface vaporization. At some critical value of the quality, dryout starts to occur on the tube wall and causes a significant rise in the wall temperature. This increase could lead to tube failure in a constant heat flux boundary condition (i.e., burnout). The region between the dryout point and the transition to single phase vapor is termed the liquid deficient region. In this region, the vapor temperature increases above that of the liquid droplets which are at the saturation state. The heat transfer

coefficients for the dryout region are at the lowest of all of the heat transfer regions involved. The vapor generated in the evaporator reaches the condenser either as saturated vapor or as a wet mixture. The vapor can also be in a superheated state at the inlet to the condenser.

In the condenser, the heat transfer can take place either by dropwise condensation or by film condensation. If the condense is partially flooded, the mode of heat transfer in the flooded region would be single phase liquid forced convection. The condensed liquid then flows by gravity to the evaporator located in the lower part of the loop and thus the cycle is completed.

The two phase flow parameters of a TLT, such as flow pattern, quality, and void fraction must be dealt in the TLT system analysis.

1.1.2 Selection of Working Fluid

The selection of a suitable working fluid for a TLT is one of the most important aspect of the design and manufacturing process, because the basis for the operation of a TLT is the vaporization and condensation of the working fluid. Factors that affect the selection of an appropriate working fluid include the operating temperature range, compatibility with the thermosyphon materials, dielectric properties for use in an electrical field, solubility in water, and thermophysical properties etc. The selection of working fluid for a TLT also depends on several thermal properties of the fluid. The most important thermal properties of the working fluid are the vapor pressure, latent heat of vaporization and surface tension. In addition, the thermal conductivity, viscosity, and vapor densities of the working fluid will also affect the selection of the working fluid for a TLT. The thermal capacity of TLTs is mainly limited by the overall heat transfer coefficient, which depends on the thermophysical properties of the working fluid and the working saturation pressure. Generally any

liquid which can be boiled without decomposition may be used as the working fluid, but in practice there can be many restrictions such as temperatures, a prohibition of flammable fluids, a electrically conductive fluids etc.

1.2 Loop Thermosyphon Simulation and its Parametric Limitations

A loop thermosyphon is a combined system of two-phase flow phenomena and almost all kinds of two-phase flow phenomena may occur in a loop thermosyphon system.

1.2.1 Two-Phase Flow Parameters Involved in Two-Phase Loop Thermosyphons

Two-phase flow is still an area in which a theoretical prediction of flow parameters, in general, is not yet possible. Still less, is the possibility to predict correctly the phenomena in the systems where two-phase flows are involved such as in two-phase thermosyphons. The two-phase parameters described by Hewitt [1978] is shown in Table 1.2.

Even though a large number of researches related to two-phase have been made earlier, there are many situations where the prediction for a two-phase system may carry with it an uncertainty over than 50 %.

With the exception of some of the experimental studies by earlier researchers, few experiments have been either properly designed to provide the basic information (although many investigations have claimed that this has been achieved) or properly instrumented to provide the necessary data. Most experiments were conducted with the purpose of obtaining data for a particular design, paying very little attention to other measurement and information which could be outside of the designer's immediate need. Such data, obtained from a large number of experiments, were often correlated by

computers, thus providing correlations with no particular physical significance. It is helpful to remember that, for an air- water mixture at ambient conditions, as an example, for the same mass flow rate the ratio of the friction pressure gradient for gas to that for liquid is about 800:1. In many branches of engineering science, experience enables engineers to guess a magnitude to within 50%; this is not the case with two-phase flow. That the uncertainty in prediction remains high is due to the large number of variables encountered in two-phase flow as Hewitt [1978] described as the two-phase flow parameters. An extremely vast and systematic test program would be required to provide data for a rationally described empirical correlation, and even our knowledge of some single-phase flow is still essentially empirical. Since the most of two-phase flow correlations are empirical as given in Table 1.3, the combination of governing variables to describe the heat transfer phenomena in a TLT can be astronomical.

With the exception of some of the papers discussed here, most of the published analyses are based on models and formulations often definitively questionable. Often, from these incorrect or questionable formulations, "general correlations" have been introduced by means of high-speed computers.

It is apparent from this brief discussion that a general method for predicting the two-phase flow parameters or for interpreting experimental results is not yet available. As shown in Table 1.2 and 1.3, parameters are still depend on the experimental results. Even though many researchers have attempted to correlate their experimental results into the empirical formula, so far such empirical correlations are under the accuracy of $\pm 20\%$ to 50% or much more. Appendix A (flow pattern map) and Appendix B (heat transfer coefficient, void fraction and pressure drop) present the uncertainty of a few important two-phase parameters. In the later sections, two-phase parameters

will be dealt for as two different cases of lumped (boiling section as one lumped section) and sectorial (depend on flow patterns).

1.2.2 Flow Regimes

Flow pattern maps and flow transition criteria are needed to simulate each flow regimes. Widely accepted flow pattern map is that for vertical flow by Hewitt and Roberts [1969]. However, a flow pattern map is influenced by a number of secondary variables but it is impossible to represent their influence using only two-dimensional plot [Collier and Thome, 1994]. Also flow regime transition criteria are still under development. As shown in Fig. A.1 in Appendix A, flow pattern maps have a large difference with the present experimental results

1.2.3 Heat Transfer Coefficient

(a) Lumped Case

In the lumped case, the evaporator section of a two-phase system is dealt as one lumped boiling section as illustrated in Fig 1.2.

Most of the heat transfer coefficient correlations for lumped case are reported without further explanation of flow state or flow pattern. There are numerous investigations on forced convection boiling heat transfer, most of them are experimental. Because of the great variety of effective parameters and the complexity of the phenomena, theoretical analysis cannot provide a general equation for the boiling heat transfer coefficients for different substances and different conditions. All correlations given in the literature have a relatively large uncertainty and can be used only in

restricted cases as shown in Table B.1 in Appendix B. For this reason measurements on the heat transfer coefficient for different substances and different conditions still appear necessary.

(b) Sectorial Case

In the sectorial case as illustrated in Fig. 1.2, all possible two-phase flow phenomena which would occur in the evaporator section are considered in detail.

For the sectorial case, the correlations of the heat transfer coefficients for each flow regimes are needed, but they are very rarely available. Some correlations are based on annular flow regime and some of them are in nucleate boiling regime. But the problem is that researchers tried to extend their correlation developed for certain specific conditions (geometry, flow condition, working condition and so on) to 'general purpose' with correction factor or constant. Like other two-phase parameters, it is almost impossible for a heat transfer coefficient correlation to satisfy all working conditions. However, the available correlations may be applied to a two-phase flow analysis with some engineering judgement with an estimated uncertainty.

1.2.4 Void Fraction

A major problem in designing for two-phase systems is to predict the effective density of the liquid-vapor mixture for use in obtaining the pressure change within the system. This requires the estimation implicitly or explicitly of the proportion of the cross-section occupied by the vapor, this is so called void fraction.

It is clear that there is a very close relationship between frictional pressure drop and void fraction. The void fraction which is important for the estimation of frictional pressure drop may be

simply related by the following relation [Collier and Thome, 1994].

$$\phi_l \approx \frac{1}{1 - \alpha} \quad (1.1)$$

(a) Lumped Case

For the lumped case, the void fraction correlation is mainly from Lockhart and Martinelli's correlation [1949]. Since Lockhart and Martinelli developed their void fraction correlation, many reseachers reported void fraction correlations. However, their correlations are mainly the correction of the basic void fraction correlations [Isbin et al., 1979].

As shown in Table B.2 in Appendix B, void fraction calculations for a given condition with various correlations show large differences among them. In Table B.2 of Appendix B, it is seen that the uncertainty is in the range of 0.4 ~ 52.2 %.

(b) Sectorial Case

For the sectorial case, the drift flux model of Zuber and Findlay [1965] has been widely used. Usually the drift flux model is recommended for use in bubbly and slug flow regime. But with Ihih's correlation [1982] for annular flow regime has been applied to some two-phase anaysis. The accuracy of this correlation has not been tested. The void fraction correlations for each flow regimes are still very rare.

1.2.5 Pressure Drop

The earliest comprehensive study in this field was concerned with the circulation in boilers and was carried out by Schmidt, Schurig and Beliringer in Germany in the early 1930s. They measured

the wall friction in a vertical pipe by weighing an internal sleeve which formed the internal surface of the pipe. They also measured the static pressure drop; hence, knowing the wall shear forces, they were able to estimate the mixture density.

Martinelli and co-workers [Hewitt, 1994] were the first to produce a systematic basis for correlating two-phase pressure drop. Armand in Russia published in 1946 [given in Ref. Chihholm, 1983] remains one of the most elegant contributions to this field.

Despite the importance of pressure drop in two-phase flow processes, and the consequent extensive research into the topic, there is still no satisfactory method for calculating two-phase pressure drop. The best current methods are cumbersome in structure, heavily dependent on empirically determined coefficients, and have considerable uncertainty. Simpler forms or firmer theoretical bases for predictive methods can only be achieved with a narrowing of the ranges of applicability. The uncertainty range is very wide as shown in Table B.3 to B.7 in Appendix B.

(a) Lumped Case

Pressure drop correlations with acceptable accuracy are available. Even though there are a few correlations which have been used as general correlations, these are still with a large uncertainty. As shown in Tables B.3 to B.7 in Appendix B, although many correlations are tested for a given condition by computational calculation, results show very large errors.

(b) Sectorial Case

For sectorial case, the frictional pressure drop correlations are very rare. Therefore, the uncertainty of correlations can not be discussed in the present study.

1.2.6 Scaling Laws in Two-Phase Systems

The transfer of the experimental results obtained under scaled conditions to the reality in an industrial plant is an old problem of engineering. There must be a sufficient similarity of the main parameters influencing the precedents in the test equipment and in the original set-up. This similarity can be of geometrical, mechanical, static, dynamic, thermal, thermodynamic, electrical and mechanical nature.

In two-phase fluid flow, scaling is much more limited to very narrowly defined physical phenomena than in single phase fluids. For complex and combined phenomena, it can be achieved not by using dimensionless numbers alone, but in addition a detailed mathematical description of the physical problem - usually in the form of a computer program - must be available.

In the absence of prediction methods for two-phase flows, which are reliable enough to predict the behavior of complex systems, the designer is often faced with the need to carry out direct, and often very expensive, experimental work on full-scale system simulations. The cost and time-scale of such experiments could be considerably reduced, if satisfactory and cheap modeling methods could be developed for two-phase flow systems. Scaling methods are, of course, extensively used in single-phase flow, but the situation in two-phase flow is highly unsatisfactory. The difficulty is that the parameters governing the two-phase flow are larger in number and less well understood than in single-phase flow.

Even in a single phase flow scaling is restricted under certain circumstances and this limitation becomes much stronger in a two-phase flow. While in a single phase fluid, we have usually similarity for hydrodynamic and heat transfer processes simultaneously, in two-phase flow each modeling law or scaling number is only valid for a single special phenomenon.

Scaling only with the help of dimensionless numbers is limited in two-phase flow to simple and isolated problems, where the physical phenomenon is a unique function of a few parameters. If there is a reaction between two or more physical occurrences, dimensionless scaling numbers mainly serve for selecting the hydrodynamic and thermodynamic conditions of the modeling tests and we have to separate the influencing parameter in primary ones determining the system and secondary ones which are of minor influence.

In not too complicated cases, scaling to the original circumstances can be done by empirical correlations considering the important physical laws of the process. For many technical applications in future we shall be forced to scale via a computer code which analyses and describes the thermo-hydraulic phenomena as good as possible.

The scaling laws for forced convection single phase flow have been well established and modeling using these criteria has long been an accepted practice. Similar reasons exist for the desire to model a TLT. However, the similarity analysis for a TLT system is much more complicated due to the coupling of the driving force and heat transfer processes. The flow condition can only be determined by the integral effect of the thermo-hydraulic processes along the entire loop. Therefore, in order to develop meaningful similarity criteria, it is necessary to consider these integral effects through some forms of simplified solutions.

The determination of scaling criteria for a single phase flow is achieved through appropriate non-dimensionalization of the well established balance and constitutive equations. However, the same approach for two-phase flow encounters considerable difficulties due to the existing uncertainties in the basic formulation related to balance equations, two-phase flow correlations, and flow regime transition criteria. This fact is well described in Appendix B.

The similarity parameters for a TLT system under a two-phase condition can be obtained from the integral effects of the local two-phase flow balance equations along the entire loop. Even if TLT is working in two-phase state of the working fluid, the TLT meets with single phase regimes such as condenser section and condensate return line. Unlike the investigation boiling section about scaling, TLT system is fully integrated system of heat transfer and fluid flow. Therefore, the scaling of TLT system is almost impossible to scale exactly without a large number of assumptions. So far the fully integrated scaling laws for a TLT system has not been presented in open literature.

(a) Scaling of Heat Transfer

As reported by Ahmad [1973], the general flow boiling heat transfer governed by a large number of variables as given in Eq. (1.2). He suggested 13 dimensionless numbers as the scaling criteria of CHF. Also Ishii and Kataoka [1984], also recommended a similar scaling method for CHF, but based on the transfer coefficients.

$$\text{Flow Boiling} = f(\phi, G, \Delta H, L, D, g, \lambda, \rho_l, \rho_g, \mu_l, \mu_g, C_{p,l}, C_{p,v}, K_l, K_g, \sigma, \gamma, \beta) \quad (1.2)$$

Scaling criteria are shown in Table 1.4.

(b) Scaling of Friction Pressure Drop

Pressure drop scaling of two-phase flow is described by Friedel [1977]. Friedel suggested the pressure drop scaling method with 7 primary dimensionless numbers and two secondary dimensionless numbers, as given in Table 1.4.

(c) Scaling of Two-Phase Integrated System

Ishii and Kataoka [1984] described the scaling method with his drift flux model in two-phase integrated system such as TLTs. The scaling of an integrated system such as a TLT meets single phase scaling criteria and two-phase scaling criteria. It is almost impossible to describe all phenomena to be scaled for a TLT.

Ishii and Kataoka [1984] recommended the loop system scaling criteria with drift flux model. As shown in Table 1.4, six dimensionless numbers for the single phase scaling and seven dimensionless numbers for the two-phase scaling are required with some assumptions.

1.3 Instability Phenomena

Boiling two-phase flow in a TLT is inherently hydrodynamically unstable. Thermal-hydrodynamic instabilities may cause flow excursions or flow oscillations of constant amplitude or diverging amplitude. Flow excursions or flow oscillations are apt to develop by means of buoyancy or compressibility effects. Hydrodynamic instability of a two-phase flow is not desirable since it usually precipitates high-temperature excursion of the heating element, thus causing so-called premature burnout. This section discusses the instabilities which cause the temperature oscillation to occur in the thermosyphon loop and the parametric effects on flow instability.

Flow instabilities in boiling, condensing, and other two-phase flow process are undesirable for several reasons. The sustained flow oscillations generate the mechanical vibration of components and system control problems. Flow oscillations can affect the local heat transfer characteristics, possibly resulting in oscillatory temperatures and may induce a boiling crisis. Flow instability phenomena can be classified on the basis of the following general definitions [Bergles et

al., 1981]. The instability of a system may be either static or dynamic. A steady flow is one in which the system parameters are functions of the space variables only, although they undergo small fluctuations due to turbulence, nucleation, or slug flow. A flow is subject to a static instability if another steady state is not possible in the vicinity of the original state when the flow conditions change by a small step from the original steady-state ones. A static instability can lead either to a different steady-state condition or to a periodic behavior. A flow is subject to a dynamic instability when the inertia and other feedback effects have an essential part in the process. The system behaves like a servomechanism. Two-phase flow instabilities are well classified in Table 1.5 which is compiled by Bergles et al. [1981].

Usually, an instability is compound when several elementary mechanisms interact in the process and cannot be studied separately. It is said to be fundamental (or pure) in the opposite sense.

First of all, the density-wave oscillations among many kinds of TLT instabilities mentioned above, will first be situated within the larger framework of thermo-hydrodynamic stability of two-phase flows. Macroscopic TLT instabilities which, in contrast to local or microscopic instabilities, involve the entire TLT system are characteristically associated with time constants of the order of a few seconds and can essentially be categorized as instabilities of TLTs. In the past, there has been considerable confusion to understand this phenomenon because many experiments were designed without making an effort to clearly define the limits of the unstable part of the system. Density-wave oscillations are perhaps the most frequently observed and analyzed dynamic instabilities. The density-wave oscillations are due to the multiple regenerative feedbacks between the flow rate, the vapor generation rate, and the pressure drop [Yadigaroglu, 1972]. In a TLT loop, inlet flow fluctuations create enthalpy perturbations in the single-phase region. When these reach the boiling region, they

are transformed into void-fraction perturbations that travel with the flow along the TLT, creating a dynamic pressure-drop oscillation in the two-phase region. Therefore, transportation delays in a TLT are of paramount importance for the stability of the system, although inertial effects are also responsible for the generation of phase shifts [Yadigaroglu, 1972]. Other auxiliary phenomena, such as the variation of the saturation enthalpy with pressure, compressibility of the gas phase, thermal non-equilibrium, variable heat transfer, and changes in the relative velocity between the phases, contribute to the fundamental feed back mechanism described above by altering the phase of the various perturbations. The following parameters cause instability [Bergles et al., 1981]:

- Geometry - TLT length, size, inlet and exit restrictions;
- Operating conditions - pressure, inlet subcooling, mass velocity, power input, charging rate of the working fluid, forced or natural convection;
- Boundary conditions - axial heat flux distribution, pressure drop across the TLT.

These parameters lead to compound instability such as flow pattern transition instability, entire flow excursion, acoustic instability, thermal oscillation, condensation oscillation and pressure drop oscillation. In the present investigation of the the causes of the instability which can decrease the TLT performace, are observed and the parametric effects are presented. The parameters observed in the experiment are:

- the effect of heat flux,
- the effect of insert (orifice) size,
- the effect of condenser length,
- the effect of condenser configuration,
- the effect of flow velocity in the condenser section,

- the effect of pressure drop across the loop, and
- the effect of working fluid amount

1.4 Objectives of Present Study

In the present study, a small scale TLT, two medium scale TLTs and two large scale TLTs are studied in detail. The experimental results obtained from various scale TLTs will be compared with simulation results.

The main objective of the study is to obtain a well documented consistent set of experimental data from properly designed TLTs over a wide range of pertinent variables and to compare this data to simulation results. On the analytical side, the problem represents a sensitive application of the equations of conservation of mass, momentum and energy in a two-phase flow system. Since the understanding of the two-phase flow phenomena is rather insufficient, an effort has been made to empiricism in order to provide adequate descriptions for engineering practice. In developing the simulation program, the present study is aimed at testing some of the mathematical models to find out whether, among other things, the existing empirical correlations are satisfactory for the operating characteristics of the system. An additional purpose of the present study is to analyze the system instabilities with the help of the available mathematical tools.

In the present study, the following have been studied:

1. The experimental studies of a small scale TLT (SSL)
 - Effect of Number of Condenser Sections
 - Effect of Different Evaporator Design
 - Effect of Cooling Mode, Forced Convection vs. Free Convection

- Effect of Air Flow Velocity, u_{max}
 - Effects of Number, Arrangement and Length of Condensers
 - Effect of Saturation Temperature, t_{sat}
 - Effect of Quantity of Working Fluid
 - Effect of Inclination of Condenser Section
 - Free Convection Mode
 - Effect of Contact Resistance
 - Effect of Non-Condensable Gas
 - Total and Apparent Mass Flow Rate of the Working Fluid
 - Instability of Working Fluid Flow and Flow Regimes
 - Effect of Insert Orifice
 - Effect of V_{wf} on Δt_{h-c}
 - Cooling Fan Noise Test
 - Transient Behaviour of TLT
2. The experimental studies of two medium scale TLTs (MSL I, II)
- Effect of Quantity of Working Fluid
 - Effect of Saturation Temperature, t_{sat}
 - Effect of V_{wfr} on Δt_{h-c}
 - Effect of Air Flow Velocity, u_c
 - Effect of t_c
3. The experimental studies of two large scale TLTs (LSL I, II)
- Effect of Quantity of Working Fluid

- Effect of Saturation Temperature, t_{sat}
- Effect of V_{WFR} on Δt_{h-c}
- Effect of Air Flow Velocity, u_c
- Effect of t_c
- Instability of Working Fluid Flow
- Transient Behaviour of the TLT

1.5 The Present Candidate's Role and His Contribution in the Study

The author participated in five different TLT projects and worked together with those employed in major research institutions and company. The projects are mainly focused on the experimental study and simulation code development. The entire projects were carried out under the supervision of professor Yung Lee, the advisor for the present candidate. The percentage of the role and contribution of the present candidate is given in Table 1.1.

CHAPTER 2

LITERATURE SURVEY

Loop thermosyphons have been used in a wide range of applications. In the last decade, the amount of research in this field has increased dramatically. Studies have been carried out on one dimensional modeling, detailed numerical calculations, system performance and turbulent transport. Recently, the complex calculations involved in two-phase flow have been made possible with high speed computers. Many researchers have focused on the single phase loop thermosyphons and toroidal loops because of the difficulties of the two-phase flow phenomena and the geometric problems of the system.

A two-phase loop thermosyphon faces a large number of variables. To simulate a two-phase loop thermosyphon, it is necessary to collect the correct information for each variable. Hence, the literature survey is broken down into many divisions such as two-phase loop thermosyphon (experimental and simulation), flow-regime transition, two-phase heat transfer, frictional pressure drop, void fraction, instability and scaling. Two-phase parameters are investigated extensively and therefore, it is not possible to describe all literature on each parameter. In the present study, representative literatures will be described. Also the literatures for the single-phase region, except a brief description, will be excluded in the present literature survey because the studies of single-phase region are found in the usual fluid and heat transfer text books.

2.1 Loop Thermosyphons

2.1.1 Theoretical Study

Keller [1966] studied the stability of a loop thermosyphon geometry operating with a point heat source at the bottom and a point heat sink at the top. He used a one-dimensional model and showed that under certain conditions the loop experiences periodic flow rate oscillations. Welander [1967] pursued the work of Keller [1966] and showed, by carrying out a transient numerical simulation, that such a loop can experience three modes of oscillations: a stable mode where the flow rate is constant; a neutrally stable case where the flow rate oscillates around a mean value that does not change direction; and an unstable case where the flow rate oscillates with time and may change direction periodically. Keller [1966] and Welander [1967] were the first to show that instabilities could result in flow within a rectangular loop thermosyphon as a result of the dynamics of the system. Creveling et al. [1975] carried out a comprehensive study of for a torus located in a vertical plane. They provided basic experimental data and made also direct comparisons with one dimensional calculations for the stability characteristics of the flow in a natural circulation loop. Also they provided a feasible argument to clarify the possibility of instabilities.

2.1.2 Toroidal Thermosyphons

To exemplify the basic approach, a closed toroidal loop of circular geometry is considered as shown in Fig. 2.1. Toroidal closed-loop thermosyphons have been studied extensively, as they are amenable to relatively simple analytical and numerical treatments.

Creveling et al. [1975] studied experimentally and analytically the flow inside a toroidal thermosyphon operating with a constant heat flux condition in the bottom half and a constant wall temperature condition in the top half. Experimental confirmation of the instabilities in a loop

containing water at a pressure of one atmosphere and at moderate temperatures was reported. The experimental apparatus consisted of a glass tube of circular geometry, which was heated from below by ribbon type heating elements and cooled from above by a concentric cooling jacket. Since the boiling on a glass surface is different from that on a metal surface, the study is subjected to question. They showed the existence of the three stability modes reported by Welandar [1967]. They also reported that the friction factor and the Nusselt number in the cooled section were higher than the standard fully developed forced convection values.

Damerell and Schoenhals [1979] used the same geometry used by Creveling et al. [1975], but with various tilt angles of the heated section. They showed that the maximum flow rate was achieved when the heated section was directly below the cooled section, $\alpha = 0$, for a fixed heat flux and a constant wall temperature cooled section. As the angle α was increased, with all other parameters kept constant, the flow rate diminished towards zero when $\alpha = 180^\circ$. For $\alpha > 60^\circ$, the agreement between their one dimensional model, which assumed $Nu = 3.66$ in the cooled section and $f = 16/Re$ throughout the loop, and their experimental observations seemed to be good. For small tilt angles ($\alpha < 60^\circ$), their model predictions overestimated the experimentally observed flow rates. Damerell and Schoenhals [1979] also produced an experimental stability map which showed, as was noticed earlier by Creveling et al. [1975], that for small tilt angles ($\alpha = 0^\circ$), stable flows were confined to a certain range of heat fluxes: stable at low power levels, unstable at intermediate power levels, and stable again at high power levels. For tilt angles greater than $\alpha = 6^\circ$, the flow was always stable, irrespective of the magnitude of the heat flux. Their one dimensional model also showed that, for a given set of operating conditions, multiple steady-state solutions were possible. Experimentally, they were unable to reproduce the results except for the simple case of $\alpha = 0^\circ$.

Greif et al. [1979] and Mertol [1983] carried out transient and stability analyses of the toroidal thermosyphon shown in Fig. 2.1 (with $\alpha = 0^\circ$) for various cooled section configurations: constant wall temperature heat-exchanger; parallel and counterflow heat exchanger. They used a one-dimensional approach, and solved the transient, coupled, momentum and energy equations using a finite difference method. They also carried out stability calculations by imposing small perturbations on the steady-state values of velocities. Greif et al. [1979] reported a good agreement between the results produced by this technique and those obtained in the linearized stability analysis of Creveling et al. [1975].

Mertol et al. [1982] carried out one-dimensional analysis of toroidal thermosyphons a step further by studying the two-dimensional aspect of the flow. Their model allowed radial and axial variations of temperature and radial velocity variations. When the governing equations were non-dimensionalized, the Graetz number emerged as a parameter. The friction factor (f) and the Nusselt number (Nu) were not needed as input data; instead they were now the outputs of the solution. It was shown that both f and Nu reach their fully developed values when the Graetz number is small. They compared the flow rates, obtained from their two dimensional numerical simulations, with the experimental results of Creveling et al. [1975] and reported a good agreement of the comparison. However, they reported that their model failed, however, to predict the streamwise flow reversals observed experimentally by Damerell and Schoenhals [1979].

Ronen and Zvirin [1985] extended the analysis of Mertol et al. [1982] by studying the flow inside toroidal thermosyphons for higher values of the Graetz number. They concluded along the same way as Mertol et al. [1982], re-emphasizing that Nu , f , and Re approach their fully developed values when the Graetz number is small.

Sen et al. [1985b] studied toroidal thermosyphons with known heat fluxes. The originality of their work lies in the treatment of the time-dependent problem where they reduced the one-dimensional momentum and energy equations to an infinite set of ordinary differential equations using the 2π -periodicity of the temperature. It was shown that stable, periodic or even chaotic solutions can be obtained.

Sen et al. [1985c] looked at the steady-state velocity of an inclined toroidal thermosyphon with constant flux heating and constant-temperature cooling. They showed that the point of the maximum velocity did not necessarily occur for a zero inclination angle ($\alpha = 0^\circ$ in Fig. 2.1), as was previously concluded by Damerell and Schoenhals [1979].

Lavine et al. [1986 and 1987] carried out three-dimensional, steady-state, numerical simulations on the toroidal thermosyphon for two Grashof numbers (Gr) and three tilt angles. Their results showed that the flow was indeed three-dimensional, with streamwise flow reversal in some cases. This corroborates the qualitative observations of Damerrell and Schoenhals [1979] and the quantitative experimental observations of Stern et al. [1988]. Secondary cross-stream motions, which were expected to be presented because of pipe curvature, were also obtained. It was found that flow reversals became stronger as Gr was augmented and weaker as the tilt angle was increased. The peripherally averaged quantities, f , Re and Nu , were generally greater than the values obtained from correlations for fully developed flows.

Stern et al. [1988] performed laser-Doppler velocity measurements at the entrance and exit of the heated section of a toroidal loop. Their measurements confirmed quantitatively the three-dimensionality of the flow and established a valuable experimental data base. Their flow visualization studies also showed the presence of streamwise flow reversals.

2.1.3 Single-Phase Closed Loops

One of the earliest studies that employed a one dimensional model to study the transient flow within a natural circulation single phase loop was by Alstad et al [1955]. The governing equations for a complex rectangular geometry were solved numerically and showed good agreement with experimental results. Key features of this study were the assumption of uniform flow rates in the loop at any instant of time and the use of steady state friction factors of the Moody type for non-isothermal, transient operations. Flow instabilities had been observed in loops operating close to the thermodynamic critical point of the fluid. However, Keller [1966] and Welander [1967] were the first to show that instabilities could result in flow within a rectangular natural circulation loop as a result of the dynamics of the system. This result was confirmed by a comprehensive series of experiments conducted by Creveling et al. [1975], and the stability characteristics of a torus located in a vertical plane were reported. The results formed a significant foundation for future experimental work. They obtained a friction factor of the form: $f = m/(Re)^n$ with $m = 151$, $n = 1.17$ for laminar flow and $m = 0.88$, $n = 0.45$ for turbulent flow, and different correlations for heat transfer.

Holman and Boggs [1960] performed experiments to investigate the heat transfer to Freon 12 near the critical state in a closed loop thermosyphon. The thermosyphon system consisted of two vertical branches, a horizontal bottom section with large curvatures at the corners, and a curved top section. A lower portion of one vertical branch was heated by passing an electric current through the tube while the upper portion of the other vertical branch was cooled in a heat exchanger. Holman and Boggs [1960] observed fluctuations in the pressure in the order of 20 to 30 psi in the regions close to the critical state. These fluctuations were accompanied by intense vibrations of the test apparatus. When the cooling water flow rate was increased, the pressure was reduced and the fluctuations subsided. However, when the cooling water flow rate decreased or the power was

increased, the fluctuations became more severe and did not subside until the pressure had risen well above the critical value.

Chato [1962] analyzed the natural circulation loop in vertical multiple channel systems with different heat inputs. He investigated analytically and experimentally a system with three vertical tubes contained between two constant temperature headers. He concluded that such systems could be unstable due to multiple metastable flow patterns from his results. The velocity was estimated using ink injection and it was noted that the transition to turbulence was observed at low Reynolds numbers.

A theoretical and experimental study of the natural circulation in an apparatus with parallel loops was made by Zvirin et al. [1981]. They used the system which was relevant to a pressurized water reactor. It consisted of an electrically heated core and two parallel loops with a once-through heat exchanger. Tests were performed for steady-state and transient conditions with heat removal from either or both heat exchangers. It was shown that the core flow resistance, input heat distribution, and upper plenum geometry yielded three-dimensional flow effects. This contributed to the overall difference (of 30 percent) between the analytical and experimental results. Under certain conditions, oscillations in the flow were observed. These were accompanied by instabilities and flow reversals. The analysis utilized the assumption of a single equivalent loop. Under steady-state conditions, the flow rate was obtained from the solution to the energy equation and the integral of the momentum equation. For the transient behavior, overall energy balances were used to estimate the characteristic time constant of the system. Other studies related to single-phase closed loops are those by Hallinan and Viskanta [1985 and 1986], Sen and Fernández [1985a], Chen [1985], Steimke [1985], Zvirin [1985], Damerell and Schoenhals [1979], Ramos et al. [1985] and Sen et al. [1985c].

A stability analysis of the rectangular loop was carried out by Huang and Zelaya [1987]. Their results indicated that reducing the loop friction and increasing the distance between the heated and cooled sections would improve the system efficiency but could, in turn, result in instabilities, which would then reduce the system efficiency. A more complex configuration is related to the Savannah River power plant, which has nuclear reactors filled with tubular assemblies containing uranium. Under certain conditions after the reactor is shut down, the residual decay heat is removed by natural convection only.

Huang and Zelaya [1988] presented analytical and experimental results for various values of the hydraulic resistance. Their one-dimensional model assumed constant flux in the heated and cooled sections and negligible heat losses from the rest of the loop. Analytically, the form losses in the gate valve were accounted for by using an equivalent length of pipe. The results obtained with their one-dimensional analysis agreed very well with their experiments. It seems that one-dimensional modeling of closed-loop thermosyphons with vertical heat transfer sections, using the just-mentioned assumptions, can be accurate. Analytical and experimental transient results were also presented. It was noted that the relation for the heat transfer was almost identical to the correlation obtained by Holman and Boggs [1960] for their data with Freon 12 as the loop fluid. It was also pointed out that oscillations occurred during the starting period but these were damped out and steady-state operation was obtained.

2.1.4 Open Loop Studies.

Figure 2.2 illustrates an open and closed loop single-phase thermosyphons. One of the major differences of an open loop from a closed loop is the existence of a pressure difference between the inlet and the outlet of the loop. Open loops can be found in geothermal applications.

Reid et al. [1975] investigated the behavior of a concentric tube open loop that is used for protection of permafrost. The device is called an air convection (or air) pile and makes use of a 0.457 m diameter pipe that is used for pilings. A smaller diameter (and shorter) tube was inserted concentric with the larger pipe. In the analysis, a boundary layer flow was assumed and both laminar and turbulent flows were considered. The authors compared the predicted vertical mean temperature profiles, total heat removal rates, and average velocities with the available experimental data and concluded that the model correctly predicted the gross performance of the loop. However, it was noted that the unavailability of detailed experimental data precluded comparisons of the local temperature and velocity profiles.

Kreitlow et al. [1978] investigated heat exchangers used to extract energy from relatively shallow geothermal wells. They examined several interesting problems of buoyancy driven heat transfer in enclosures. They considered thermosyphoning through and around the wellbore casing which was perforated at two or more depths. They developed analytical models for thermosyphoning in a cased well both with and without a heat exchanger installed. Theoretical results were compared with experimental values. Their comparisons showed that the observed energy extraction rates and flow rates through the well casing were possible with thermosyphoning as the only circulation mechanism within the well bore. The model with a heat exchanger installed was parametrically evaluated to illustrate the sensitivity of the model to estimated parameters and the effect of changes in design variables or constraints. An analysis was also made with a heat exchanger with downward flow inside the casing. This model was used to predict the influence of design variables on the behavior of the system and results showed that minor design changes could increase the energy output by over 50 percent.

Torrance [1979] proposed a one dimensional model for an open loop embedded in a highly conductive wall (the Earth). The ambient geothermal gradient which was linear was assumed for the aquifer wall temperature. He investigated several loop geometries. Since the loop was open, flow oscillations were negligible. A critical Rayleigh number associated with onset of motion was derived. The maximum possible exit temperature of the fluid out of the aquifer was found to be 68.8% of the temperature at the bottom of the loop. Higher exit temperatures require longer residence time in the bottom section of the loop.

Torrance and Chan [1980] extended the above study to include the effect of a finite conductivity of the wall with the aquifer being a half torus. Steady state flow and heat transfer were investigated with conduction in the surrounding medium. Critical Rayleigh numbers for onset of motion were determined for various loop aspect ratios.

Certain loop configurations can be considered neither as completely closed nor open. The loops are open; yet there is no pressure difference between the starting and end points as in the case of a closed loop. One such system was discussed by Bau and Torrance [1981a and 1981b]. This distinction was made by the authors only in their second paper. The system consisted of two vertical and one horizontal legs with the sink being a cooled reservoir. In the reference of Bau and Torrance [1981a], they studied the transient and steady behavior of a symmetrically heated loop which include both a water loop and a porous loop filled with water. Their results showed that the system was always unstable when there was no critical Rayleigh number associated with onset of motion. This was also confirmed by a linearized stability analysis. The time delay for the onset of motion was that required for secondary flows to diffuse warm fluid into the vertical legs. They developed a one dimensional analytical model and they solved the governing equations numerically. They also concluded that secondary flows tend to increase the friction factors above that of the conventional

correlations. In the second paper, the effect of asymmetric heating in the loop was discussed. Here, two heaters were used; one in the horizontal segment of the loop and other in one of the vertical legs. With no power to the heater in the vertical leg, the motion was equally probable in both directions. It was concluded, both from the experiment and analysis, that if the heating is in the leg with an upward flow, (in addition to the horizontal segment) the system is always stable. On the other hand, with heating in the leg with downward flow, flow oscillations were observed and results in flow reversal only if the ratio of the power input to the vertical leg to that in the horizontal leg is less than a critical value.

2.1.5 Multidimensional Studies

Presence of multidimensional flow patterns in natural circulation loops was reported by many investigators, e.g., Creveling et al. [1975], Damerell and Schoenhals [1979], Bau and Torrance [1981a and 1981b], Zvirin et al. [1981] and by Stern and Greif [1988].

Mertol et al. [1982] performed a two dimensional analysis of a toroidal loop. The variations in the axial and radial directions were allowed while symmetry about the axis was assumed. The steady state velocity and temperature distributions were obtained by numerical solution of the two-dimensional governing equations. The Graetz number emerges as the governing non-dimensional parameter. New correlations for friction factor and Nusselt numbers were obtained. The results as compared with the experimental data of Creveling et al. [1975] show a good agreement.

Lavine et al. [1986] performed a three-dimensional analysis of a toroidal loop. The governing equations were solved numerically. The results show that the flow was strongly three-dimensional with existence of cross and reverse flows. The flow was assumed to be steady, laminar, and symmetric about the vertical plane containing the circle of radius formed by the pipe centerline.

The conservation equations were solved using a modified version of the TEACH code [Patankar, 1980]. Some of the interesting conclusions were drawn. The variation of friction factor and Nusselt number around the loop was presented. These were different from those obtained from one and two dimensional analyses. The presence of stream wise flow reversals causes the friction factor to decrease contrary to expectations. Flow reversals also decrease buoyancy and have strong influence on the mean axial velocity. This study was followed by Lavine et al. [1987], where the effect of Grashof number is discussed. It was found that three dimensional effects increase with increasing Grashof number.

Velocity measurements using laser-Doppler velocimetry at the entrance and exit to the heating section of a toroidal loop were carried out by Stern et al. [1988]. Results showed three dimensional velocity profiles for four different heat inputs. Flow visualization techniques were also used to observe the complex flow field. In particular, flow recirculation in the longitudinal (axial) direction and circumferential secondary flows were noted, as well as transient motions. The longitudinal velocity and temperature profiles were in the region leaving the heated section and were along the diameter that extended from the inner surface at the radial distance $R - a$ (Fig. 2.1), to the outer surface at $R + a$ (Fig. 2.1). It was concluded that the presence of cross stream secondary flows and stream wise flow reversals emphasized the importance of including three dimensional effects.

2.1.6 Two-Phase Loop Thermosyphons

Japikse [1973] and Greif [1988] reviewed many types of natural circulation loops. Japikse [1973] was the first to present an in-depth review. He differentiated closed thermosyphons from loop thermosyphons, the former being characterized by a flow along the heated (or cooled) wall with a flow in the opposite direction along the core of a tube and the latter by flow being predominantly

in one direction. In later investigations, these systems were called natural circulation loops as opposed to closed thermosyphons. However, Japikse's review concentrated more on closed thermosyphons.

Wallis and Heasley [1961] carried out a mathematical study with three modes of oscillation in a simple TLT. They also attempted to explain the oscillations in physical terms. The qualitative descriptions were supported by their experimental observations of a glass natural circulation loop utilizing pentane as the heat transfer fluid. They considered the loop as a dynamic system of nonlinear time delays, storage elements (capacitors), and resistances. They employed a Lagrangian coordinate model which permitted an exact solution of the conservation equations for a channel subjected to a cyclic variation in inlet flow and then applied small perturbation techniques. The use of Lagrangian variables resulted in the elimination of the problems associated with mixed partial differential equations in time and space by expressing the position of a fluid particle in terms of its residence time in various parts of the loop. Three possible mechanisms that they distinguished were: (a) oscillations caused by changes in riser buoyancy, (b) oscillations excited by the heater section, and (c) oscillations caused by a restriction at the top of the riser. They investigated the first two analytically and enumerated the general conditions for stability although they did not apply the solutions to any particular physical problem.

The first important theoretical and experimental study of oscillations of two-phase loop thermosyphon was performed by Wissler [1956]. In this investigation a TLT was built, using water as the working fluid. Wissler studied the periodic oscillations of flow rate and fluid temperature using a loop that operated in the two phase region. His experimental results showed that stable operation was possible at both low and high power. If the temperature in the riser did not exceed the boiling point, low power stable operation was possible. Also if the entire riser contained a water-

steam mixture, high power stable operation was possible. He also observed that intermediate power resulted in oscillatory modes of operation. He showed that the period of oscillations was inversely proportional to the mean velocity provided that some steam was in the riser at all times. When the presence of the steam was intermittent, the periods were found to be much longer. He also studied analytically that the product of the coefficient of expansion of the fluid, dp/dh , and the vertical height of the riser must exceed a certain value if the flow perturbation was to be sustained. His results showed that the period of oscillations was approximately equal to the residence time of the fluid in the heater and vertical riser. In order to predict the period of oscillation, the basic equations such as the conservation equations and an equation of state, were applied to the loop. This analysis showed the similar trend in oscillations from his graphical presentation, but the periods were less than the experimental results.

McDonald et al. [1977a, 1977b] and Ali and McDonald [1977b] studied a waste heat recovery system that evolved from the concept of combining the physical flexibility of a liquid coupled heat exchanger with the natural circulation characteristics of a two-phase loop thermosyphon. The studies were carried out to recover ventilation waste heat. Experiments were carried out in two separate test loops. One was used only for data acquisition and the second was used for data acquisition and flow visualization [McDonald et al., 1977a]. The performance of the loops were evaluated in terms of a loop conductance, U . They tried to investigate many parameters to see its affect on the performance of TLT heat exchangers. Parameters included the inclination of tubes, temperature differences between evaporator and condenser, different tube lengths, and different tube diameters. They used two different working fluids, R-113 and R-11, for various fills (or charges), orientations, and source and sink temperatures. They showed that the conductance increased with larger angles of rotation of the loop as a result of the decreased flooding in the condenser and dryout

in the evaporator (especially for small fillings). For the small fillings at high inclination angles, the conductance dropped off as a result of the increased transport resistance required to pump the liquid and vapor up the vapor header through a change in elevation. It was also shown that an optimum filling existed that would yield the maximum conductance for a given loop rotation angle and temperature difference.

Ali and McDonald [1977b] developed a computer program that simulated the experimentally observed behavior reported by McDonald et al. [1977a]. Many empirical correlations available in the literature were used at conditions beyond the range of variables over which they were established. They also noted a difficulty in determining the best method to treat the transition regions between the various flow regions in the evaporator. In spite of these difficulties, the computer program predicted values of the surface temperature, heat flux, and overall loop conductance that agreed well with their own experimental results. Using this program, McDonald and Ali [1977b] presented the results for a number of calculations to illustrate the typical behavior of thermosyphon loops. The calculations were performed for unidirectional and bidirectional studies. Experiments and calculations for multiple-loop two-phase thermosyphons were compared to each other by McDonald et al. [1978], for unidirectional operation and by McDonald and Sampath [1980] for bidirectional operation. It was shown for both systems that a careful selection of the coil tube diameter and length, as well as the loop filling, could provide very good loop conductance values over a wide range of temperature differences in multiple-tube, two-phase thermosyphon loops. The operational range of the systems was found to be limited by evaporator tube dryout for large temperature differences and by boiling suppression for small differences. The bidirectional system was found to yield a smaller loop conductance than the corresponding unidirectional system. This was thought primarily due to the flooding in the condenser which thus decreased its performance. In addition, the bidirectional system

was found to be much more sensitive to the loop filling.

Large amounts of work on TLTs were also reported by McDonald and Raza [1984], McDonald et al. [1984], Mathur et al. [1986], Stauder et al. [1986], Mathur et al. [1987] and McDonald et al. [1987].

Ramos et al. [1985] carried out an analysis on two-phase zones of a TLT with heat addition and extraction. The single-phase regions were adiabatic. In the two-phase zones, the properties were assumed to vary linearly from one phase to the other. Several cases were studied and a particular example showed that friction forces were unimportant. This resulted in simplified expressions for the mass flow rate and the maximum velocity. It was concluded that the heat could be transferred from one point to another regardless of its relative position when certain geometric conditions were satisfied. Multiple solutions were found but it was noted that one or both solutions could be unstable.

Ramos et al. [1990] analyzed the steady state flow in a two dimensional square loop where the opposite legs of the loop were held at constant but different temperatures. Several temperature differences and tilts of the loop were considered. The governing equations were solved numerically using a method similar to the SIMPLE algorithm [Patankar, 1980]. The streamlines were presented for the various tilts. Flows with either single or with multiple local cells were reported for various tilts. Striking changes occurred in flow pattern for changes in tilt of only one degree. This study demonstrated the existence of multiple solutions for a two dimensional loop.

It seems that there is no systematic study made on the size effect of TLTs. It must be noted that the size of a TLT has a significant influence on the two-phase flow involved.

2.2 Vertical Upward Two-Phase Flows

2.2.1 Flow Pattern Map and Flow Pattern Transitions

The main difficulties in any two-phase flow modelling arise from the existence of interfaces between phases and discontinuities associated with them. The internal structures of two-phase flow are classified by the flow regimes or flow patterns. Various transfer mechanisms between two-phase mixture and the wall, as well as between two phases, depend on the flow regimes. This leads to the use of regime dependent correlations together with two-phase flow-regime criteria. From this point of view, many works have been undertaken to predict flow regimes. Although nearly all investigators agree that the flow pattern has a major influence on most two-phase (gas-liquid) transport phenomena, the influence of flow pattern is often ignored when predictions are made. This tendency may, in fact, be ascribed to the lack of generally accepted criteria for the transition between the various flow patterns.

The first one that recognized the importance of the flow pattern as a starting point for the calculation of pressure drop, void fraction, heat and mass transfer was Baker [1954]. He published the earliest flow pattern map for horizontal flow. In this map one coordinate is dimensionless while the other is dimensional. A popular map for horizontal flow was proposed by Mandhane et al. [1974]. This map is based on extensive data bank and is given using u_{GS} and u_{LS} as the coordinates. Taitel and Dukler [1976] derived a flow pattern map for horizontal and slightly inclined pipes which is based solely on mathematical modelling. Based on their modelling they were able to plot a two dimensional generalized map for the horizontal case and for pipes with shallow inclination. Weisman et al. [1979], Weisman and Kang [1981], and Crawford et al. [1985] published a set of correlations for horizontal as well as vertical flows.

Weisman et al. [1979] concentrated on the effect of fluid properties on the flow patterns. Their results, however, differ considerably from Taitel and Dukler theory with regard to the intermittent to annular transition in high liquid flow rates and with regard to the stratified-intermittent transition at high viscosity. Also experimental data [Taitel and Dukler, 1987, and Andritsos and Hanratty, 1987] suggest that the effect of viscosity is quite significant.

Spedding and Nguyen [1980] published a set of two dimensional maps for horizontal and inclined pipes, using $(u_s / (gD)^{0.5})^{0.5}$ versus Q_l/Q_g for each specific inclination. Their experiments were performed in 4.54 cm, water-air system and the choice of dimensionless coordinates based primarily on intuition.

Recently, many different flow regime maps have been developed for the prediction of flow patterns under various flow conditions. Lin and Hanratty [1987] have provided a plausible explanation as to why such regimes have failed to be observed by some workers and they tried to explain some degree of disagreement amongst investigators about some of the transitions between flow regimes and the mechanisms which are involved.

Spedding and Spence [1993] have provided experimental results which were obtained from the two-phase air-water co-current flow in a 0.0935 m ID pipe. They attempted to identify the flow regimes by a combination of visual/video observations, pressure loss and holdup data examinations and pressure loss fluctuating characteristics.

Upward two-phase flow is usually much more chaotic and difficult to describe. As a result, flow pattern maps of different researchers, differ considerably from each other. Owing to the practical importance of vertical flow, considerable efforts were devoted to resolve this problem. Many flow patterns maps were proposed, usually based on experimental data alone and plotted on a two dimensional map with various dimensional or dimensionless coordinates. Wallis [1969], Govier

and Aziz [1972] used u_{GS} and u_{LS} as coordinates. Hewitt and Roberts [1969] as shown in Appendix A used the momentum fluxes as coordinates $\rho G u_{GS}^2$ and $\rho_l u_{LS}^2$. Gould et al. [1974] used $u_{GS}(\rho_l/g\sigma)^{1/4}$ and $u_{LS}(\rho_l/g\sigma)^{1/4}$. Griffith and Wallis (1961) used the dimensionless coordinates u_{GS}/u_s and u_s^2/gD . Oshinowo and Charles [1974] used $(u_{GS}/u_s)^{1/2}$ and $u_s^2/(gD(\Lambda)^{0.5})$ where $\Lambda = \mu_s/(S_s\sigma_s^3)^{1/4}$ and μ_s , S_s and σ_s are the specific viscosity, specific gravity and specific surface tension of the liquid, each relative to that of water, respectively.

A model for the case of vertical flow was proposed by Taitel et al. [1980]. He modelled the transition boundaries between bubble, intermittent, and annular flows and proposed the resulting transition boundaries presented in the form of algebraic relations that are sufficiently simple as shown in Appendix A.

Recently, a large number of papers were presented the pattern transition models for vertical flow [Mishinia and Ishii, 1984, McQuillan and Whalley, 1985, and Bilicki and Kestin, 1987]. In these papers additional attempts to model flow pattern transition for vertical flow were made.

Even though the vertical and the horizontal cases attracted the prime attention, the inclined cases were also considered. Weisman and Kang [1981], and Spedding and Nguyen [1980] presented flow patterns maps for the upwards inclined pipes for various degrees of inclination started from horizontal base.

Stanislav et al. [1986a, 1986b], and Mukherjee and Drill [1985] collected data for upwards as well as downwards inclinations and reported usually good comparison with existing flow pattern maps.

Barnea et al. [1980, 1982a, 1982b, 1985a, and 1985b] collected the systematic experimental data for all inclinations from -90° (vertical down) to $+90^\circ$ (vertical up). Also he modified the theory for the transition boundaries and extended to include the effect of small deviations from the horizontal

or vertical cases. Weisman and Kang [1981] proposed correlations for the transition to dispersed bubble and to annular flows which do not depend on the angle of inclination. Even if the accuracy of their correlations is uncertain, the use of the same correlations for all inclinations are probably well justified.

In light of this dilemma, the so called a 'unified' model, that is applicable to all angles of inclination in a natural smooth way was proposed by Barnea [1986, 1987]. In this work the 'old' transition mechanisms were reevaluated and modified and some new ones were presented to yield a complete theory for friction boundaries that are applicable to all angles of inclination. Although the validity of this recent approach is still subject to experimental testing, it seems to handle the dependence of the flow pattern on the inclination angle rather well. The model identifies transition mechanisms for each individual boundary and a logical path for a systematic determination of the flow pattern is suggested. This model incorporates the effect of fluid properties, pipe size and inclination angle in a unified way that is not restricted to a specific range of pipe inclination. This model relies, in many ways, on the previous works of Taitel and Dukier [1976], Taitel et al. [1980] and Barnea et al. [1982a, 1982b], but it provides the needed extensions that make the prediction of the flow pattern free of limitation due to the inclination angle.

2.2.2 Heat Transfer Coefficient Correlations

In an evaporator tube, the heat transfer may take place in various forms, as the working fluid is transformed from a subcooled liquid at the entrance to the evaporator tubes to either a wet or a superheated vapor at its exit. Each of these mechanisms will be reviewed in order in which they would normally be expected to occur in the evaporator and a summary of the literature survey is presented in tabular forms.

(a) Lumped Models (Boiling Heat Transfer Coefficients)

The development of two-phase heat transfer coefficient correlation is an important area. Many researchers are trying to attach the term 'general correlation' to their correlation. But this can not be possible just as one two-phase heat transfer coefficient correlation can not accommodate all heat transfer phenomena. A literature survey reveals that there are numerous correlations available for this particular heat transfer mechanism. These correlations were derived for different sets of physical conditions and for a variety of working fluids. There are numerous correlations for heat transfer coefficient correlation, but the most of these correlations are some modifications of Martinelli's [1949, given in Collier and Thome, 1994], Chen's [1966, given in Collier and Thome, 1994] and single-phase forced convection heat transfer equation such as Dittus and Boelter [1930]. The saturated forced convection is mainly classified into the following two categories:

1. Heat transfer due to convection only, and
2. Heat transfer due to convection and saturated nucleate boiling

Martinelli's suggestion [1949] represents for the first category and Chen's correlation [1966] for the second. As suggested by Martinelli, many of the correlations which have been proposed for the two-phase convective heat transfer are in the form as;

$$\frac{h_p}{h_l} = f_n \left(\frac{1}{X_n} \right) \quad (2.1)$$

The Martinelli's suggestion given as Eq. 2.1 was confirmed by Dengler and Adams's experimental results [1956]. Since then, a number of researchers are attempted to extend Martinelli's suggestion by the following form:

$$\frac{h_{fp}}{h_l} = C_1 \left(Bo + C_2 \left(\frac{1}{X_{fr}} \right)^{n_1} \right)^{n_2} \quad (2.2)$$

Even since Chen [1966] proposed his equation which attempted to describe the sum of the contribution of correction and bubble nucleation, his equation has been modified by many researchers based on their own experimental results. This trend still continues. Although Martinelli's and Chen's correlation were developed over 30-50 years ago, many researchers are still trying to make their experimental results to fit the following format [1966].

$$h_{\varphi} = Fh_{FC} + Sh_{NB} \quad (2.3)$$

Chen's correlation is very popular in flow boiling heat transfer coefficient calculation.

The correlation by Dengler and Addoms [1956] which was developed based on Martinelli's suggestion, had also been modified by many researchers [Appendix C, 1956 - to-date]. However, since Chen's correlation appeared, many researchers have tried to fit Chen's correlations with their experimental data [1966 - todate]. Many later experimental studies such as those of Anderson et al. [1966], Chaddock and Noerager [1966], Jallouk [1974], and Mohr and Runge [1977] have indicated that the Chen's correlation tends to overpredict the effect of nucleation, thereby resulting in large deviations. Further refinements in Chen's correlation were directed at predicting the nucleate boiling component with other pool boiling correlations. Jallouk incorporated the Rohsenow pool boiling correlation [1952], while the Cooper correlation [1984] was used by Gungor and Winterton [1986]. Efforts were also directed at correlating the enhancement factor, F , and the suppression factor, S , in the Chen's correlation to a number of operating and system variables, e.g., Ross [1985] and Bennett

and Chen [1980].

Shah [1976] suggested a graphical correlation which could successfully represent experimental data for water, freons, ammonia, hydrocarbons from 18 independent literature sources. Better than 90% of 780 experimental points were found to be in the $\pm 30\%$ range. However, an attempt to apply this correlation to the data on cryogenic liquids was not successful. .

Bjorge et al. [1982] extended the version of Rohsenow's well known interpolation formula which is supposed to be valid for both nucleate and convective boiling regions. They employed the Colburn's equation [1933] for the single phase convection, the Mikic and Rohsenow's correlation [1969] for the nucleate boiling contribution, and the procedure of Bergles and Rohsenow [1964] for the wall superheat at the incipience of boiling. From a comparison with a limited body of data on water, Bjorge et al. [1982] concluded that their correlation was more accurate than that of Chen because it gave a mean deviation of 15%, as against 17.4% in Chen's correlation. His correlation is only applicable to steam-water flows.

In the high-quality region, the correlation of Traviss et al. [1972] for annular flow condensation was slightly modified after testing it against non-nucleating forced convection boiling data. Bjorge et al.'s final correlation [1982] could correlate the water data derived from eight different sources better than the Chen correlation (13.9 percent against 17.4 percent). The basic heat transfer model of Bjorge et al. was employed by Yilmaz and Westwater [1980] to study the effect of velocity on the nucleate boiling and flow boiling mechanisms of R-113 flowing outside a circular tube in crossflow.

Other related studies can be found in the works of Kandlikar and Thakur [1982], Kandlikar [1983], Khanpara et al. [1986], Gungor and Winterton [1986], Gungor and Winterton [1987] and Kilmenko [1988]. The uncertainty and detailed equations are shown in Appendix B.

(b) Sectorial Models (Boiling Heat Transfer Coefficients based on Flow Patterns)

Heat transfer coefficient based on flow pattern model is very rare. [Kutatelaze, 1961, Mikielewicz, 1973, Bohdal et al., 1999, Wadekar and Kenning, 1990, Kenning and Cooper, 1989, Lavin and Young, 1956, and Rhee and Young, 1974]. But as described above, two-phase forced convection heat transfer coefficients are basically developed in some specific flow patterns. For example, Martinelli's [1949] and Chen's correlations [1966] are based on annular flow. However, their correlation is developed for general purpose from their experimental data. Other researchers have tried to confirm or modify these correlations for their own data obtained from the lumped flow boiling regime.

A few correlations are developed for specific flow patterns. Even though there are few research in developing two-phase heat transfer coefficient based on the specific flow pattern, the basic flow patterns are limited to bubbly flow [Kutatelaze, 1961, Mikielewicz, 1973 and Bohdal et al., 1999], slug [Wadekar and Kenning, 1996 and Kenning and Cooper, 1989], annular flow [Lavin and Young, 1956 and Rhee and Young, 1974], and drop flow [Lavin and Young, 1956]. The few correlations based on the flow pattern model are given in Appendix C.

(c) Condensation Heat Transfer Coefficients

When vapor flow enters into a condenser, vapor is reduced to its saturation temperature, and condensation process starts. The latent energy of the vapor is transferred to the cold surface by the condensation heat transfer process. Mainly, the condensation processes are divided into filmwise condensation and dropwise condensation. In the present study, only film condensation will be considered. In the film condensation process, the surface is blanketed by the film which grows in thickness as it moves down the plate or a tube. A temperature gradient exists in the film, and the film

represents a thermal resistance to heat transfer.

A literature survey on this topic reveals that there are a number of correlations available to predict the heat transfer coefficients for the condensation process. However, most of these correlations are for falling film, i.e. for film condensation outside surface. The film condensation process inside tube is based on the Nusselt's theory [1916] and from this, many condensation heat transfer coefficient correlations have been developed. The condensation heat transfer inside a vertical tube has been extended from the Nusselt-type falling film condensation analysis.

In an early study, Ananiev et al. [1961] proposed to correlate the local heat transfer coefficient for convective condensation. In more recent studies, investigators have proposed correlation techniques for the local condensing heat transfer coefficient that allow for variation of the two-phase flow regime along the flow passage.

Chato [1962] developed a detailed analytical model of the heat transfer for these circumstances. His results implied that this type of flow configuration exists for the inlet vapor with Reynolds numbers (Re) less than 35,000. Predictions of the analytical model and experimental data were found to agree well. His relation for condensation heat transfer coefficient correlation is identical to the relation obtained from the classic Nusselt analysis for a vertical flat plate of height except that the multiplying prefactor has been changed. Flow of this type may be encountered near the end of the condensation process when the void fraction is still large, but the deceleration of the flow has dropped the vapor velocity low enough to produce stratified flow with a thin film of condensate on the upper portion of the tube wall.

At moderate to high inlet vapor velocities, annular flow film condensation is established almost immediately at the inlet and persists over most of the condensation process. A number of investigators have proposed ways of predicting the condensation heat transfer coefficient for annular

flow conditions. Three of the more useful correlations are those developed by Soliman et al. [1968], Traviss et al. [1973] and Shah [1989]. In the annular flow film condensation analysis, it became clear that the shear at the interface and at the tube wall were linked directly to the transport of heat across the liquid film. Based on a purely empirical approach, Shah [1989] has proposed the correlation as a best fit to available convective condensation heat transfer data for round tubes. In a similar fashion to Shah's, Chen et al. [1987] proposed the relation for the local Nusselt number in the presence of vapor shear.

There are a few correlations for condensation heat transfer coefficient for the condensation on the external horizontal pipe with fins [Beatty and Katz, 1948, Smirnov and Lukanov, 1972, Honda et al, 1983, Marto et al., 1990, Sukhatme, 1990, Cavallini, 1999 etc.]. This kind of condensation process in a special geometry is very difficult to describe. Usually, the condensation in the extended surface is estimated with geometrical heat transfer efficiency. But this kind of condensation meets a few other difficulties such as flooding, innundation, etc. Previous research on this type is focused on the extension of the Nusselt's film condensation analysis [1916]. Researchers tried to include the effect of the extended surface into the Nusselt's correlation. Sukhatme [1990] reviewed the condensation heat transfer on extended surface well. Recently, Cavallini et al. [1994a, 1994b, and 1994c], Marto et al. [1983, 1984, 1986, 1987, 1988a, 1988b, and 1990] and Honda et al. [1983, 1984a, 1984b, 1985, 1986, 1987a, 1987b, 1987c, 1987d, 1987e, 1989a, 1989b, and 1991] studied extensively about the condensation heat transfer on integral fin tubes.

Detailed equations for condensation heat transfer coefficient correlations are given in Appendix D.

2.2.3 Void Fraction Correlations

Void fraction is an essential parameter in a two-phase flow because the frictional, accelerational and gravitational pressure drop components depend directly on the void fraction. There are many empirically and theoretically derived models in the literature, but an extensive investigation has revealed that none of the correlations is generally applicable and that the accuracy in extreme conditions is limited. The uncertainty of the void fraction correlations has large differences according to the working fluid and geometries. In fact, the void fraction is affected by a variety of variables including fluid properties, mass flow rate, quality, channel geometry, heat flux, flow pattern etc. Indeed a rather strong influence is also set up by the flow pattern, making it therefore desirable to express the void fraction individually for each flow pattern. Various void fraction models are given in Appendix E.

(a) *Lumped Models*

Even though there are many empirical and theoretical correlations to predict the void fraction as the “General correlation”, void fraction correlations are mainly either homogeneous or velocity slip ratio model. They are shown in Table E.1 in Appendix E.

The homogeneous void fraction is often quite different from the real void fraction and is not recommended. Armand [1946, 1947] has therefore proposed a simple corrections for the homogeneous void fraction model to accommodate the problem which is given as:

$$\alpha = E\alpha_H \quad (2.4)$$

where, E , is either a constant or a weak function of properties. However, this simple form of

correction is of limited applicability.

Levy [1960] has proposed a momentum exchange model which expresses quality as a function of void fraction and density ratio. The equation introduces no empirical parameter but the calculated void fractions are too low except at $p = 14480 \text{ kN/m}^2$.

The approach based on the slip ratio model has been applied to many two-phase systems mainly developed by Lockhart and Martinelli [1949]. Lockhart and Martinelli correlated low-pressure air-water and air-oil void fraction data using the parameter, X , which has been called as the Martinelli parameter.

The void fraction correlation based on the Martinelli parameter has been used by many researchers. Butterworth [1975] explained the simple void fraction relationship with six correlations based on Martinelli parameter as:

$$\alpha = \frac{1}{1 + A \left(\frac{1-x}{x} \right)^p \left(\frac{\rho_l}{\rho_g} \right)^q \left(\frac{\mu_l}{\mu_g} \right)^r} \quad (2.5)$$

Constants p , q , r have been obtained by many researchers by fitting with their experimental results [Zivi, 1964, Barozy, 1965, Thom, 1964 and Turner and Wallac, 1965]. These values are given in Table E.1 in Appendix E.

Martinelli and Nelson's correlation [1948] was developed from a semi-empirical base for horizontal flows without any phase change. However, this correlation which is basically for systems at low pressure with negligible body and inertia forces has been widely used for other situations. An alternative void correlation is developed by Kowalozewski [1964, given in Friedel, 1977]. He started with an analysis of the variables that significantly affect the void fraction in a vertical upflow. From his correlation, the Froude number and the critical pressure ratio are supposed to cover the influence

of hydraulic diameter, mass flow rate and fluid properties on void fraction.

Friedel [1977] attempted to apply the relationship of Kowalczewski [1964] to R-12 data in vertical upflows and arrived at values of the exponents and of the coefficient which further augment the influence of the Froude number but significantly reduce the pressure influence back to that originally proposed by Kowalczewski. The range of validity for these quantities is limited. Löscher et al. [1973] suggested a void correlation for horizontal flow similar to the basic type of Kowalczewski [1964] where the constant coefficient is replaced by an additional critical pressure ratio raised to a negative power.

Beggs and Brill [1973] have shown that the void fraction depends on the orientation of the flow channel unless the flow is homogeneous. They define the effect of the inclination with the following expression; $(1-\alpha)_\theta = (1-\alpha)_H \{1 + C[\sin(1.8\theta) - (1/3)\sin^3(1.8\theta)]\}$. The quantity C was correlated as a function of the mixture Froude number, liquid velocity number, $[V_{Ll}(\rho_l/\rho_g)^{0.25}]$, and $Q_l/(Q_l + Q_g)$. The functional relationship to be used depends on the flow pattern which would exist in horizontal flow at the same gas and liquid flow rates. It is interesting to note that for vertical upward flow it is close to unity. Greskovich [1973] also presented the effect of pipe inclination on the liquid fraction based on his experimental data. He showed that the liquid fraction increases with the increase of the angle at the same Froude number.

Isbin and Biddle [1979] described 12 different correlations for void fraction and compared the calculated results of these 12 correlations under a given condition. Their comparison showed that 12 correlations has large uncertainty in a specific condition. This comparison is shown in Fig. B.1 in Appendix B.

(b) Sectorial Models

The large uncertainty from the correlations based on the homogeneous flow model led to a few new approaches which attempt to model the hydrodynamic feature of two phase flows under different flow patterns.

Armand [1946] correlated his experimental data for the void fraction in air-water two-phase flow system. His correlation is mainly explained by plotting α against β and he observed the void fraction with the range up to 0.72 ($\beta = 0.0$). He recommended the following relationship with $C_A = 0.83$.

$$\alpha = C_A \beta \quad (2.6)$$

Bankoff [1960] suggested a variable density model to predict the void fraction. His approach is mainly based on the homogeneous model with the correction for two-dimensional effects. He also suggested a new treatment for C_A and showed that the value of C_A increases with pressure.

According to Dukler et al. [1964], the method of Hughmark [1962] most successfully correlated the void data. The Hughmark's correlation is based on the variable density model of Bankoff [1960], a hybrid of the completely separated and the homogeneous models. He considered a mixture in which the two phases flow with equal velocity as a suspension of bubbles in the surrounding liquid and where radial gradients exist in the concentration of the bubbles. It is suggested by Hughmark [1962] that Reynolds, Froude and Weber numbers will influence the distribution and thus the flow parameter. In addition, the parameter will be affected by the ratio of liquid to vapor volume rate if there is no slip between the phases.

The evaluation of the dependency of the flow parameter or void on the product of the four numbers raised to powers proved that the Weber number was not significant and that the remaining three variables represented the data with sufficient accuracy.

The Hughmark correlation was originally derived for the prediction of the void fraction in vertical upflows. However, the subsequent testing by Hughmark [1962] and Dukler et al. [1964] have demonstrated also its applicability to horizontal flows.

Zuber and Findlay [1965] developed drift flux model. The drift flux model is mainly fully developed by Wallis [1969] and further improved by Ishii [1975, 1977, and 1982] and Inayatullah [1977] with the basic relationship as given in Eq 2.7. Zuber and Findlay tried to apply the drift flux model for the bubbly and slug flow regimes and Ishii [1982] tried to apply for the annular flow regime. From these studies, it is seen that the drift flux model is valid only in case of $\bar{u}_{gj} > 0.05j$. This implies that the drift flux model is applicable for bubbly and slug flow regimes.

$$(\alpha) = \frac{(\beta)}{C_0 + \bar{u}_{gj} / (\bar{j})} \quad (2.7)$$

Even though the slip ratio model is not strictly applicable solely to bubbly flow, both Smith [1969] and Chisholm [1983] have derived simple and useful expressions for the slip ratio. Smith [1969] assumed a separated flow consisting of a liquid phase and a gaseous phase with a fraction, e , of the liquid entrained as droplets. He assumed that the momentum fluxes of the two separated phases were equal.

Ishii [1977] suggested the value of C_0 and the expression of $\bar{u}_{gj}$ applicable for annular flow regime but he concluded that the drift flux model is not really suitable for annular flows. The annular flow void fraction model was derived for annular flows by Smith [1969]. The main feature of this

model is the dependence of the void fraction upon the liquid entrainment factor, E_f , i.e. the fraction of the total liquid flow which is entrained in the gas-core, in addition to the other flow parameters (e.g. quality, fluid properties etc.). Although, the main drawback is the validity of the assumptions which have been used for the derivation, these correlations can provide reasonably accurate predictions for void fraction. However, assumptions such as equal momentum fluxes between the two phases or minimum kinetic energy or homogeneous flow of gas and the entrained liquid droplets, are questionable.

All equations for void fraction are given in Appendix E.

2.2.4 Pressure Drop Correlations

In any evaporating and condensing heat exchanger system, the working fluid undergoes a phase change and it may exist in either single- or two-phase state. In the following section a literature survey is presented on the correlations available to predict the pressure drop and for single-phase and two-phase regions. Empirical correlations are usually the most reliable prediction methods for two-phase pressure drop within their data base. However, the uncertainty of their prediction becomes high once extrapolated beyond the range of validity as seen in Appendix B. This may be mainly due to the incorrect parametric trends exhibited by most correlations. Phenomenological models, on the other hand, provide generally correct parametric trends, but their predictions are less reliable than those of empirical correlations over specific conditions. This is primarily due to the simplified assumptions introduced in the calculation.

A number of models have been developed for predicting two-phase pressure drop. Some correlations are developed in graphical formats but some correlations are difficult to use due to the extremely complicated approach.

(a) *Lumped Models*

The literature on generalized two-phase frictional pressure drop is voluminous and includes a wide variety of calculation procedures. Designers and analysts, who encounter two-phase flow problems only occasionally, are often in need of some guidance in selection of an appropriate calculation approach.

Two-phase frictional pressure drop correlations are well described by Chisholm [1983] and Friedel [1977]. The correlations for the two-phase pressure drop for diabatic and adiabatic flows are of importance in many technical flow problems. A literature survey on the correlations available is presented in Appendix F for the calculation of two-phase frictional pressure drop, total pressure drop and the void fraction. The methods proposed have varied from semi-theoretical to empirical correlations. Lockhart and Martinelli [1949] developed an empirical correlation for the two-phase frictional pressure drop calculation and this correlation can still be found in many recent research paper as a correlation to calculate frictional pressure drop in two-phase flows. Although the proposed method of Lockhart and Martinelli [1949] has perhaps been the most widely used, where the use of two-phase frictional multipliers whose actual values have been determined empirically, have allowed helpful predictions to be made for certain flow regimes. From literature, there are a large number of correlations which have been used as the generalized correlations. They are;

- the homogeneous model,
- the Lockhart and Martinelli correlations [1949],
- the Baroczy correlation [1966],
- the Thom correlation [1964],
- the Chisholm correlation [1967], and
- the Friedel correlation [1979]

In homogeneous flows, the two phases are assumed to travel at the same velocity and have similar flow characteristics (e.g., velocity profile) as the single-phase flow. Therefore, this model employs the equations derived for single-phase flows with modified fluid properties to account for the presence of the two phases. The application of the homogeneous-flow model is limited to the fully developed bubbly and dispersed-droplet flows, where the two phases travel at approximately the same velocity.

Above correlations with homogeneous flow model have been used widely in two-phase frictional pressure drop analyses regardless to geometries. The Lockhart and Martinelli's correlation can be found in almost all two-phase research papers and sometimes is modified to fit their data. But it is reported that the Lockhart and Martinelli's correlation could be worst for frictional pressure drop correlations. However, the term 'generalized' is used in the sense of not being flow pattern oriented, nor limited to a particular flow pattern.

To describe the frictional pressure drop, first of all, a two-phase friction multiplier must be introduced because the frictional pressure drop has usually been computed by use of two-phase friction multipliers. This approach was pioneered by Martinelli and coworkers [1948] who defined the two-phase friction multipliers, ϕ_g^2 and ϕ_l^2 in terms of pressure drops for the liquid or gas flowing alone; that is

$$\phi_l^2 = \frac{(dp/dz)_{lp}}{(dp/dz)_l} \quad \text{and} \quad \phi_g^2 = \frac{(dp/dz)_{gp}}{(dp/dz)_g} \quad (2.8)$$

Baroczy [1965] has defined a frictional multiplier in a slightly different fashion. He has proposed that the multipliers be defined in terms of the pressure gradient obtained if the entire mass flow was in a single phase,

$$\phi_{lo}^2 = \frac{(dp/dz)_p}{(dp/dz)_{lo}} \quad \text{and} \quad \phi_{go}^2 = \frac{(dp/dz)_p}{(dp/dz)_{go}} \quad (2.9)$$

where $(dp/dz)_{lo}$ and $(dp/dz)_{go}$ are the pressure gradient calculated assuming the entire flow is liquid and vapor respectively. They point out that at the quality of 1.0, the ratio of two-phase to single phase friction will vary with the relative roughness of the channel. More recent work appears to indicate that ϕ^2 is also a function of mass velocity.

Chisholm [1967a] have proposed

$$\phi_l^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \quad (2.10)$$

where

$$X = \sqrt{\frac{(dp/dz)_l}{(dp/dz)_g}} \quad (2.11)$$

and C is a complex function of the mass velocity and density ratio. X is usually known as the Martinelli parameter. Thom [1964] who extended the graphical method of Martinelli and Nelson [1948] to vertical steam-water flow boiling published alternative variations of α , ϕ_{lo} , I_ϕ , a_ϕ , and I_ρ developed from a more extensive body of two-phase pressure drop data for steam-water flow in horizontal and vertical tubes. The resulting variations of I_ϕ , a_ϕ , and I_ρ are plotted in graphical formats. The local two-phase multipliers and the average two-phase multipliers over the heated channel were presented with experimental data covering a much wider range of conditions than those used by Martinelli and Nelson. Both sets of graphs exhibit similar trends, but the Thorn's method

predicts generally smaller two-phase multipliers than the Martinelli-Nelson method. Furthermore, the Thom graphs cover a smaller range of pressures (from 1 to 22 MPa) than the Martinelli-Nelson graph (from 0.1 to 22 MPa).

The simplistic Lockhart-Martinelli-Nelson correlations for two-phase multiplier clearly do not account for effects other than the fluid properties and quality.

Several attempts have been made to develop improved correlations for the two-phase multiplier and void fraction. However, it has been generally found that improved accuracy can be obtained only with more complicated correlation techniques or by developing correlations only for a specific gas-liquid pair over a limited range of conditions.

A subsequent correlation proposed by Baroczy [1965] attempted to account for the effects of mass flux on the two-phase multiplier, and attempted to develop a broadly based correlation technique that could be used for a variety of gas-liquid combinations. This correlation technique is fundamentally the same as the Martinelli-Nelson (separated flow) method with a few modifications. Firstly, the two-phase multiplier, ϕ_{lo}^2 , corresponding to a reference mass flux of $1356.3 \text{ kg/m}^2\text{-s}$ was correlated as a function of quality x and a property index $[(\rho_g/\rho_l)/(\mu/\mu_g)^{0.2}]$. Secondly, the effect of mass flux on the two-phase multiplier was taken into account by providing a separate correlation for the correction factor γ as $[\phi_{lo}^2/((\phi_{lo}^2)_G - 1356.3 \text{ kg/m}^2\text{-s})]$. Baroczy correlation is mainly described by graphical format. Baroczy correlation seems to be better than the Martinelli-Nelson's graph, mainly because of its inclusion of the mass flux correction.

Chisholm [1967a] carried out a number of analytical and experimental studies on two-phase frictional pressure drop. He showed that the two-phase multipliers of the Lockhart-Martinelli graph can be represented by Eq. 2.10. Chisholm and Sutherland [1969] extended the applications by introducing a varying parameter, C . Their correlation is valid only for adiabatic flow. Chisholm

[1970] attempted to extend these equations to flow boiling, and noted that numerical integration is required. To simplify the applications, he derived a correlation that fits well with curves of various graphical methods. In a later study, Chisholm [1973] extended his correlation to both smooth and rough pipes and suggested the generalised correlation, introducing a new coefficient, B , and showed a good agreement between his equation and the Baroczy [1965] curves. The optimized values of 2.365 for the coefficient, B , and 0.25 for the Blasius exponent, n , seem to provide also a good agreement between his correlation and the Martinelli-Nelson curves. For flow boiling, Chisholm [1973] did not indicate any effect of heat flux on the frictional pressure drop. However, a systematic overprediction was seen and this could be caused by the effect of heating on the two-phase multiplier.

More recently, Friedel [1979] used a data base of 25,000 points to develop the frictional pressure drop correlation for predicting the two-phase multiplier, ϕ_{lo}^2 , for vertical upward and horizontal flows in round tubes. Friedel [1979] assessed 14 correlations for experimental data, and found that none of them was applicable for a wide range of conditions. Based on an expanded data base, he subsequently derived a generalised correlation that is independent of flow pattern [Friedel, 1979]. His correlation provides generally satisfactory predictions over a wide range of conditions. However, the uncertainty of its predictions remains high: the standard deviation is 26%, 32% and 25% for upward, horizontal and downward flow of a single-component mixture, respectively, and close to 52%, 43% and 54%, respectively, for two-component flow. This high uncertainty seems to be primarily due to the large scatter among the data. Friedel correlation provides the best agreement with their experimental data. Friedel correlation has been recommended for use when $(\mu_l/\mu_g) < 1000$.

For large-diameter tubes, Chenoweth and Martin [1955] found that a good fit to available data was provided by correlating ϕ_{lo}^2 as a function of the inverse of the property index used by Baroczy [1965] as described above and the liquid volume fraction, equal to $1-\beta$. This correlation is usually

recommended for tubes having inside diameters greater than 5 cm.

The uncertainty range and equations of pressure drop correlations are given in Appendix B and F

(b) Sectorial Models

It is ambiguous to describe the flow pattern model because the previously mentioned frictional pressure drop correlations are basically for some specific flow pattern. Few studies of frictional pressure drop in some specific flow patterns are found in the literature.

Levy [1960] derived a steam-slip equation from the momentum equations by assuming different pressure gradients for the two phases in a heated channel. His equations shows that steam slip is affected by the heat generation rate, Q , and latent heat of vaporization, H_{fg} , in addition to other parameters for pressure-drop calculations such as void fraction and quality. Since the pressure gradient for each phase cannot be obtained from experimental data, Levy derived a momentum-exchange model to calculate the steam slip.

Bankoff [1960] imposed a power-law distribution for both the local velocity and the local void fraction. Compared against the experimental data of void fraction, this model is valid for most conditions and is better than the Levy momentum-exchange model. However, it tends to underpredict the two-phase pressure drop.

The homogeneous model is generally valid for the bubbly flow and dispersed flow regimes. For a uniform homogeneous mixture, a frictional pressure drop correlation is proposed by Owens [1961] who claimed that his correlation could be used to predict the pressure drop for the range of 60 to 381 psi and the exit quality of up to 57 %. Govier and Aziz [1965] also suggested a frictional pressure drop correlation based on the homogeneous model.

Dukler et al. [1964] derived a set of equations based on a similarity analysis. They proposed that the same flow characteristics between two systems are exhibited in both single- and two-phase flow, provided that the systems are dynamically similar (i.e., having the same Reynolds and Euler numbers). When compared against the available experimental data, the equation for a slip flow is better than that for a no-slip flow. However, the agreement with the data is poor, especially for steam-water flows.

There has been only a limited number of investigations on the pressure drop in slug flows. Govier et al. [1957] proposed a pressure drop correlation for vertical upward slug flow regime. They suggested that the overall pressure drop could be attributed to liquid phase pressure drop, while the gas pressure drop is neglected. Hoogendorn and Buitelaar [1961] presented a correlation for the bubbly flow, slug flow in terms of two-phase friction factor. Their correlation defined the two-phase friction factor as the ratio against the single phase liquid friction factor.

The earliest empirical approach of annular flow pressure drop was carried out by Armand [1946] who obtained an empirical relationship between friction and liquid fraction. According to the liquid fraction, he suggested different constants, C , and exponents, n , for $\phi_l^2 = C/\alpha^n$.

Levy [1952] derived equations analytically for the two-phase pressure drop for two-phase and two-component annular flows in horizontal pipes. The gravity effect is neglected. He assumed that the phases travel separately with vapour in the core and liquid between the vapor and channel wall. The momentum equation of laminar flow was used for viscous flow, and the one-seventh power profile was employed for turbulent flow. No liquid entrainment in the vapor or vapor bubbles in the liquid film was modeled. At the liquid-vapour interface, the two phases were assumed to have the same velocity. The equations covered the viscous-liquid/viscous-gas, the viscous-liquid/turbulent-gas, and the turbulent-liquid/turbulent-gas conditions. The two-phase pressure drop was presented

in terms of the two-phase multiplier for liquid-only flow, ϕ_l^2 , which was expressed for viscous-liquid/viscous-gas flow. The Levy model follows the same trend as the Lockhart-Martinelli curves, but predicts generally a larger two-phase multiplier.

Wallis [1970] introduced a simple annular-flow model where the interfacial friction factor is based on a force balance of the vapour core and the assumption of negligible shear-stress variation in the liquid film. With the available experimental data base, Wallis correlated the interfacial friction factor. The pressure gradient is presented in terms of a dimensionless pressure-drop function and from the force-balance equation of the vapor core. Based on the analysis of liquid film, the pressure gradient is also presented for turbulent and laminar films. Both the pressure drop and void fraction can be obtained by solving these equations. The pressure drops predicted with this model were compared against low gas-flow rate data. A good agreement was observed; in particular this model appeared to capture the transition from a laminar to turbulent film accurately despite its simplicity. At high vapor-flow rates, however, the model overpredicted the pressure-drop measurements. This could be caused by neglect of the entrainment process that led to overpredictions of the film thickness and pressure drop. Wallis [1970] discussed other effects that may have a strong impact on the prediction of two-phase pressure drop in annular flows. Compared against the predictions of a more complex model of Levy [1960], this model was less accurate but provided reasonable predictions.

Chen and Spedding [1981] suggested simple pressure drop correlations for annular and annular-mist flows. An adaptation of the Lockhart-Martincili analysis is presented to which analytical and empirical methods were applied, which resulted in solutions for the holdup and pressure loss for the case of separated flow. For annular flow, the analysis is compatible with the pressure loss data for large diameter pipes, where liquid surface effects are negligible. The analysis overpredicts holdup because interfacial shear has a pronounced effect on this parameter. However, an

empirical modification enabled holdup to be predicted in annular flows. Further developments of the Lockhart-Martinelli derivation enabled a correlation to be developed for annular flow in which the effect of geometry is included. Such a correlation was reported to be useful in predicting the frictional pressure loss in steam-water systems.

The other studies for the pressure drop in annular two-phase flows can also be found in those of Ishii and Kataoka [1983], Yao and Sylvester [1987] etc.

Dukler et al. [1986, 1987 and 1995] carried out the experimental and analytical studies in annular two-phase flow regime. A series of comprehensive experiments were performed on two gas-liquid systems in upward annular flow. Through core and overall momentum balances, a possible new mechanism for momentum transfer in annular flow has been identified. Measurements are consistent with the idea that depositing droplets displace the film locally, decelerate at the wall, and therefore transfer negligible momentum to the liquid film.

Droplet interchange is important in this respect which has been demonstrated to account for as much as 20% of the pressure gradient [Lopes and Dukler, 1986]. The objective of Lopes and Dukler is to determine how momentum is transferred from depositing droplets, and whether deposition should be considered a simple addition to the interfacial friction. To achieve this, experiments have been performed using two gas-liquid systems in a 5.08-cm-ID tube at the pressures slightly above atmospheric, for the gas velocities ranging from film-churning well into the regime of disturbance waves. A momentum balance analysis and physical arguments are then used to develop a new insight into the deposition process.

In the past the influence of the flow pattern on the friction pressure drop in vertical upflows has been considered to be rather insufficient or its importance is overshadowed by the accuracy of the measurement. In an extensive study of pressure drop data has substantiated that in upward flows,

the various flow patterns in the vicinity of the duct wall rather than in the turbulent core have a profound influence [Beattie, 1973]. Moreover his analysis revealed that two phase friction pressure gradients can be calculated using the familiar single phase pressure drop relationships with a friction coefficient that depends on a form of Reynolds number, the value of which is appropriately defined for the particular type of two phase flow close to the wall.

Beattie and Walley [1982] suggested a simple frictional pressure drop correlation based on the mixing length theory. Their correlation was compared with 11 other correlations for HTFS (Heat Transfer and Fluid Flow Service) data and there was a good agreements.

(c) *Two Phase Flow Pressure Losses due to Area Changes*

Single phase or two-phase flow systems often encounter flow area change and flow direction changes. The pressure drop of the flow area and the direction change in single-phase flows has been studied extensively and most fluid mechanics textbook include well known correlations. But in two-phase flows, the pressure drop correlation for the area change is still under development.

The pressure loss due to local disturbances can be caused, as in single-phase flow, by either a direction change (ie., bend), a area change (ie., sudden expansion or contraction) or a flow obstruction (ie., valve or orifice). The pressure loss is produced by the flow separation, the turbulent wake formation and the acceleration effect, and usually expressed in terms of the two-phase multiplier similar to the frictional pressure in a straight pipe.

Although there have been a number of experimental and analytical investigations [Mendler, 1961, Janssen, 1966, Geiger, 1966 and Fitzsimons (given by Chisholm, 1983)] of vapor-liquid pressure drop across area changes, there appeared to be no fully coherent picture available in the literature. The review of Collier and Thome states that a slip-flow model can describe expansion

pressure drop but the homogeneous model describe better for the pressure drop across contractions. [Collier and Thome, 1994]. Fitzsimmons [given by Chisholm, 1983] on the other hand, reported that both his expansion and contraction data could be correlated by the homogeneous model. Since bends are common fixtures in pipe systems, the pressure losses produced by them have been extensively investigated. Chisholm and Sutherland [1969] proposed a two-phase multiplier for bends using the Lockhart and Martinelli parameter. An improved correlation was provided by introducing the single phase loss coefficient [Chisholm, 1980]. Pressure losses in both sudden expansion and sudden contraction are caused by the separation and the acceleration of the phases, and the change of interphase shear stress due to variation of phase velocities. Lottes [1961] compared four different correlations and recommended the Romie correlation for sudden expansion. In the study of the sudden contraction, Geiger and Rohrer [1966] found that the pressure drop was influenced by the flow regime in the re-expansion. They suggested the use of the homogeneous two phase multiplier for the prediction. Chisholm [1969 and 1981] presented a similar equation for both sudden contraction and sudden expansion, but with different coefficients for each case. For the sudden contraction, the equation is identical to the homogeneous two-phase multiplier relation.

Janssen [1966] observed a homogeneous mixture after an orifice. He reported that his data for well separated-expansion-contraction combinations were best described by a model which assumed slip flows for the upstream and downstream, and mixing at the vena contracta. However, for short inserts, where the expansion and contraction are close together, Janssen reported that the assumption of slip flow everywhere yielded best agreement with experimental data. Correlations for orifices have been proposed by various authors [Sher, 1965, Chisholm, 1981 and Sadatomi, 1988] and for valves [Simpson et al., 1985]; however, some of them are subjected to specific assumptions or conditions. For other blockages such as valves or plates, Chisholm provided a general equation

with different coefficient values to adapt for various cases [1969, 1981]. Simpson et al. [1985] carried out experimental studies of the pressure drop in bends and valves, and their experimental data were compared with the calculations using other correlations. They showed that Chisholm's correlation overpredicted by over 100% and others in the range of ± 10 - 20%. Beattie [1973] extended the mixing length theory for estimating the two-phase multipliers for orifices. The proposed equations agreed with his experimental results.

The equations for area change pressure drop correlations are given in Appendix F.

2.2.5 Two-Phase Flow Scaling

The scaling laws are important for designing, performing and analyzing simulations and experiments using a scale model. They are particularly important for studying phenomena which can not be easily observed in a prototype model such as a two-phase loop thermosyphon system. In view of the inherent difficulties associated with the full scale testing of such systems, it is necessary to use scale models and simulation experiments. As described in previous section, the scaling of a TLT may encounter all of single and two-phase flow phenomena. Energy, momentum, mass balances of single and two-phase flows must be considered.

In two-phase flows, scaling is much more limited to very narrowly defined physical phenomena than in single phase fluids. For complex and combined phenomena, it can be achieved not by using dimensionless numbers alone but in addition a detailed mathematical description of the physical problem. The scaling of the thermodynamic data of the modeling fluid plays an important role. However, without a theoretical description of the phenomena nondimensional numbers for scaling two-phase flows must be handled with great care.

A similarity analysis for a TLT system is much more complicated due to the coupling of the driving force and heat transfer processes and systems which encounter the both flow features of single phase and two-phase flows. Therefore, the flow condition can only be determined by the integral effect of the thermohydraulic processes along the entire loop. In order to develop meaningful similarity criteria, it is necessary to consider these integral effects through some forms of simplified solutions. This similarity can be of geometrical, mechanical, static, dynamic, thermal, thermodynamic, electrical etc.

Several scaling techniques have been developed to establish similarity relations between prototype and scale models. There are some publications regarding to the two-phase flow scaling but the publication for the scaling of TLT systems is very rare. The similarity analysis for a two-phase flow system has been carried out by Dukler [1964], Zuber and Findlay [1965], Ishii and Zuber [1970], Armand [1973], Kutateladze [1974], Ishii and Jones [1977] and Friedel [1977] and others. The linear scaling method is the simplest technique in which all length ratios are preserved. While there are some single-phase flow situations where linear scaling is adequate, its use results in thermal and flow distribution distortions which are generally considered unacceptable for two-phase flow conditions, [Ishii et al. 1970 and 1977]. Volumetric scaling techniques are frequently used for thermohydraulic experiments [Zuber and Findlay, 1965]. Model facilities using this approach are usually full-length, while area, volume and power are reduced proportionally. Both linear and volumetric scaling techniques are developed based on single-phase flow conservation equations.

Armand [1973] suggested the criteria of flow boiling heat transfer with 13 dimensionless numbers as shown in Table 1.4. Even though his criteria is for CHF scaling, his fluid to fluid scaling approach is the starting point emphasizing the important of scaling laws in two-phase flow systems.

Dukler et al. [1964] and Friedel et al [1977] introduced two-phase frictional pressure drop scaling. Dukler et al. [1964] developed the frictional pressure drop correlation based on the dynamic similarity requirements they explored two cases in detail and tested with selected data. But their similarity approach was to develop the parameters for two-phase flows corresponding to the Euler and Reynolds numbers for single phase flows. Their analysis began with assumption that the two-phase flow in a single phase system is dynamically similar and the Reynolds number and Euler number for the two-phase system must be equal. Friedel [1977] introduced his scaling approach of two-phase frictional pressure drop with seven primary dimensionless numbers and two secondary dimensionless numbers. He reported that there is a limitation in fluid to fluid scaling in the frictional pressure drop scaling and that basically the scaling method which is based on single phase flows is restricted to simple geometries and for limited conditions.

Mayinger [1976] extensively reviewed the two-phase scaling and concluded that even in single phase flow, scaling is restricted under certain circumstances. This limitation becomes much stronger in two-phase flows. While in a single phase fluid we have usually a similarity for hydrodynamic and heat transfer processes simultaneously, in two-phase flows each modeling law or scaling number is only valid for a single specific phenomenon.

Scaling with dimensionless numbers only is limited in two-phase flows to simple and isolated problems where the physical phenomenon is a unique function of a few parameters. If there is a reaction between two or more physical occurrences, dimensionless scaling numbers mainly serve for selecting the hydrodynamic and thermodynamic conditions of the modeling tests and we have to separate the primary influencing parameters from the secondary ones which are of minor influence.

Recently, extensive investigations have been undertaken by Ishii et al. [1970, 1977, 1984 and 1987] to develop a similarity criteria for the natural circulation loops under single-phase and two-

phase flow conditions. The determination of scaling criteria for single phase flows is achieved through appropriate non-dimensionalization of the well established balance and constitutive equations. However, the same approach for two-phase flows encounter considerable difficulties due to the existing uncertainties in the basic formulation related to balance equations, two-phase flow correlations, and flow regime transition criteria.

The available methods to develop similarity criteria for two-phase flow systems have been reviewed by Ishii and Jones [1977]. Ishii and Kataoka [1984] utilized the results based on the local conservation equations and ones based on the perturbation method. The extension of the similarity analysis to a natural circulation system is achieved by considering the scaling criteria from a small perturbation method and the steady state solution.

2.3 Closure and Brief Summary

Due to the importance of the subject, the literature available is extensive and thorough showing the experimental and theoretical efforts of many investigators. However, even with this vast quantity of information, only experimental studies of two phase loops could be found. Also in the area of the instability problem, many researchers have avoided the difficulties of studying the two-phase region. Therefore, it is very rare to find any literature on the instability in whole TLT systems. The performance of thermosyphon loops has been extensively investigated because its comprehensive understanding is very important for many practical applications, such as in the design of solar thermosyphon collectors, reactor emergency cooling systems, and other industrial equipment.

It is very complicated to analyze the loop thermosyphons theoretically due to the complex coupling between the momentum and energy transport. It can be argued that the disagreement between analyses and experiments arises mainly from the adoption of the conventional friction factor

correlation [Creveling et al., 1975 and Bau and Torrance, 1981a], convective heat transfer correlations [Bau and Torrance, 1981a; Zvirin et al., 1981], and with the use of one-dimensional approximation analysis [Mertol et al., 1982 and Zvirin et al., 1981].

To simplify the analysis, the loops studied were always in some simple configuration such as two vertical branches with a point heat source at the bottom and a point heat sink at the top [Keller, 1966, Welander, 1967, Zvirin and Greif, 1979], circular toroidal loop heated uniformly over the lower half and cooled over the upper half through a constant wall temperature or an annular heat exchanger [Creveling et al., 1975, Damerell and Schoenhals, 1979, Greif et al., 1979, Mertol et al., 1981; Mertol et al., 1983, Mertol and Greif, 1982 and Lavine et al., 1986], U-shaped open loops heated symmetrically or asymmetrically [Torrance, 1979, Bau and Torrance, 1981a and Bau and Torrance, 1981b], closed loops with heating leg and cooling jacket [Zvirin et al., 1981, 1982], or a rectangular thermosyphon loop with bottom heating [Chen, 1985].

Various geometrical effects were investigated from one loop geometry to others with different operating conditions [Creveling et al., 1975, Bau and Torrance, 1981a, Zvirin et al., 1981]. In addition, it has been frequently stated that the friction factor as well as the convective heat transfer correlations in a natural circulation loop are different from the conventional ones for straight pipes. Hence, a modification based on experimental data is needed so that the one-dimensional approximations could be valid. This was attributed to the multidimensional or secondary flow effect and natural convection, which were analytically verified recently by Lavine et al. [1986] using a three-dimensional analysis.

In general, the overall loop friction comes from two sources. The wall friction within the straight pipe sections, and the eddy or form losses due to the bends, fittings in the loop, or the loop curvature. Multidimensional effects near the curved part of the loop may be propagated and

enhanced. This affects the loop friction as the ratio of the diameter to the length becomes large or the radius of the curvature of the loop becomes small. Thus, the conventional friction factor correlation for one-dimensional straight pipe flow needs to be modified if it is to be applied to a curved loop. Modification to the convective heat transfer correlations in the loop is also required because of the combined natural and forced convection effects.

As seen in proceeding sections, the studies on two-phase flow parameters such as heat transfer coefficient, void fraction, pressure drop and scaling etc. have been carried out by many researchers. Although there are a large number of empirical and semi-empirical correlations on parameters, only a few correlation methods are acceptable under the reasonable uncertainty as shown in Appendix B. These acceptable correlation methods, however, are also unacceptable in some geometrical and circumstantial conditions. Still many researchers in the field have tried to apply many popular correlation methods for various two-phase flow conditions. The use of empirical and semi-empirical correlation methods in two-phase flows needs a careful engineering judgement on the accuracy of results.

- iii. Cooling method: Forced convection using a fan and allowable noise level of the fan be less than 55 dBA.
- iv. Working fluid: Dielectric, non-flammable, and environmentally friendly working fluid.

The test assembly was manufactured with the following specifications to accommodate the practical requirement.

(a) Heat capacity

- Heat flux : up to 10 W/cm²
- The maximum surface temperature of the evaporator section: 75°C

(b) Dimensions of the system

- 35 mm x 35 mm x 9.5 mm (thickness); Material - copper: Three different designs of the evaporator section were used.
- Four different designs of the condenser section were used: 20 mm x 35 mm x 200 mm (height) (the exposed area of the condenser section can be varied); material - tube; brass and fins; copper.
- Transport-section: 1.2 m in both directions; material of the tubes - brass

(c) Cooling method for the condenser-section

- Forced convection, using a fan.
- The maximum allowable noise level: 55 dBA.

(d) Characteristics of the transport-section

- Outside diameter of the upward tube - 7.9 mm and the returning tube - 6.4 mm.
- Bend both of these tubes in the middle section to 40° (0° being the horizontal).

- Provisions were also made for flow visualization.

(e) Heat generation sections

- Rated up to 150 W.

Heating element: BeO film

Heater outer surface: $50 \text{ mm} \times 50 \text{ mm} = 2,500 \text{ mm}^2$

Heat generation area: $38 \text{ mm} \times 38 \text{ mm} = 1,444 \text{ mm}^2$

Test evaporator section: $35 \times 35 \text{ mm} = 1,225 \text{ mm}^2$

Five thermocouples are imbedded in the heater.

(f) The contact material (filler) between the Heat Generation Section and the Evaporator of TLT

- Chomerics T500-series.

(g) Working fluids

- Acetone, ethanol, FC72, FC87, R11, R113 and water.

3.1.2. Experimental Apparatus for SSL

The experimental apparatus, illustrated in Figs. 3.1 and 3.2, and Plate 3.1, consists of the main TLT assembly, the cooling system for the condenser section, the heat generation section and the charging system.

3.1.2.1 Main SSL Assembly

Evaporator Section

Three different designs were adopted for the evaporator sections of the system (i.e., Evaporators A, B and C). For evaporator A, a square cavity ($31 \text{ mm} \times 31 \text{ mm} \times 7.5 \text{ mm}$) was cut

out on the back side (non-working surface) of the evaporator as shown in Fig. 3.3 and Plate 3.2. Two holes ($\phi = 7.9$ mm and 6.4 mm, respectively) were drilled through the upper side (35 mm x 9.5 mm) into the cavity for the transport tubes which connect the evaporator section with the condenser section. A square thin copper plate (thickness 2 mm) was silver soldered on the cavity to create a box. Two brass tubes (7.9 x 0.4 mm and 6.4 x 0.4 mm, respectively) were silver soldered into the holes. The tube with $\phi = 6.4$ mm has an extension tube having an outside diameter of 4.7 mm and inside diameter of 3.7 mm for the condensate return as illustrated in Fig. 3.1. The evaporator was tested under pressure and vacuum for leaks.

For the two other evaporators, B and C, only the internal cavity structure and physical dimension are different. All other parts of the system remained the same. The details of Evaporators B and C are illustrated in Fig. 3.4 and Fig. 3.5, respectively. The reason for the choice of the particular configurations is given below.

Both the analytical and experimental results with Evaporator A indicated that the heat transfer wall of the evaporator was not strong enough to maintain its flatness under the changing internal cavity pressure during the experiment. Consequently, the value of the contact thermal resistance between the heater and the evaporator together with the filler varied with the pressure. For this reason, Evaporator B was designed which has three ribs on the evaporator cavity wall. The ribs incidentally increased the area of the boiling surface.

Evaporator C was designed to simplify the manufacturing process of the evaporator and at the same time to maintain the same advantages of Evaporator B. The cavity in the evaporator was created with eight 5.6mm drilled holes. Four in the top and the other four in the side as illustrated in Fig. 3.5.

All dimensions of the three evaporators are given in the Figs 3.3., 3.4 and 3.5.

Condenser Section

Condenser #1 consists of 66 copper fins (20 mm x 35 mm x 0.51 mm and pitch - 3 mm) with two tubes in line as shown in Plate 3.3. These finned tubes were soft soldered into two parallel horizontal headers made of copper cylinders ($\phi = 22.2 \text{ mm} \times 1.6 \text{ mm} \times 57.2 \text{ mm}$). Swagelok fittings were used for connecting the header with the transport tubing for vapor and condensate. Condenser #1 was used with Evaporator A only.

Condenser #2 consists of 51 copper fins (20 mm x 35 mm x 0.51 mm and pitch - 4 mm) with three tubes in line holes as shown in Plate 3.4. As was the case for the condenser #1, these finned tubes were soft soldered into two parallel horizontal headers made of copper cylinders ($\phi = 22.2 \text{ mm} \times 1.6 \text{ mm} \times 57.2 \text{ mm}$). Condenser #2 was used with Evaporators B and C.

The design of the third condenser, Condenser #3 as shown in Fig. 3.6 is essentially the same as that of Condenser #2. The main difference is in the number of the fins attached (63 fins with the pitch of 3.2 mm). Condenser #3 was used only with Evaporator C. In one particular series of experiment, Condensers #2 and #3 were used together in line configuration. Characteristics of Condensers are given in Table 3.2 below.

Transport Lines

The evaporator and condenser are connected with two transport lines as shown in Fig. 3.2 and Plates 3.5 and 3.6. One is for the two-phase flow from the evaporator to the condenser, and the other for the condensate return from the condenser to the evaporator. The line for the two-phase flow was a brass tube (7.9 x 0.4 mm) with a 40° bend to the horizontal plane at the mid section. The inclined section is about 650 mm in length. A glass tube ($\phi = 12.7 \times 2.5 \text{ mm}$ and 310 mm in length)

is provided in the upper vertical section of this line. The line for the condensate return is also made of the brass tube ($\phi = 6.4 \times 0.4$ mm) with a 40° bend to the horizontal plane at the mid section. The inclined section is about 800 mm in length. A glass tube (9.5×2.1 mm and 600 mm in length) is provided in the inclined section of this line. The glass tubes are for the visualization study of the flow regimes as well as for the fluctuation of the flow in the tubes.

Loop Seal

A temporary closure of the loop thermosyphon was made with a NUPRO vacuum valve (B4HK-TW, SWAGELOK Co.) as shown in Plate 3.6. This valve made it feasible to charge working fluids as they were required. The valve was connected to the upper header through a brass tube ($\phi = 6.4 \times 0.4$ mm).

Temperature Measurement

As shown in Fig. 3.1, several types of the thermocouples were used to measure the temperature distribution over the length of the loop thermosyphon.

- No. 1 - ready-made insulated thermocouples TT-T-30-36 [Teflon insulation, T-type, 30 AWG (0.25 mm O.D.), 914 mm in length];
- No. 2 - OMEGA subminiature thermocouple probes (quick disconnect style) KMQSS-040-G-6 (K-type, 1 mm O.D. 304 SS sheath, 152 mm in length, grounded junction);
- No. 3 - OMEGA subminiature thermocouple probes (transition junction style) KMTSS-062-G-6 (K-type, 1.58 mm O.D. 304 SS sheath, 152 mm in length, grounded junction);

- No. 4 - ready-made insulated thermocouples GG-K-24-36 (glass braid insulation, K-type, 24 AWG (0.5 mm O.D.), 914 mm in length).

Thermocouples, type No. 1, are placed inside the heat transfer front wall, at each of the four corners and one in the center as shown in Fig. 3.7. Due to the uneven temperature distribution over the surface of the heater, and the different dimensions of the heater and the evaporator we used, only the readings of the temperature of the central thermocouple used as the reference temperature of the heater. Thermocouples, type No. 2, are placed inside the evaporator walls as follows: one each at the front and the back, two in the centre and two in the left corner walls. Thermocouples, type No. 3, are placed as follows: at the right corners of the inside of the evaporator front and back walls, one inside the evaporator cavity (used to measure the saturation temperature) and three in the flow channel (one at the entrance of the condenser (vapor temperature), one at the exit of the condenser (condensate temperature), the third at the entrance of the evaporator (subcooling condensate temperature)). All thermocouples were placed using SWAGELOK connectors or T-connectors. Thermocouples, Type No. 4, were placed on the edges of the second fins from the bottom ends of the condenser (one each for the two condensers), on the external surface of the tube connecting the upper header and on the seal valve for the detection of the presence of non-Condensable gases in the system.

All the thermocouples (T and K-types) used were calibrated *in situ* and connected through two rotary selector switches (OSW3-16 type, 16 positions and 2 poles) to two digital thermometers (DP-460, resolution 0.1 °C, OMEGA Co.), respectively.

Orifice Inserts

To determine the effect of a change in the pressure drop in the flow channel of the TLT on the flow instability and temperature fluctuation, three orifice inserts (cylindrical : $d = 5.5$ mm and 6 mm in height, $d_{\text{orifice}} = 0.51, 0.71$ and 1.2 mm, Plate 3.7) were used. These inserts are placed inside the condensate line using a Tee connector situated near the evaporator section (140 mm from the upper edge of the evaporator).

Flow Visualization

For visualization of the two-phase flow regimes in the transport line just above the evaporator section, a photographic technique called as shadowgraph was used. This method has been widely used for two-phase flow visualization.

3.1.2.2 Cooling System in the Condenser Section

A fan was used for cooling as illustrated in Plate 3.8. As can be seen in the picture, the finned parts of the TLT's were installed into the channel (290 mm long and in cross section - rectangular with the dimensions: 201.5×35 mm) and made from transparent acrylic sheets (with 6 mm thickness). The finned tubes were installed at the distance of 255 mm from the channel entrance. This front channel was installed perpendicularly to the desk board with a thickness of 18 mm into the opening cut. From the back of the desk board, another channel made from the same material was bonded to the first. The back channel was also rectangular in cross section but was tapered from the size of the front channel (201.5×35 mm), to the size of the fan (80×80 mm). The long sides (

bottom and top) of the back channel were 278 mm long and created at an angle of 24° in the vertical plane between each other. A rectangular shape flange was mounted on the end of the back channel.

A cooling fan (Archer 273-242: 80 mm 3" cooling fan, 32 cfm, 120 VAC, 60 Hz, 12 W, $1.51 \cdot 10^{-2} \text{ m}^3/\text{s}$ at 3100 rpm, the rated sound level - 38 dB) was screwed to another (same size) flange. The two flanges were connected one to another with a hinge.

A gasket of soft rubber was installed between two flanges to prevent the air leakage. Twelve grooves were made on the internal surface of the channels for installing plate inserts (one from the front and one from the back). Thus, the heat transfer area of the finned tubes was decreased by means of changing the length (height) of the channels. Each plate insert from the bottom created a channel with the dimensions and the flow velocity as shown in Table 3.3.

The hot-wire probe (Wallac Inc. - Type Ni-125 ANE) was installed at the bottom side of the front channel before the finned tubes at a distance of 180 mm from the air entrance as shown in Plate 3.1. The measuring head of the probe was in the center of the channel cross section (the same position as in the cross section area decreased by inserts). The hot-wire probe was connected to a thermo-anemometer (Wallac Inc. - Type GGA22F), which had the ability to measure not only air velocity but also air temperature. The probe was calibrated for velocity and temperature, using a hot-wire calibrator set, and a NBS [National Bureau of Standards (USA)] standard mercury in glass thermometer. The cooling fan was connected to a variable autotransformer with an output of 0 - 140 V and up to 10 A.

For the case where two condensers were placed in the flow channel, a short extension section with the same cross sectional dimensions and 51 mm in length was added to the front of the first condenser to minimize the air flow disturbance.

Both forced and free convection modes were used for cooling of the condenser. For the free convection mode, the condensers were made inclined with respect to the horizontal plane from angles of 90° (vertical position) to 8° .

3.1.2.3 Heat Generation Section

A special flat plate heater (Type A), designed and manufactured for the simulation of the MCM operation, was provided by the Electronics and Telecommunications Research Institute (ETRI) of Korea.

The Type A heater, shown in Fig. 3.7, consisted of a heating element (film) made from BeO on a ceramic sheet plate. It contained metal as an electrical conductor from one side, a copper box, and was covered by a thin layer of electrical insulating material. The heating element film of 38×38 mm was installed into the copper box ($50 \text{ mm} \times 50 \text{ mm} \times 10.2 \text{ mm}$) with an opening of $35 \text{ mm} \times 35 \text{ mm} \times 8 \text{ mm}$. The heating layer faces into the box and the opening was covered with a copper plate. The copper wall facing the heating layer was 2.2 mm thick.

The power to the heater was supplied by a variable autotransformer (type 116B, input - 120 V, output - 0 - 140 V and up to 10 A). The five T-type thermocouples, made from the thermocouple wire (OMEGA TT-T-30), were installed in the grooves on the inside surface of the heat transferring copper wall as illustrated in Fig. 3.7. The grooves were 1.4 mm wide and 1.2 mm deep. Thus, the five thermocouples were located across from the heat transferring copper wall in the center and at the four corners of the heating element. The heater, with the heat transferring copper wall, was installed face up into the cut opening of the nylon box ($70 \text{ mm} \times 70 \text{ mm} \times 10.2 \text{ mm}$). It contained 7 holes in the back; five for the thermocouples and two for the power wires.

The heat transferring copper wall of the heater was covered with a CHO-Therm T500 insulator-elastomer [Chomerics Inc. minimum breakdown voltage - 4,000 V, volume resistivity - 10×10^{14} Ohm-cm, thermal conductivity - 2.7 W/(m K), thickness - 0.25 mm and operating temperature range - -60°C ~ $+200^{\circ}\text{C}$]. The filler used for the tight connection between the heater and the evaporator of the TLT was made from silicone and boron nitride and is bonded to the copper surface. This composition allowed the properties of electrical insulation and heat conduction to be combined. The two nylon boxes, one with the evaporation section and the other with the heater, were screwed together in such way that the heater, with the filler, was placed onto the evaporation section. Six screws located along the perimeter of the nylon boxes allowed for a good contact between the heater and the evaporation sections.

The heater and the test section were installed on the vertical plywood board. The tubes were supported at two points by nylon bridges mounted on the desk board. The nylon boxes were screwed into the plywood board and covered with an additional thick insulation [(Styrofoam box with external dimensions: 163 mm $\times$ 130 mm $\times$ 135 mm (height - 163 mm), inside opening: 110 mm $\times$ 76 mm $\times$ 75 mm (depth - 75 mm))] to prevent heat loss to the environment. The parts of the tubes soldered to the evaporation section and the fins are located perpendicular to the horizontal plane with the finned parts at the top of the tube.

3.1.2.4 Charging System

The charging system consisted of a vacuum pump, a McLeod gauge, vacuum valves and a syringe to charge the desired amount of the working fluid into the test TLT.

3.1.3 Working Fluids

To find the working fluid most suitable for the system, refrigerants such as R-11, R-113, ethanol, water, FC-72, FC-87, and acetone were tried under various conditions. Some physical properties of the working fluids used in the experiment are presented in Table 3.4. Theoretically, any kind of fluid could be used as the working fluid but water could be the best choice. Water has a high latent heat thermal property, permits the transmission of heat more than all other working fluids, is cheap, easily available, and is non-combustible. On the other hand the disadvantages of water are that it has a high freezing temperature, has a high vapour pressure at moderate temperature, and has a tendency to react with some substances (alkali metals, etc.) which releases hydrogen and under certain conditions could cause an explosion.

For low temperature range applications, refrigerants with chlorine are commonly used. But the use of these refrigerants has been already restricted by the Montreal Protocol in 1987 and is now completely prohibited [Satutory Authority, 1995]. These refrigerants with chlorine deplete the UV filtering atmospheric ozone layer. In the present experimental study, R-113 and R-11 were used as references only. That is why it was decided to use for the new environmentally friendly fluids such as fluorinert liquids FC-72 and FC-87. These were suitable for low temperature ranges and were in full accordance with the restrictions mentioned above. Unfortunately, the behaviour of these new coolants as boiling working fluids has seldom been studies (only for the conditions close to the pool boiling of FC-72) and none was reported in the literature on the heat transfer rates in TLTs except for the study by Kishimoto et al. [1994] for FC-87.

In this study, acetone, FC-87 and water were used as the primary working fluids and FC-72, R-113, R-11 and ethanol as the supplemental. Acetone was used in initial series of experiments to investigate the heat transfer characteristics of the proposed system.

3.1.4 Experimental Procedure

Before the start of an experiment, the test loop thermosyphon was thoroughly cleaned and washed with ethanol and then with distilled water. It was then vacuum dried, and tested for vacuums and leaks. The inside air and other non-condensable gases were removed from the thermosyphon tube with a Leybold mechanical vacuum pump (type ADEA 71 N4). The pump usually reached a magnitude of 8×10^{-3} torr for a dry system.

After the thermosyphon was evacuated, a known amount of the working fluid was charged into the tube using the charging system. In this study, the amount of the working fluid usually charged was about 50 ml for the thermosyphon. This was about 10 times larger than the volume of the thermosyphon evaporator. The initial system pressure without the working fluid was not higher than $(5 - 10) \times 10^{-3}$ torr.

A great care was taken to remove all the air and the other non-condensable gases from the charging system and the thermosyphon was tested before a new working fluid was charged. Usually non-condensable gases accumulate in the upper region of the TLT during the operation. This creates a gas “plug” in the condenser section which would prevent the vapour from condensing. Because of this “plug”, the temperature at the upper end is lower than temperatures along the tube. In such a case, a vacuum pump is connected to the vacuum valve and the non condensable gases were ‘vacuumed’ out.

The power to the evaporator heating section was increased carefully in increments to the desired heat flux. It took approximately 1 hour to reach the steady state. Voltage was seen to fluctuate within a range of ± 0.2 V. While supplying the power, a great care was needed not to pass the maximum heat flux for that particular condition.

The air flow velocity and air temperature were measured with the thermo-anemometer. The air flow velocity carefully controlled by a slip switch when the velocity setting was changed, it took about 10 minutes to achieve steady state readings. Once a steady velocity was obtained, various values were recorded.

The velocity of the flow depended on the length of condenser because a change of the length of the wind duct in the condenser section lead to a change of the air flow velocity as mentioned in Section 3.1.2. The temperature was checked for a steady value after a steady velocity and heating power were obtained. To determine these temperatures, the signals from the thermocouple in the heater centre were observed until no temperature fluctuations were registered (approximately 40-60 minutes). When this occurred, the signals from all the thermocouples installed in the system were recorded.

As mentioned earlier, if non-condensable gases were present, then the temperature readings of the upper end and the condenser will show much lower values than those for other sections of the TLTs. This indicates that the vapor from the evaporator could not move up to the condenser section because the non-condensable gases block the tube. In these cases, the non-condensable gases were removed as discussed above. If it was too difficult to remove the non-condensable gases, the experiment was restarted from the beginning. Therefore, all the working fluid was removed from the test TLT. This procedure was repeated many times with FC 87 because it has a large amount of dissolved gas before it is charged. (see Table 3.5). To remove the non-condensable gases, the quantity of the working fluid was charged about 10% more than that intended, because it was expected that a small amount of the working fluid would be lost during the removing process of non-condensable gases.

Once the system is stable, all the data were recorded. After each series of tests for a given working fluid, the thermosyphon was again thoroughly cleaned, washed, vacuum dried and checked for leaks. After the system was evacuated, a known amount of a new test working fluid was charged into the test tube and a subsequent series of tests was performed using the same procedures.

3.1.5 Data Reduction

The overall heat transfer coefficient, U_T , of the test TLT is defined as:

$$U_T = \frac{Q / A_h}{t_h - t_c} \quad (3.1)$$

The heat flux per unit area was calculated from the power measurements which was obtained from the voltmeter and ammeter coupled into the power supply circuit of the main heater. The heat losses through the insulation of the heating and evaporation section were seen negligible because of the good insulation of the heating and the evaporator sections.

The errors involved in the calculation of the heat transfer coefficient were generally due to the inaccuracy of the temperature and the power measurements. Even if the readings of the power and the temperatures were recorded after the steady state has been reached, a small fluctuation was observed (± 0.2 V for voltage, ± 0.01 A for current and ± 0.2 °C for temperature).

3.1.6 Instability Measurement

At the start of this investigation, the transient temperature variation of the test SSL was recorded on three X-Y pen-recorders (FISHER RECORDALL Series 5000, LINEAR model: 1201-0000 and BBC GOERZ). The transient temperature variation could be recorded on the papers (LINEAR model: 1201-0000 and BBC GOERZ: feeding speed was 1 cm per minute at 2.5 mV range, FISHER RECORDALL Series 5000: feeding speed was 2.5 cm per minute at 10 mV range). All recorders were calibrated before the experiments. The noise was specified to be a maximum of $\pm 2\%$ rms.

3.2 Medium Scale Two-Phase Closed Loop Thermosyphons (MSL)

Two medium scale TLTs were used in the study. One (MSL I) has a heat transfer rating of 60 W and the other one (MSL II) of 1,500 W. MSL I was heated by hot air and cooled by a direct conduction heat sink and MSL II was heated electrically and cooled by cold water.

3.2.1 Medium Scale Two-Phase Loop Thermosyphon for Heat Extraction of Enclosed Spaces (MSL I)

3.2.1.1 Design and Specification for MSL I

The experimental apparatus which is schematically illustrated in Fig. 3.8 to 3.11 consists of the main TLT assembly, the cooling system for the condenser section, a well insulated confined space and a charging system. The main TLT assembly which is to replace the conventional refrigeration systems is made of three parts; the finned evaporation section which extracts heat from the air in a

confined space, the long smooth transport section, and the condenser section cooled by an external cooling source.

Two fin & tube type ($\phi 8 \sim 9.52$ mm) evaporator sections (Figs. 3.10 and 3.11) were made from the same aluminum tubes and plates with the dimensions for Case 1: 520mm x 200 mm x 50 mm and for Case 2: 440mm x 200 mm x 50 mm, respectively, and the heat transfer to the evaporator surface is by natural convection. The effective *outer* surface areas (heat transfer surfaces) are: for Case 1, 1.53 m² and for Case 2, 0.89 m², and the effective *inner* surface areas are: for Case 1, 0.158 m² and for Case 2, 0.105 m², respectively.

The details of the external cooling source is a proprietary information of the host company, Samsung Electronics, and therefore could not be provided in the thesis. However, the lack of the technical information on the cooling source has no bearing on the execution of the present study, i.e., the heat transfer performance of a new TLT. The effective surface areas of the cooling source are 0.0114 m² for both Case 1 and 2 and the cooling surface temperature was controlled and maintained at a desired temperature within ± 0.5 °C. Fig 3.9 shows the details of the condenser sections.

The evaporator and condenser sections are connected with two transport lines as seen in Fig.3.8; one for the two-phase flow from the evaporator to the condenser, and the other for the condensate return from the condenser to the evaporator.

A temporary closure of the loop thermosyphon was made as was the case of SSL with high grade vacuum valves which were connected to the upper part of the systems through aluminum tubes. To measure the temperature distribution over the length of the loop thermosyphon, a total of twelve thermocouples were used for each system. All the thermocouples used were calibrated and connected to a computer controlled data acquisition system.

The charging system for the working fluids consisted of a vacuum pump, a vacuum gauge, vacuum valves and a connector system which allow to charge the desired amount of the working fluid into the MSL I.

For relatively low temperature range applications of interest, refrigerants with chlorine would be most convenient. However, as stated previously the use of these refrigerants is now completely prohibited by the Montreal Protocol. For the ambient air average temperature of the confined spaces of 1, -17 °C for Case 1, and +3 °C for Case 2 , respectively, which are the temperature of interest in the present study, the refrigerant R-134a was seen to be the most suitable for the systems and used as the working fluid.

The cleaning processes of the test loop thermosyphons were the same as that for the case of SSL. The thermosyphon tubes were connected to vacuum pumps to remove the inside air and other non-condensable gases. The pump usually reached to the magnitude of 5×10^{-3} torr for a dry system.

After the thermosyphons were evacuated, known amounts of the working fluid were charged into the tubes using the charging system. In the present study, the amounts of the working fluid charged were 100 g for Case 1 and 130 g for Case 2, respectively.

3.2.1.2 Data Reduction

The overall heat transfer coefficient, U_T , of the test TLT is defined as the same as that given in Section 3.1.5. It was seen that the heat loss in the condenser section was negligible because of the thick insulation. The errors involved in the calculation of the thermosyphon heat transfer coefficient were generally due to the inaccuracy of the temperature and the flow rate of the coolant. The

readings of the coolant flow rate and the temperatures were recorded after the steady state had been reached and fluctuations of $\pm 0.5\%$ for the coolant flow and $0.2\text{ }^{\circ}\text{C}$ for temperature were observed.

3.2.2 Medium Scale Loop II (MSL II)

3.2.2.1 Design and Specification for MSL II

The medium scale TLT No. 2 used in the present study are located at KIER (Korean Institute of Energy Research), Korea. This loop was designed for a heat transfer capacity of 1500W and the objective of this MSL II was to identify the basic parameters which govern large industrial size TLTs to be tested at KIER. This was a joint collaborative project between the University of Ottawa and KIER.. The MSL II consists of two sets of vertical pipe sections defined as the transport section, which are joined to the evaporator and the condenser sections of the loops. The overall design specification of MSL II is given in Table 3.1.

Distilled Water was used as the working fluid as well as the coolant for condenser section. The small electric powered heat exchanger was installed at the condenser section. This allowed the cooling water in the condenser section to be controlled at a constant temperature. The test assembly as shown in Figs. 3.12, and Plates 3.9 and 3.10 was designed and manufactured with the following considerations:

Heating Section

The evaporator section shown in Fig. 3.12 and Plate 3.10 consisted of a single pipe (ID 20.2 mm, OD 27.2 mm and 300 mm long) with an electrical heating wire wound around along the length of the pipe. As shown in Fig. 3.12, at a 30 mm above the evaporator, a glass tube of 70 mm in

length was installed for flow visualization studies. This made it possible to observe the flow phenomena of the working fluid leaving the evaporator which is an essential information needed for the subsequent comparisons with the simulation program results. The power input to the heating wire could be varied continuously from 0 to 2400 W of the rated input and was measured by a Watt Transducer (KTG-12-0). The changes in the line voltages were less than a 2% variation for the power input. A 100 mm thick insulation was placed around the evaporator to prevent heat loss. The constant heat fluxes were supplied at 500, 700, 1000, 1200 and 1500 W.

Cooling Section

Fig. 3.12 and Plate 3.10 show the condenser section of MSL II. The condenser section was made from a single pipe (ID 20.2 mm, OD 27.2 mm and 300 mm long) and a water jacket outside to supply the cooling water. The temperature of the cooling water was controlled by an electric heater inserted in the water reservoir tank. A rotary type flow meter (OVAL FLOWPET, 200 - 1,200 l/hr) was used to measure the flow rate of the cooling water supplied to the condenser section. The flow rate of cooling water was supplied at 310 l/hr. The temperature of the cooling water was fixed at 50, 60 and 70 °C.

50 mm below the condenser, a 70 mm glass tube was provided for flow visualization of the condensate leaving the condenser.

Transport Section

The transport section was made from single pipe (ID 20.2 mm and OD 27.2 mm) which connects the evaporator and the condenser forming a rectangular loop as seen in Fig. 3.12 and Plate 3.9.

General Descriptions

The MSL II could be charged with a working fluid to any desired percentage of the full capacity. In the present study, three filling rates were charged the loop at 30, 50 and 70 % of the volume of the evaporator. The loop was well insulated to minimize the heat loss from the loop to the surrounding air. The temperature of the evaporator and condenser pipe surfaces were measured using seven thermocouples (K-type ϕ 0.3 mm) mounted along its length. The thermocouples mounted in the evaporator and the condenser surfaces were silver soldered at a equal spacing along the length. As shown in Fig. 3.12, the inlet and outlet temperatures of the evaporator and the condenser were monitored using 4 probe type thermocouples (K - type sheath dia. 1.6 mm). The positions of installed thermocouples are shown in Fig. 3.12.

Pressure transducers (PDCR 130/W, 0~5 kg/cm²) were installed at the evaporator entrance and exit to obtain the saturation pressure. Also, the pressure of the condenser entrance and exit were measured by pressure transducers (P21AA, 0 ~ 5 kg/cm²) installed in the entrance and the exit of the condenser. Fig. 3.12 also shows the positions of the pressure transducers. The MSL II was designed to allow for a change in the vertical distance between the evaporator and condenser sections at intervals of 0.5 m, 1 m and 1.5 m. Also the horizontal distance could be changed to a maximum of 3 m. As shown in Plate 3.9, all measured data were recorded by a computer based data acquisition

system (FLUKE, HELIOS Plus 2287A). The data acquisition system was programmed to scan all of the thermocouples and then the electromotive force from the thermocouples were converted to digital forms for the computer.

Vacuum

All the loop components were cleaned to remove the contaminants using distilled water before assembling the loop. After assembly, the vacuum of a loop was tested under pressure and Nitrogen was injected into the loop to check leaks .

3.3 Large Scale Two-Phase Loop Thermosyphons (LSL I and LSL II)

Two large TLT loops (LSLs) were used to obtain both performance data and some insight into the flow behaviour within large loops which consist of the evaporator section, the condenser section and transport sections. The heating and cooling methods were designed so that the heat balance of each loop could be easily made.

As a part of the joint collaborative study between the University of Ottawa and KIER, the two LSLs used in the present study were located at KIER (Korean Institute of Energy Research), Korea. The heat transfer capacities of the two LSLs were 7,500W and 100,000W, respectively. The design specification of the two LSLs are given in Table 3.1.

3.3.1 Large Scale Loop I (LSL I) for Waste Heat Recovery System

3.3.1.1 Design and Specification for LSL I

The 1st large scale TLT which is shown in Fig. 3.13 is a test loop to be used as a waste heat recovery system. The LSL I was designed to have the maximum heat capacity of 7500 W and the objective of the LSL I is to see it. Such a system could be used economically as a waste heat recovery unit. The LSL I is also a precursory test TLT for the preparation of a much larger system, that is LSL II. Figs. 3.14, 3.15, and Plates 3.11, 3.12, and 3.13 show the details of the test LSL I.

Heating Section

As shown in Fig. 3.14 and Plate 3.13 (a), the evaporator of the LSL I consists of a five-pipe unit made of 5 carbon steel pipes (ID 20.2 mm, OD 27.2 mm and 300 mm long). The five carbon steel pipes which have electric heating wires wrapped along the lengths were connected to the evaporator headers (bottom headers and top headers). The both top and the bottom headers are $\phi 61$ mm ID, $\phi 70$ mm OD and 271 mm in length. The lower end of the transport pipe was connected to the evaporator by means of a pair of flanges (5K-20A). The flow path to the test section was tapered to minimize any disturbance in the system due to a change in cross-section area. Gaskets were used to seal the flange joints.

The evaporator pipes were connected to auto-transformers separately and the input power to each pipe was controlled by different power supplies. The evaporator pipes are well insulated to minimize heat loss.

Cooling Section

Fig. 3.15 and Plate 3.13 (b) show the details of the condenser assembly of the LSL I. The condenser section consists of five finned pipes (ID 20.2 mm, OD 27.2 mm and 300 mm long) which were connected to the two condenser headers. The cooling of the condenser section was by cold air forced convection. The cooling air temperature was limited by the outdoor conditions which were in the range of about 20 to 30 °C. The cooling air was supplied to the condenser section by a Blower (40-50 CCM flow rate, 120 mmAq) and the RPM of the blower was controlled by an inverter. A hot wire anemometer (V-01-AND) was placed in the condenser inlet duct to measure the cooling air velocity. Each pipe row of the condenser assembly has an exhausting valve to release any non-condensable gas remained inside.

Transport Section

The transport section of LSL I was made of carbon steel pipes (ID 20.2 mm, OD 27.2 mm) and the total length was 2173 mm as indicated in Fig. 3.13. One end of the transport pipe lines was connected to the header of the evaporator by means of a pair of flanges (5K-20A) and the other end to the condenser header. Two valves were provided at the end and the top of the transport pipe line on the evaporator side. The top valve was used to remove the superheated vapor and the bottom valve to drain off the working fluid.

General Descriptions

The LSL I was well insulated to minimize the heat loss from the loop to surrounding air. As shown in Fig. 3.13, the temperature of the evaporator and condenser pipe surfaces were measured using three thermocouples mounted along its length.

The details of the evaporator and the condenser assemblies are shown in Fig. 3.14 and 3.15. The vertical distance between the evaporator and the condenser was 1 m and the horizontal distance 10 m. The LSL I was designed so that the vertical distance could be changed from 0.5 m to 1.5 m.

Pressure Measurement

Four absolute pressure transducers (Transmetrics P21AA, 0 ~ 20 kg/cm²) were installed at the inlet and outlet of the evaporator and the condenser sections individually to record the pressure variations.

Temperature Measurement

The working fluid temperatures were measured at various locations along the loop with 0.011-inch diameter chromel-alumel sheathed thermocouples (K-type sheath dia. 1.6 mm ϕ OD). The thermocouples were provided at the inlets and outlets of both the evaporator and the condenser sections (a total of 4 thermocouples) as shown in Fig. 3.13. The time constant of these thermocouples was estimated to be in the range of 50-150 milliseconds. Three thermocouples (K-type ϕ 0.3 mm) per each evaporator pipe were installed, respectively, on the surface of the pipe for the evaporator pipe surface temperature (a total of 15 thermocouples). As shown in Fig 3.13, three thermocouples (K-type ϕ 0.3 mm) were also installed at the inlet and the outlet of the cooling air duct (a total of 6 thermocouples).

Data Acquisition

For the LSL I, a data acquisition system (Fluke Helios Plus 2287A) shown in Plate 3.12 (b) was used to determine temperatures, the saturation temperature of the working fluid and pressures.

The auto-transformers (KTG-12-0) were connected into the electrical heating wires of the evaporator pipes. With these auto-transformer, the input voltage and current were measured and supplies were controlled.

3.3.2 Large Scale Loop II (LSL II) for Waste Heat Recovery System

3.3.2.1 Design and Specification for LSL II

The 2nd large scale TLT which is schematically shown in Fig. 3.16 is to be used as a full size industrial waste heat recovery system. LSL II was designed to have the maximum heat capacity of 10 kW and to be used as a prototype for a full size industrial heat recovery system. Figs. 3.16 to 3.21, and Plates 3.14 to 3.16 show the details of the test assembly of LSL II.

Heating Section

As shown in Fig. 3.17 and Plate 3.15, the evaporator consists of five pipe rows which have 12 carbon steel pipes (ID 20.2 mm, OD 27.2 mm and 800 mm long) each, making a tube bank of 12 × 5 pipe array. Each individual evaporator pipe has fins (0.8 mm thickness and 11.4 mm height) plasma welded at 3.6 mm intervals on the surface. The pipe bank has a triangular configuration in geometry as shown in Fig. 3.18, and the fined pipes were welded to the top and the bottom each evaporator headers. Each transport pipe was connected to the evaporator by means of a pair of flanges (5K-20A). There are five headers and five transport pipes. The flow path to the test section

was tapered to minimize any disturbance in the system due to a change in cross-section area. Gaskets were used to seal the flange joints.

A high temperature plastic tubing was connected at the bottom header outlet and the top header inlet of the each evaporator pipe to check the filling rate. The first pipe row of the evaporator section directly faces the hot air from the burner (Baltur BT 8G). The working fluid in the first pipe row would start to boil earlier than those in the other pipe rows. Therefore, the first pipe row has a valve to release the superheated vapor at the top of its transport pipe on the evaporator side because this high temperature would cause can make high pressure which would damage the system.

The LSL II was heated by the exhaust air from the burner (Baltur BT 8G) as illustrated in Fig. 3.16. and Plate 3.14. The hot air temperature was limited to 300 °C. The oil burner (TYPE VE3.2, Brahma, max. 150,000 kcal) is rated 150,000 kcal/h heat which could heat air up to the maximum temperature of 300 °C. The temperature of the hot air was controlled by a three way damper at the range of 200 ~300 °C.

Cooling Section

Fig. 3.19 and Plate 3.16 show the condenser assembly which has the same pipe bank configuration as that of the evaporator section except that, the first pipe row had 11 pipes (ID 20.2 mm, OD 27.2 mm and 800 mm long). The cooling air temperature was limited by the outdoor conditions which were the same as for the case of the LSL I. The cooling air was supplied to the condenser section by a Blower (ShinPung SPTF, 10HP, 200 CCM flow rate, 1895 RPM). An air-water heat exchanger was installed to control the temperature of the cooling air. As was the case of

the LSL I, each pipe row of the evaporator has an exhausting valve to release only non-condensable gas remained inside.

Transport Section

The transport section of Loop II has five parallel pipe lines and each pipe line was made of carbon steel pipes (ID 20.2 mm, OD 27.2 mm). The total length of the transport pipe line was 12208 mm as indicated in Fig. 3.16. The ends of the transport pipe lines were connected to the header of each pipe rows of the evaporator by means of a pair of flanges (5K-20A) and the other line ends to the condenser headers. Two valves were provided at the end and the top of each transport pipe on the evaporator side. The top valve was used to remove the superheated vapor and the bottom valve to drain off the working fluid.

General Descriptions

The LSL II was insulated to minimize the heat loss from the loop to the surrounding air. As shown in Fig. 3.20, the temperature of the evaporator and condenser pipe surfaces were measured using three thermocouples mounted along its length.

The evaporator and the condenser assemblies are shown in Fig. 3.17 and 3.19. The vertical distance between the evaporator and the condenser was 1 m and the horizontal distance was 10 m.

As shown in Fig. 3.20, the LSL II has two different test configurations (counter and parallel) depending on the air flow direction of the condenser section. The condenser section of LSL II was designed so that the flow direction could be changed easily; either counter flow or parallel flow as shown in Fig. 5.20.

Pressure Measurement

Two differential pressure transducers (P55D 1N132S44, 0 ~ 1000 mmH₂O) installed at the inlets and outlets of the condenser and evaporator sections to measure the pressure drop across the loop. Three absolute pressure transducers (Transmetrics P21AA, 0 ~ 20 kg/cm²) were provided at the evaporator and the condenser section outlets (1st, 3rd and 5th pipe row) to record the local pressure variations. It was found that in most cases this pressure variation was quite small.

Temperature Measurement

The temperatures of the working fluid temperatures were measured at various locations in the loop with 0.011-inch diameter chromel-alumel sheathed thermocouples (K-type sheath dia. 1.6 mm ϕ OD). The tips of the thermocouples were placed approximately at the center of the flow cross sections. The location of the thermocouples (a total of 20 thermocouples) are shown in Fig. 3.20. The time constant of these thermocouples are known to be in the range of 50-150 milliseconds. As shown in Fig 3.21, a set of three thermocouples (K-type ϕ 0.3 mm) were placed at the inlets and the outlets of the hot air and cooling air ducts (a total of 18 thermocouples).

Data Acquisition

For the LSL II, the data acquisition system (Yokogawa Hokushin Electric, Model 3081 and 2500E Hybrid recorder) was used to determine temperatures, the saturation temperature of the working fluid and pressures. The pressure heads were measured using pressure transducers (P55D 1N132S44, 1000 mm H₂O) connected to micromanometers.

CHAPTER 4

ANALYSIS

4.1 Simulation Overview

As mentioned in Chapter 1, a TLT (Two-Phase Loop Thermosyphon) is an integrated system with both single-phase and two-phase flows. Unlike single phase flows, two-phase flow are affected by many flow parameters and very difficult, if not impossible, to obtain any solution from the first principles. Therefore, a purely analytical solution for a TLT system is not possible due to the two-phase fluid flows involved. Because of this difficulty, the analytical aspect of the TLTs is replaced with a system simulation code development in the present study. The aim of any simulation for a mechanical system is to provide the capabilities to predict many variables which would affect the performance of the system under different operating conditions.

In the present study the simulation study is based on two different simulation methods. The major difference is how to manage the simulation procedure for the section between the evaporator and the condenser inlets. The lumped method does not consider the different flow patterns which may exist in the section, the evaporator section is dealt as one lumped section with a lumped two-phase parameter. The sectorial method is built on the flow regimes of the two-phase flows involved because the heat transfer involved in a TLT is that of flow boiling.

The TLT simulation code is developed to provide the following capabilities.

- the heat transfer rate versus the temperature difference between the heat source and the heat sink,
- the temperature distribution along the loops,
- the saturation temperature of the TLT and the heat transfer capacity of the system,
- the effect of the mode of cooling at the condenser section (, i.e., forced and free,) on the heat transfer capacity of the system,
- the characteristics of the cooling in the condenser section with various heat transfer enhancing mechanisms (e.g. fins),
- the effect of the loop length on the heat transfer capacity of the system,
- the choice of the working fluid,
- the effect of the contact resistance and the heat transfer capacity of the system. (only for SSL and MSL I), and
- the presentation of the pertinent information in graphs.

4.2 Simulation Code Development

The simulation model which was derived from the ideal model such as shown in Fig. 3.1 is based on a thermal resistance network as illustrated in Fig. 4.1.

4.3 Thermal Resistance Network

Under steady-state conditions, the thermal resistance model as shown in Fig. 4.1 can be used to estimate the heat transfer rate of a TLT cooling system.

The sectorial thermal resistance between the hot section and the cold section may be written

as:

$$R_T = R_{hot} + R_{loop} + R_{cold} = \frac{1}{U_T A_h} \quad (4.1)$$

$$R_{hot} = R_{fil} + R_{pl} \quad \text{for SSL,} \quad R_{hot} = R_{conv} + R_{ht} \quad \text{for other TLTs} \quad (4.2)$$

$$R_{loop} = R_{ev} + R_{tr} + R_{cond} \quad (4.3)$$

$$R_{cold} = R_{ct} + R_{conv} \quad (4.4)$$

- where
- R_T : the total resistance between the hot section and the cold section.
 - R_{hot} : the thermal resistance of the evaporator
 - R_{loop} : the thermal resistance of the thermosyphon loop
 - R_{cold} : the thermal resistance of the condenser
 - R_{fil} : the thermal resistance of the filler (for SSL only)
 - R_{pl} : the thermal resistance of the holding plate (for SSL only)
 - R_{ev} : the thermal resistance of the evaporation
 - R_{tr} : the thermal resistance of the transportation section
 - R_{cond} : the thermal resistance of the condensation
 - R_{ct} : the thermal resistance of the condenser tube wall
 - R_{conv} : the thermal resistance of the forced convection to the condenser section

Each of these thermal resistances are presented in Appendix G which show that the thermal resistance method requires a number of empirical correlations for various thermal resistances. Several empirical correlations for flow boiling (Appendix C), condensation (Appendix D), enhanced forced and free convection (Appendix H), and the equations of fin efficiency (Appendix I) were collected and evaluated for its applicability.

For SSL, the conduction shape factors needed for the evaporator section were estimated by analytical solutions from the literature, which are given in Appendix I.

4.4 Definitions

The vapor quality is defined as,

$$x = \frac{(h - h_f)}{h_{fg}} = \frac{\dot{m}_g}{\dot{m}_g + \dot{m}_l}, \quad (1-x) = \frac{\dot{m}_l}{\dot{m}_g + \dot{m}_l} \quad (4.5)$$

The void fraction can be defined as the ratio of the flow area occupied by vapor to the total flow area as:

$$\alpha = \frac{A_g}{A_T} \quad \text{and} \quad (1 - \alpha) = \frac{A_l}{A_T} \quad (4.6)$$

Mean density is defined as

$$\rho_{mean} = \alpha \rho_g + (1 - \alpha) \rho_l \quad (4.7)$$

Phase velocities,

$$v_f = \frac{\dot{m}_l}{\rho_l A_x (1 - \alpha)} \quad (4.8)$$

and

$$v_g = \frac{\dot{m}_g}{\rho_g A_x \alpha} \quad (4.9)$$

Slip Ratio,

$$S = \frac{v_g}{v_l} \quad (4.10)$$

Conservation Equation

Mass conservation :

The equation of mass conservation for a steady state flow can be given as:

$$\frac{d\dot{m}}{dy} = 0 \quad (4.11)$$

Momentum conservation :

The momentum equation for steady state flow is,

Total pressure drop = acceleration pressure drop + frictional pressure drop + hydrostatic pressure drop + the pressure drop in bends, fittings etc.

$$-\frac{dp}{dy} = \left(\frac{dp}{dy}\right)_F + G^2 \frac{d}{dy} \left\{ \frac{(1-x)^2}{\rho_l(1-\alpha)} + \frac{x^2}{\rho_v \alpha} \right\} + g \sin \theta [\alpha \rho_v + (1-\alpha) \rho_l] + \left(\frac{dp}{dy}\right)_{ms} \quad (4.12)$$

In a closed flow loop such as TLTs, the simulation of the pressure drop is zero.

Energy conservation :

The heat transfer rate to a mixture per unit length, q_l , is equal to the rate of the enthalpy change in the mixture, neglecting the changes in the potential and kinetic energies.

$$q_l = \frac{d}{dy}(GH) = G \frac{dH}{dy} \quad (4.13)$$

$$\frac{Q}{GA} = \frac{dH}{dy} \quad (4.14)$$

4.5 Lumped Method

4.5.1 Simulation Logic

As seen in Fig. 4.1, the resistance, R_{loop} , must be determined through various two phase flow parameters involved. Initially, the working fluid of a loop thermosyphon enters at the bottom of the evaporator section as either saturated or subcooled liquid in either laminar or turbulent flow with developing momentum and thermal boundary layers. As the fluid rises up the thermosyphon, its pressure and hence saturation temperature drops along the flow channel. As the fluid progresses along the evaporator section of the TLT, the vapor fraction increases and hence the apparent flow velocity increases. Two-phase forced convection becomes dominant.

Therefore, the main concept of the lumped method is as was illustrated in Fig. 1.2 that for a steady state condition, the two-phase flow in the evaporator section of a TLT may be assumed to be in a lumped forced convective boiling state, and accordingly the evaporator section of the TLT can be assumed as one lumped section.

In the condenser, only film condensation with negligible vapor velocity is assumed in the present analysis. If the condenser is partially flooded, the mode of heat transfer in the flooded region becomes single-phase liquid forced convection.

The program is made to estimate an initial pressure and temperature in the evaporator liquid inlet and the mass flow rate of the TLT, then proceed to calculate the heat transfer coefficient, the exit state of the working fluid and the fluid temperature at the downstream of the evaporator. The pressure drop and the heat loss along the section between the evaporator and condenser sections are to be calculated. Assuming a negligible pressure drop along the condenser section, and by knowing the inlet and the exit states of the condenser, the heat transfer in the condenser and the temperature of the fluid leaving the condenser section is calculated. The inlet pressure of the evaporator section is varied until the incoming liquid temperature is achieved. Once this convergence is reached, an energy balance applied to the evaporator section would yield a new value for the fluid temperature at the inlet of the evaporator section. This new value is used to generate the final values including the static liquid level which would exist in the loop upon shutdown. The whole simulation procedures by the lumped method are shown in the flow charts of Fig. 4.2 (a) for the SSL and MSLS and Fig. 4.2 (b) for LSL I and II.

4.5.2 Flow Regimes

When a fluid is flowing inside a tube either in laminar or in turbulent flow condition, single-phase convection heat transfer will occur. There are numerous correlations for this mode of heat transfer and can be found in most heat transfer text books.

As heat is added into an evaporator surface, the transition from single phase convection to nucleate boiling occurs when the liquid layer adjacent to a heating surface becomes superheated. It is now an accepted fact that the presence of nucleation cavities is required to establish boiling on a metal surface. Many investigators have presented correlations to predict the incipient boiling point. Most of these investigators studied the phenomenon of the wall superheat during the incipient boiling in tubes. The expression of Davis and Anderson [1966] related to the incipient heat flux and superheat is generally now accepted which is given as:

$$\left(\frac{q}{A} \right) = \frac{k_f h_{fg} \rho_g}{8 \sigma t_{sat}} (t_w - t_{sat}) \quad (4.15)$$

When the liquid near a wall surface is superheated, then bubbles are formed. The bubbles transport the latent heat of the phase change and also increase the convective heat transfer by agitating the liquid near the heating surface. This boiling mechanism is called nucleate boiling and is characterized by a very high heat transfer rate for only a small temperature difference. Nucleate flow boiling occurs in tubes when the quality is low. When the quality increases, a transition occurs from nucleate boiling to two-phase forced convection. The quality plays an important role in determining the onset of this transition. Unfortunately, there is hardly any published data to predict the onset of such a transition. Next stage in the heat transfer regimes is that of two-phase forced convection heat transfer. This is the most important heat transfer regime when the fluid is undergoing

a change of phase in an evaporator tube. It is again very difficult to find the transition criteria between the nucleate boiling and forced convection evaporation in open literature. The reason is mainly from experimental difficulties and because of this, many researchers tried to integrate these flow regimes as one two-phase heat transfer regime. One of the best example is a correlation of Chen [1966]. Chen's correlation include both effects of the two heat transfer regimes, i.e., forced convection and boiling.

As the quality reaches $x \approx 1$, the transition from two-phase forced convection to the liquid deficient region occurs. Although the transition from two-phase forced convection region to liquid deficient region is not encountered in the present system (from the experimental experience), this is given for the completeness. In this region, the correlation of Rhee and Young [1975] for x_{cr} could be used. if the quality exceeds the critical quality, dryout occurs, and x_{cr} is defined as:

$$x_{cr} = \frac{We \sigma \rho_g}{\dot{m}_l \mu_l \left(\frac{Re_l}{(1-\alpha)^{0.125}} \right)} \quad (4.16)$$

$$We = 1.87 \times 10^{-7} Re_g \left(\frac{\rho_l}{\rho_g} \right) \left(\frac{D}{0.0608} \right)^{-0.48} \quad (4.17)$$

The flow regimes considered in the lumped method are summarized in Table 4.1.

4.5.3 Heat Transfer

For a TLT system, the heat transfer coefficients for various flow regimes available in the literature can be predicted with correlations or models. Although there are a large number of

prediction methods available, it is difficult to select a suitable one for given conditions. As described in Chapter 1, most of the prediction methods in the literature are valid only for limited range of conditions with large uncertainties.

(a) **Single-Phase Liquid Flow Region**

The first heat transfer region encountered when a working fluid enters the evaporator of a TLT is that of single phase heat transfer to the liquid phase. In this region, the working fluid may enter the evaporator in a subcooled state. As the fluid progresses up, heat transfer takes place by **single phase forced convection** and the pressure decreases.

For the calculation of the heat transfer in the single phase forced convection region, the correlations of Sieder and Tate [1936] given as Eqs. (4.18) and (4.19) below, were used to estimate the local heat transfer coefficient. Sieder and Tate's equation is mainly the correction of Dittus and Boelter [1930] equation to take account into the property variations in a heated pipe.

For laminar flow:

$$Nu = 1.86(Re_l Pr_l)^{1/3} \left(\frac{d}{L} \right)^{1/3} \left(\frac{\mu_l}{\mu_w} \right)^{0.14} \quad (4.18)$$

For turbulent fully developed flow:

$$Nu = 0.027 Re_l^{0.8} Pr_l^{1/3} \left(\frac{\mu_l}{\mu_w} \right)^{0.14} \quad (4.19)$$

(b) Two-phase Flow Heat Transfer (Forced Convection)

At the point where the bulk fluid temperature reaches the saturation temperature, a two-phase flow heat transfer coefficient h_{tp} is estimated using correlations from literature shown in Appendix C. There are numerous investigations on forced convection boiling heat transfer, most of them if not all are experimental. Because of the great variety of the effective parameters and the complexity of the phenomena, purely theoretical analysis cannot provide a general equation for the boiling heat transfer coefficients for different substances and different conditions. All correlations given in the literature have a relatively large range of uncertainty and can be used only in restricted cases. For this reason measurements on heat transfer coefficient for different substances and different conditions still appear necessary.

Ten flow boiling correlations from the literature have been checked for their applicability to our case, and they are listed in Table 4.2, and their results are compared with our experimental data in Chapter 5.

(c) Dryout Region

As stated previously, this region is not encountered in the present study, but the relevant empirical equation is given for the completeness. The heat transfer in this region may be calculated using the following equation of Rhee and Young [1975];

$$h_{dryout} = 0.0162 \left(\frac{k_g}{D} \right) \text{Re}_g^{0.84} \text{Pr}_g^{0.33} (1 - x_{cr})^{0.1} \quad (4.20)$$

(d) Single-Phase Vapor Flow

As the flow progresses up, the magnitude of the quality reaches 1, and the flow is in vapor phase. The relations described in the previous section '*single phase flow*' is applicable.

(e) Condensation

The TLT system is mainly operated by the evaporation and the condensation processes. Condensation, the process by which a vapor is converted to a liquid, is one of the two most important transfer operations. Condensation may occur in a variety of modes (filmwise and dropwise). Here, it is assumed that the condensed liquid forms a continuous liquid film on the heat transfer surface as is the general industrial practice for the design of a condenser. The best known analysis of the film condensation is the laminar film condensation on a vertical surface by Nusselt [1916]. The analysis of Nusselt has been modified for various geometries and orientations.

- Vertical film condensation by Nusselt theory [1916]

If the vapor is in a superheated state, vapor desuperheating takes place and the wall temperature is calculated using single-phase vapor heat transfer correlations. Condensation heat transfer takes place when vapor is cooled sufficiently below the saturation temperature. The equations used in the determination of the condensation coefficient are given in Appendix D.

- Condensation through integrated horizontal finned tube [Smirnov and Lukanov, 1972]

The film condensation on integral-fined surface in horizontal tube were first studied by Beatty and Katz [1948] who made the condensation analysis on two-dimensional integral fin tubes with a rectangular cross section. In this geometry, the Nusselt analysis could not be directly applied to such tubes. They applied Nusselt's equations for the film

condensation on horizontal tubes and vertical plates to the horizontal portion between the fins and to the short vertical fins, respectively, and this has been modified by many others [Smirnov and Lukanov, 1972 and Marto et al., 1990 and so on.]. In the present study, the average condensation heat transfer coefficient correlation of Smirnov and Lukanov [1972] developed for the horizontal tube with fins is used in the simulation for simplicity given as:

$$h_{cond} = 0.689 \left\{ \frac{k_i^3 \rho_i^2 g h_{fg}}{\mu_i \Delta t D_{eq}} \right\}^{1/4} \quad (4.21)$$

$$\left\{ \frac{1}{D_{eq}} \right\}^{1/4} = 1.30 \eta_f \frac{A_{fs}}{A_{ef}} \frac{1}{L^{1/4}} + \eta_f \frac{A_{fs}}{A_{ef}} \frac{1}{D_0^{1/4}} + \frac{A_u}{A_{ef}} \frac{1}{D_r^{1/4}} \quad (4.22)$$

The details are also given in Appendix D .

4.5.4 Pressure Drop

4.5.4.1 Pressure Drop in Straight Pipe

(a) Pressure Drop Calculation for Single-phase

A TLT system encounters both single phase and two-phase flow regimes. The frictional pressure drop in a single phase flow regime is calculated using the following relationship for smooth pipes.

$$\Delta p_f = f \frac{G^2}{2\rho_i} \frac{\Delta z}{D} \quad (4.23)$$

$$\begin{cases} f = 16/\text{Re} & \text{for laminar} \\ f = 0.079/(\text{Re})^{0.25} & \text{for turbulent flow in } \text{Re} \leq 20,000 \\ f = 0.046/(\text{Re})^{0.2} & \text{for turbulent flow in } \text{Re} > 20,000 \end{cases}$$

(b) Pressure Drop Calculations for Two-Phase Forced Convection

There are a large number of pressure drop correlations for the lumped model for two-phase flows. In the simulation, the correlation of Friedel [Hewitt, 1994] was used to relate the two-phase frictional pressure drop to the single-phase frictional pressure drops for the vapor and liquid phases given as;

$$-\left(\frac{dp}{dz}\right)_p = \left(\frac{dp}{dz}\right)_o \left(E + \frac{3.24 FH}{Fr^{0.045} We^{0.035}} \right) \quad (4.24)$$

(c) Pressure Drop Calculations for Dryout Region

The method recommended by Ali and McDonald [1977b] for the pressure drop calculation for the region was adopted where the frictional pressure drop was calculated as a function of x .

$$\Delta p_{f,do} = (C \Delta p_{f,g} + D \Delta p_{f,p}) \quad (4.25)$$

where

$$C = \frac{(x - x_{cr})}{(1 - x_{cr})} \quad \text{and} \quad D = 1 - C \quad (4.26)$$

at $x = x_{cr}$ $\Delta p_{f, do} = \Delta p_{f, tp}$ and at $x = 1$ $\Delta p_{f, do} = \Delta p_{f, g}$. The pressure drop decreases linearly with the increase of x_{cr} between the above two limits. The hydrostatic and acceleration pressure drops were calculated in a manner similar to relations that described in the calculation of the two-phase flow pressure drop. In the present study, the dry-out region is not encountered as previously mentioned.

(d) Pressure Drop Calculation for Single-Phase Vapor Flow

The pressure drop was calculated from the relations as indicated in section '*single phase flow*' by using vapor properties because the flow is single phase as all vapor.

(e) Pressure Drop in Transport Section

In the transport pipe of a TLT, the flow was considered to be an annular flow with same entrainments which were deduced from the calculated quality of the flow at the outlet of the evaporator. The frictional pressure drop for the section is calculated from the same method as described in Section 4.5.4.1 (b) above based on the thermal and hydraulic state of the working fluid at the evaporator outlet.

(f) Pressure Drop in Condenser Section

- Vertical Film Condensation

The frictional pressure drop of the condenser section with vertical geometry is calculated by the method as described in Section 4.5.4.1 (a) as a single phase flow. It is assumed that the vapor flow from the transporting pipe is immediately liquidized in the condenser inlet.

- Horizontal Integrated fin structure.

It is very difficult to calculate the frictional pressure drop for a horizontal pipe with fin structure because the flooding and innundation are involved. To simplify the calculation, the simulation needed a few assumptions, “It is assumed that in condenser section, there is no flooding and innundation and the flow is single phase state”. For the frictional pressure drop in a horizontal condenser with integrated fin structure is calculated with the correlation of Lukonovich et al. [1972].

$$\Delta p_f = f_k \frac{G^2}{2\rho_l} \frac{\Delta y}{D_{eq}} \quad (4.27)$$

where D_{eq} is the equivalent diameter of the condenser.

(g) Pressure Drop in Condensate Return Line

The pressure drop in the condensate return line of a TLT is calculated as a single phase flow and the method described in Section 4.5.4.1 (a) is used.

4.5.4.2 Pressure Drop with Area Change Model (Expansion, Contraction, Orifice)

All TLTs contain piping networks with many fittings having, e.g., sudden area changes such as valves, junction and orifices. These fittings would encounter single phase fluid and/or two-phase flow. One example is described in Fig. 4.3 for LSL II.

(a) Single-Phase Abrupt Area Change Model

The modeling techniques used for the dynamic pressure losses associated with the abrupt area

change in a single-phase flow are extended to two-phase flows. In a steady incompressible flow, the losses at an area change are modeled by the inclusion of an appropriate dynamic head loss term, h_L , in the one-dimensional modified Bernoulli equation [Welty, 1984].

- Expansion

$$\Delta p_{\text{expansion}} = -2 \left(\frac{A}{A_b} \right) \left(1 - \frac{A}{A_b} \right) \left(\frac{G^2}{2\rho_l} \right) \quad (4.28)$$

- Contraction

$$\Delta p_{\text{contraction}} = \left\{ \left(\frac{1}{C_c} - 1 \right)^2 + 1 - \left(\frac{A}{A_b} \right)^2 \right\} \left(\frac{G^2}{2\rho_l} \right) \quad (4.29)$$

- Insert

The most general case of an abrupt area change is contraction and expansion with an insert. In this case, the insert is considered as a combination of the contraction and the expansion. The first is the contraction area ratio at the vena-contracta relative to the minimum physical area, $\epsilon_c = A_c/A_T$. The second is the ratio of the minimum physical area to the upstream flow area, $\epsilon_T = A_T/A_1$. The third is the ratio of the downstream to upstream area, $\epsilon = A_2/A_1$. The dynamic pressure loss through an insert is analyzed in a manner parallel to that for a simple contraction and expansion. The dynamic pressure loss associated with the insert is given by

$$\Delta p_I = \frac{1}{2} \rho_l \left(1 - \frac{1}{\epsilon_c \epsilon_T} \right)^2 v_2^2 \quad (4.30)$$

where $\epsilon_c = 0.62 + 0.38 (\epsilon_T)^3$

(b) Two-Phase Abrupt Area Change Model.

It is worthful to consider that many industrial piping systems have

– Sudden Expansion [Chisholm, 1983]

For the two-phase abrupt change model, the separated flow multiplier which is suggested by Chisholm [1973] is applied. This model follows from the separated-flow momentum equations when the effects of gravity and wall shear are neglected.

$$\psi_s = \left\{ 1 + \left(\frac{\rho_l}{\rho_g} - 1 \right) [Bx(1-x) + x^2] \right\} \quad (4.31)$$

where B is recommended by Chisholm [1973] as 0.25.

$$-\Delta p_{lo} = \frac{G_1^2 \zeta (1 - \zeta)}{\rho_l} \quad (4.32)$$

where ζ is the area ratio through the area change.

$$\zeta = \frac{A_1}{A_2} \quad (4.33)$$

Therefore, the frictional pressure drop through the sudden expansion is,

$$-\Delta p_{expansion} = \psi_s \Delta p_{lo} \quad (4.34)$$

– Sudden Contraction [Chisholm, 1983]

For a sudden contraction, the homogeneous model is used with homogeneous flow multiplier.

$$\psi_H = \left[1 + x \left(\frac{\rho_l}{\rho_g} - 1 \right) \right] \quad (4.35)$$

$$-\Delta p_{lo} = \frac{G^2}{2\rho_l} \left[\left(\frac{1}{C_c} - 1 \right) + 1 - \frac{1}{\zeta^2} \right] \quad (4.36)$$

$$C_c = \frac{1}{0.639[1 - (1/\zeta)]^{1/2} + 1} \quad (4.37)$$

$$-\Delta p_{contraction} = \psi_H \Delta p_{lo} \quad (4.38)$$

4.5.4.3 Flow Direction Change Model (Bend, Valve, Flow turning)

(a) Single Phase Flow

- Flow Direction Change [LSL I and II]

At the evaporator inlet, condenser inlet and outlet, the fluid flow encounters changes in the flow direction which is not extensively studied. Hence the technics which have been used in industrial heat exchanger designs [Hewitt, 1994] was also used in the present study.

$$-\Delta p_{nm\_sp} = k_{nm} \frac{(\dot{m} / A)^2}{2\rho_l} \quad (4.39)$$

– Bend

$$-\Delta p_{f-bend} = 0.000241 a_b \theta \left(\frac{G_b D_b}{\mu_l} \right)^{-0.17} \left(\frac{2R}{D_b} \right)^{0.84} \left(\frac{G_b^2}{2\rho_l} \right) \quad (4.40)$$

for 90° bend

$$a_b = \begin{cases} 0.95 + 17.2(2R/D_b)^{-1.96} & 2R/D_b < 19.7 \\ 1 & 2R/D_b > 19.7 \end{cases} \quad (4.41)$$

(b) Two-Phase Flow

– Flow Direction Change [Hewitt, 1994]

$$-\Delta p_{turn-tp} = k_{turn} \frac{\left(\frac{m}{A} \right)^2}{2\rho_l} \psi_H \quad (4.42)$$

– Bend [Chisholm, 1983]

$$\psi_s = \left\{ 1 + \left(\frac{\rho_l}{\rho_g} - 1 \right) \left[B_\theta x(1-x) + x^2 \right] \right\} \quad (4.43)$$

$$B_\theta = 1 + [B_{90} - 1] \frac{k_{90}}{k_\theta} \quad (4.44)$$

$$-\Delta p_{lo} = 0.15 \left[\frac{m^2}{2\rho_l} \right] \quad (4.45)$$

$$-\Delta p_{bend} = \psi_s \Delta p_{lo} \quad (4.46)$$

where for 90° bend,

$$B_{90} = 1 + \frac{2.2}{k_{BLO}(2 + R_C / D)} \quad (4.47)$$

k_{BLO} = the loss coefficient for the bend calculated for the total flow flowing with the liquid phase properties ($k_{BLO} = 0.15$ for 90° bend)

R_C = the bend radius of curvature

D = the tube diameter

For 180° bends

$$\frac{k_{90}}{k_{180}} = \frac{1}{2} \quad (4.48)$$

For other angle bends

$$0 < \theta < 90^\circ$$

For the bends with angles less than 90°, it was recommended that the B coefficient is taken as that for a 90° bend.

$$90^\circ < \theta < 180^\circ$$

For the bends with angles greater than 90°, it was recommended that the B coefficient is taken as that for a 90° bend.

4.5.5 Void Fraction

The void fraction correlation used is that of Lockart and Martinelli given in Appendix E.

4.6 Sectorial Method

4.6.1 Simulation Logic

It is generally accepted that the best simulation method for a TLT system is the sectorial method (see Fig. 1.2) which is also called as the flow pattern model. The sectorial method is seen to be much more realistic and sophisticated approach to a two phase flow. It is based on an idealized representation of various possible flow phase configurations or flow pattern into which the two-phases are considered. The success of the sectorial method is, however, how empirical correlations available in the literature can be arranged and managed as was the case for the lumped method. The majority of analysis is made in terms of force and energy balances existing in a particular flow patterns make such a treatment extremely difficult. Attempts have, therefore, mainly been confined to the annular pattern, due to its structural simplicity compared with other flow patterns. An intensive attention was devoted to slug flow since this flow pattern is inherently unsteady with large time variation of the mass flow rate, pressure and velocity at any cross section normal to their direction and poses therefore special and difficult problems for the design if this condition is to be avoided. In practice, the sectorial models have been often used in conjunction with some numerical method to define the flow pattern and the extent of the flow regime or other transitions from one to another.

However, the sectorial method encounters a large number of programming difficulties much more than the lumped method. Unlike the lumped simulation method for a two-phase loop system, the simulation code based on the sectorial method includes many generic component models (the flow pattern models, the flow transition models, the pressure drop models and the heat transfer models etc.) from which general systems can be simulated.

As describe above, two-phase flow problem has a large number of unknown variables, according to Hsu, Y. [1976]. With these unknown variables, two-phase flow problem can be solved analytically by 12 equations (energy, momentum and mass equations) with many assumptions which are impossible to obtain only exact solution of the two-phase flow problem.

Unlike the lumped simulation method of TLT systems, the sectorial simulation method requires a large number of empirical correlations for generic component models. The heat transfer in the evaporator section of a TLT was calculated as one lumped heat transfer area in the lumped method. But in the sectorial method, the evaporator section of a TLT is subdivided into each flow regime. Also, for these flow regimes, different heat transfer correlations and pressure drop correlations are needed.

As was for the lumped method, the sectorial method are based on the thermal resistances involved in a TLT system. In the lumped method, the thermal resistance for the evaporator of a TLT is defined as that of one lumped heated section. However, in the sectorial method, the thermal resistance, R_{loop} , must be determined through various flow regimes involved in the evaporation phenomena. The sectorial simulation program starts with entering of the working fluid at the bottom of the evaporator. A pre-assumed iteration set of the inlet condition will be set into the program start. This pre-assumed inlet condition is reset by ensuing iteration procedures. The description of two-phase flow in the loop is complicated by the existence of an interface between the two phases. The working fluid flows in a TLT system, not only in a vapor phase but also in a liquid phase flow in the same direction, continually changing its flow regime along the loop. Therefore, the two-phase flow and heat transfer involved are very complex and it is very difficult to explore each heat transfer characteristics respectively, according to the different flow regimes. When the returning condensate from the condenser of a TLT enters the evaporator, it is heated and the flow regime may change to

bubble flow. In simulation, the inlet condition is checked first with some criterion such as the criterion for the existence of the bubbly flow. If the bubbly flow exist, two-phase flow parameters in this flow regime are calculated. Consequently, increasing void fraction and flow velocity encounters the next flow patterns. When the rise velocity of bubbles in the bubbly regime exceeds the velocity of Taylor bubble, it is considered that the bubbly flow can not exist [Taitel et al., 1980]. If the flow satisfying the transition criteria from bubbly to slug flows, a check has to be made for the consequent transition criteria, i.e ; slug to annular transition and annular to drop flow transition. If the state of the two- phase flow is placed in only slug flow, then slug flow regime starts and the calculation of two-phase parameters in this regime proceeds.

In the present simulation study, the vertical flow-regime map [Taitel et al., 1980] combines slug and churn flow-regimes into a single regime called slug flow with the first criterion controlling the transition in small tubes and the second criterion applying in large tubes. The transition from the slug flow to the annular flow is checked by Taitel's transition criteria [Taitel et al., 1980]. If the transition criteria from the slug to the annular is satisfied, then the next transition criteria is checked as in the previous procedure. Consequently, in the evaporator of a TLT, the flow changed to annular flow. In the annular flow, the gas flows along the center of the loop tube. The liquid flows partially as a film along the walls of the tube, and partially as droplets in the central gas core. If only the transition criteria from the slug to the annular flow is satisfied, then the calculation procedure of the annular two-phase parameters is proceeded. The next step is followed by the same procedure. Finally the evaporator outlet condition will be met. Furthermore, because the transitions between the various flow regimes do not occur suddenly, it is possible to observe a number of transition flow patterns which possess characteristics of more than one of the main flow patterns described above. With a steady state assumption, mainly two or three flow patterns will be observed such as bubbly and slug,

bubbly and annular along the evaporator .

Since there could be no absolute number of flow regimes exist, in the present sectorial simulation method, only four patterns, i.e., bubbly, slug, annular and drop flows, are considered because of the lack of comprehensive analytical and empirical analyses. For the flow transitions between one flow pattern to other flow patterns, the vertical volume flow-regime map is considered, available in open literature.

The individual heat transfer and pressure drop models for bubbly, slug, annular, and drop flow regimes are discussed in the following sections. Various models for the transition regions for the regimes are also discussed.

The heated working fluid in the evaporator section flows into the connecting transport pipe with increased phasic velocity, quality and void fraction. The state of the inlet fluid flow into the transport pipe of a TLT is considered to be an annular flow.

The pressure drop along a TLT is calculated using adiabatic annular flow pressure drop correlations. For a transport pipe, it is assumed that the transport pipe is well insulated to prevent heat loss. The working fluid from the transport pipe reaches at the condenser inlet in the state of saturation. The vapor is condensed on the condenser tube inside surface. The condensation heat transfer coefficient is calculated and the pressure drop calculated as a single phase liquid state. The condensed liquid flows into the condensate return line and condensed liquid accumulates in the return line pipe. This accumulated liquid pushes the working fluid to flow into the evaporator section, and the cycle repeats.

The pressure drop along the loop is calculated with the condition of the working fluid state (single phase or two-phase).

To obtain the convergence of the program, the convergence routine is provided in the calculation procedure between the evaporator inlet and the end of condensate return line. Mainly, the sectorial simulation is considered converged when the simulation iteration procedure satisfied energy, mass and momentum balances. The simulation must also satisfy that the total pressure drop summation through the TLT system is zero. The simulation flow charts for single and multi pipe TLTs by the sectorial methods and given in Fig. 4.2 (c) and Fig. 4.2 (d), respectively.

4.6.2 Sectorial Model based on Flow Patterns

4.6.2.1 Flow Pattern Map

When a two-phase flow occurs in tubes, it is important to be able to predict which flow pattern is likely to occur for any given combination of phase properties and flow rates, and for any given tube diameter. Flow pattern predictions are usually obtained from flow pattern maps, of which there are two types as described below. In the present study, the flow pattern map and transition methods developed semi-experimentally by Taitel et al. [1980], Ishii et al. [1984] and others [Hewitt and Roberts, 1969, Mishima and Ishii, 1980, Ishii and Grolmes, 1975, etc.] are adopted. These flow pattern maps were developed from geometrically horizontal orientations, but an unified model was further developed so that it could become also applicable for various other geometries. The flow pattern maps and transition criteria by these workers are given in Appendix. A. The major problems of the flow pattern maps developed by Hewitt and Roberts [1969], Taital et al. [1980] and Ishii et al. [1980] are that they are for the two-phase flows of adiabatic, large mass flow rate and high pressure as shown in Table 4.3. The flow pattern maps used widely are also shown in Appendix A.

4.6.2.2 Flow Regimes Considered in Simulation

To describe two-phase flow phenomena in tubes is very difficult because a two-phase flow is complicated by the existence of an interface between the two phases. This interface for gas-liquid two-phase flows exists in a wide variety of forms. This interface depends on the flow rates and physical properties of the phases, on pressure, and on the geometry and inclination of the tube. Although all investigators agree that the flow pattern has a major influence on two-phase (gas-liquid) transport phenomena, the influence of flow pattern is quite often ignored. For the particular case of the upwards flow in vertical tubes, we consider only the flow patterns of bubbly, slug, annular and drop as previously discussed. These are illustrated in Figure 1.2 and their main features are briefly described below.

- Bubbly flow : In a bubble flow, the vapor phase flows as discrete bubbles in a liquid continuum. The bubbles are usually distorted spheres.
- Slug flow : When the bubble concentration in bubble flow becomes high, bubble coalescence occurs and the largest bubbles are of the same order of size as the tube diameter. Further coalescence results in the deformation of the bubble into the bullet shaped pocket of vapor which is characteristic of slug flow. Slug flow then consists of these pockets of vapor, commonly called plugs or Taylor bubbles, separated by regions of bubbly flow, commonly called slugs. The slugs of vapor are surrounded by a thin liquid film which flows vertically downwards. The nose of the bubbles has spherical shape.
- Annular flow : In annular flow the vapor flows along the centre of the tube. The liquid flows partially as a film along the walls of the tube, and partially as droplets in the central vapor core.

- **Drop flow** : Unlike the unheated tubes, in the case of a vertical heated wall where any liquid at the wall is evaporated does true drop flow occurs.

It is possible to extend the above description of flow patterns. For example, the annular flow regime may be sub-divided into wispy and non-wispy annular flows, with wispy annular flow occurring as a result of the agglomeration of the liquid droplets in the vapor core into large streaks or wisps. Furthermore, because the transitions between the various flow regimes do not occur suddenly, it is possible to observe a number of transition flow patterns which possess characteristics of more than one of the main flow patterns described above.

4.6.2.3 Flow Regime Transitions

The topic of flow pattern transitions from one flow pattern to another is still a matter of active discussion. Even most researchers are trying to make a generalized flow pattern map for vertical and horizontal two-phase flow, the flow pattern transition is still unsolved phenomena. Therefore, the maps given in most open literatures should be used as a general guidance only and, if there is a crucial interest in any specific transition, then a closer study should be made. The flow regimes and flow transitions which are considered in the present study are described in Table 4.4.

(a) Criterion for the Presence of Bubbly Flow [Taitel et al., 1980]

At the beginning of a simulation for a TLT, the flow state is checked for the given inlet temperature. The following criterion will check the minimum evaporator pipe diameter where a bubbly flow may exist.

$$D > \frac{2.34 \sin^2 \theta}{(0.35 \sin \theta + 0.54 \cos \theta)} \left[\frac{(\rho_l - \rho_g) \sigma}{g \rho_l} \right]^{0.5} \quad (4.49)$$

Dispersed bubbly flow will exist if the stable maximum diameter of the dispersed bubbles, $d_{b,max}$, is less than the critical bubble diameter, $d_{b,crit}$.

$$d_{b,max} = (0.725 + 4.25\beta^{0.5}) (\sigma/\rho_l)^{0.6} \left(\frac{2f_w}{D} j^3 \right)^{-0.4} \quad (4.50)$$

$$d_{b,crit} = \left[\frac{0.4\sigma}{(\rho_l - \rho_g)g} \right]^{1/4} \quad (4.51)$$

(b) Bubbly Flow to Slug Flow Transition [Taitel et al., 1980]

The transition from a bubbly flow to a slug flow occurs as a result of bubble coalescence leading to gradual bubble growth and the formation of large Taylor type bubbles which occupy the whole evaporator cross-section. Bubbly flows may not generally exist in tubes of small diameter where the rise velocity of small bubbles exceeds that of the Taylor bubbles as shown in Eq. (4.52). The small bubble rise velocity is given by the correlation given as Eq. (4.53) below. Typically, the transition to slug flow occurs when the void fraction is around 25-30% [Taitel et al., 1980]. The bubbly to slug flow transition model suggested by Taitel et al. [1980] is:

$$v_{tb} = 0.35D^* \left[\frac{\sigma}{D\rho_l} \right]^{1/2} = 0.35 \left[\frac{gD(\rho_l - \rho_g)\sigma}{\rho_g} \right]^{1/2} \quad (4.52)$$

and the Taylor bubble rise velocity is:

$$v_{tb} = 1.53 \left[\frac{D^* \sigma}{D\rho_l} \right]^{1/2} = 1.53 \left[\frac{g(\rho_l - \rho_g)\sigma}{\rho_g^2} \right]^{1/4} \quad (4.53)$$

Then, the criteria of the transition from bubbly flow to slug flow is:

$$v_{sb} > v_{tb} \quad (4.54)$$

This criteria is supposed to be valid in the range of $\alpha = 0.25$ [Taitel et al., 1980].

(c) Slug to Annular Flow Transition [Taitel et al., 1980 and Mishima and Ishii, 1984]

According to Taitel et al. [1980] and Mishima and Ishii [1984] indicate that the annular flow transition is principally governed by the criteria given as:

$$j_g^* = \frac{\alpha_g v_g}{\left[\frac{gD(\rho_l - \rho_g)}{\rho_g} \right]^{1/2}} \geq j_{g,crit}^* \quad (4.55)$$

$$Ku_g = \frac{\alpha_g v_g}{\left[\frac{g\sigma}{\rho_g^2} \right]^{1/4}} \geq Ku_{g,crit} \quad (4.56)$$

with the first criterion (flow reversal) controlling the transition in small tubes, and the second criterion (droplet entrainment) applying in large tubes. Unfortunately, the data reported by the authors are not sufficient to make any reasonable judgment as to the most appropriate values of $j_{g,crit}^*$ and $Ku_{g,crit}$.

$$j_{g,crit}^* = 1 \text{ and } Ku_{g,crit} = 3.2. \quad (4.57)$$

This criteria is satisfied in the range of $\alpha = 0.52$ [Taitel et al., 1980].

- (d) The Transition to Drop Flow, the value of α_{A-D} is [Taitel, 1990]

$$\alpha_{A-D} = 0.9999 \quad (4.58)$$

4.6.2.4 Programming Implementation

In the simulation programs, the flow pattern transition criteria from a open literatures are provided. As has been discussed, the flow transition phenomena in a two-phase flow are very complex and important subjects which may be further investigated. The models in the literature are mostly for very narrow experimental conditions (pressure, flow rate and geometrical factors).

4.6.3 Heat Transfer

The assessment of the correlations for heat transfer coefficients used is complicated by the complex nature of the correlations themselves. In general, each correlation is designed to represent heat transfer under a specific set of thermal-hydraulic and thermodynamic conditions, and each is typically measured for a fairly limited range of those conditions. A determination of accuracy may be available for the developmental range of parameters, but an extension of the accuracy estimate outside that range is difficult at best, and perhaps impossible physically. By treating each correlational model individually, a critical reviewer might generally conclude that the database over which the model was developed does not apply directly to the geometries or thermal-hydraulic conditions of the present interest. The use of any given heat transfer correlation, either directly or in a modified form, is an engineering judgment, and the use of integral assessments becomes meaningful.

4.6.3.1 Two-Phase Boiling

(a) Bubbly Flow:

A bubbly flow is defined as a two-phase flow where small bubbles are dispersed or suspended as discrete substances in a liquid continuum. Typical features of this flow are moving and deformable interfaces of bubbles in time and space domains and complex interactions between the interfaces, and also between the bubbles and the liquid flow. The heat transfer coefficient correlation of Bohdal et al. [1999] given below was used in the simulation study.

$$h_{bub} = A \left(\frac{k_l}{D} \right) (Re)^{2.42} (Fr)^{-0.35} (Ku)^{1.38} (Bo)^{-1.08} \quad (4.59)$$

(b) Slug Flow :

The heat transfer characteristics of the slug flow region are very different from those of the annular flow region. Flow boiling heat transfer at low qualities, low mass and heat fluxes, where convective heat transfer associated with slug dominates is important for a TLT system. The heat transfer characteristics under these conditions are significantly different from those exhibited at higher quality and higher mass flux. In spite of its significance and the different heat transfer characteristics, very little published information exists on the subject.

The correlation for the heat transfer coefficient correlation of slug flow by Wadekar and Kenning [1996] is used in the present simulation.

$$h_{slug} = h_p \left[\frac{L_p}{L_p + L_s} \right] + h_s \left[\frac{L_s}{L_p + L_s} \right] \quad (4.60)$$

(c) Annular Flow :

Vapor-liquid annular two-phase flow regime is frequently encountered with flow boiling in a vertical boiling tube. In a vertical tube evaporator, as much as 90 % of the tube length may be in annular flow [Collier and Thome, 1994]. In this region, the fluid flow in a tube could be separated into the liquid film flowing adjacent to the wall and the core vapor flow with liquid droplets entrained. The heat transfer coefficient can be predicted from the estimation of velocity and temperature profiles across the two fluid layers.

It is very difficult to choose the heat transfer correlations for annular flows because the general two-phase flow heat transfer correlations used in the lumped method are originally developed from the annular flow model and mostly from empirical or semi-empirical base. During last few decades, these correlations are often used in various conditions and geometries without proper justification. In the present sectorial method, the correlation of Lavin and Young [1956] and Rhee and Young [1974] are tested in the simulation, and the relevant equations are given in Appendix C.

(d) Drop Flow :

When the vapor quality reaches a certain critical value, the amount of liquid present becomes insufficient to maintain a stable film on the walls. This is the condition referred to as two-phase burn-out or dry-wall condition. The heat transfer mechanism of this drop flow regimes has been considered as a single phase forced convection to a vapor stream with migration of droplets to the wall with vaporization on contact. In present study, the empirical equation by Lavin and Young [1956] is used in the simulation study, which is given as:

$$h_{drop} = \frac{k_g}{D} 0.0162(Re)^{0.84} (Pr)^{0.33} (1-x)^{0.1} \quad (4.61)$$

4.6.3.2 Programming Implementation

In the present simulation study, the annular - drop flow transition is not encountered, nor observed in the present experimental program. The drop flow heat transfer coefficient correlation given above was in the simulation computer program only as a reference.

4.6.3.3 Condensation

For the film condensation inside vertical pipes, the calculation of condensation heat transfer coefficient correlation is the same as that was used for the lumped method.

4.6.4 Pressure Drop

4.6.4.1 Pressure Drop in Straight Pipe

The influence of the flow pattern on the frictional pressure drop in a vertical up-flow has been considered to be rather insufficient that its importance is overshadowed by the accuracy of the measurement. Moreover, it is very difficult to find the frictional pressure drop correlations of each flow regime. In the present study, the following flow patterns only will be considered, i.e. single phase regime, bubbly flow, slug flow, annular flow with no entrainment, annular flow with entrainment and drop flow. The sectorial model based on flow patterns can be applied to the simulation. However the coherence of this approach is restricted as the classification of the specific flow regimes is still an unsolved problem.

To obtain the friction pressure drop in heated channels where the qualities vary axially, the common practice is to use the correlations of a particular flow regime, and to integrate the pressure drop between the points of interest taking into account the variations in the local quality. But this method is just stopgap and strictly speaking not correct because all correlations do not correctly

describe the physics of the processes involved, particularly the departure from the conditions of hydrodynamic equilibrium to which the adiabatic correlations may be presumed to approximate, are not taken into account. However, the average pressure drops predicted for diabatic situations are often not affected significantly by the inequality, and the latter assumption, therefore, is reasonable if the variation of the frictional gradient with quality is approximately linear and/or the quality difference between the two points is relatively small.

(a) **Pressure Drop in Single Phase Regime**

The calculation method for this regime is the same that used in the lumped method.

(b) **Frictional Pressure Drop in Two-Phase Boiling Section**

- **Bubbly Flow**

Bubbly Flow Frictional Pressure Drop Correlation [Beatti, 1973]

$$\phi_{lo}^2 = \left[1 + x \left\{ \frac{\rho_g}{\rho_l} - 1 \right\} \right]^{0.8} \left[1 + x \left\{ \frac{(3.5\mu_g + 2\mu_l)\rho_l}{(\mu_g + \mu_l)\rho_g} - 1 \right\} \right]^2 \quad (4.62)$$

$$-\left(\frac{dp}{dz}\right)_{b-f} = \left(\frac{dp}{dz}\right)_{lo} \phi_{lo}^2 \quad (4.63)$$

- **Slug Flow [Chisholm, 1983]**

Slug Flow Frictional Pressure Drop Correlation

$$-\left(\frac{dp}{dz}\right)_{s-f} = \frac{2f_{lo}G^2}{D\rho_l}(1-\alpha) \quad (4.64)$$

- **Annular Flow**

Annular flow frictional pressure drop is intensively depends on the annular flow state between liquid film and vapor interface. The amount of liquid entrainment fraction is can be very significant according to the geometrical features, working fluid and system operating conditions. It was seen that the pressure fall abruptly as the liquid entrainment increases. This effect is related directly to a loop system and subsequently it is possible that the loop system encounters instability problem. Hence the liquid entrainment model [Ishii and Mishima, 1982] was applied to the simulation, which is given in Appendix K. This process has an interrelationship with the instability analysis discussed later.

Liquid Entrainment Equation [Ishii and Mishima, 1982]

$$e = \tanh \left[7.25 \times 10^{-7} (j_g^+)^{2.5} (D^+)^{1.25} (Re_l)^{0.25} \right] \quad (4.65)$$

$$-\left(\frac{dp}{dz} \right)_{a-f} = \frac{\rho_g f_{a-f} V_g^2}{2D} \quad (4.66)$$

- **Drop Flow:**

In the drop flow regime, the pressure drop was considered as a single phase vapor flow.

(c) **Pressure Drop in Transport Section**

It was observed from experimental study that the flow pattern in the transport section of a TLT is dominated by annular flow. Hence, in the simulation, the pressure drop calculation model used for vertical adiabatic annular flow was adopted, which is given in Appendix K.

(d) **Pressure Drop in Condenser Section**

The pressure drop calculation method of the condenser is the same as the lumped method.

(e) **Pressure Drop in Condensate Return Line**

The pressure drop in the condensate return line of a TLT is calculated as single phase pressure drop which is the same as that for the lumped method.

4.6.4.2 Pressure Drop with Area and Flow Direction Change Models

The pressure drop calculation of the sectorial method through the sudden area changes (such as valves, junction and orifices) and the flow direction changes is the same as that for the lumped method.

4.6.5 Void Fraction

It is necessary to predict the void fraction or liquid hold up for the evaluation of the elevation and friction pressure drop. Many empirically and theoretically derived models for the lumped case are available, but an extensive search for the sectorial case has revealed that none of the correlations are generally applicable and the accuracy is limited.

Void fraction is affected by a variety of variables including fluid properties, mass flow rate, quality, loop geometry, heat flux If any, etc. A rather strong influence is also set up by the flow pattern making it therefore desirable to express the void fraction individually even for each flow pattern. However, to do so at the present time, would probably only complicate the calculations and introduce another source of error to the computation method which at best is less than 10 to 20 % accurate in the low quality region.

Collier [1995] recommended the drift flux model for bubbly and slug flow and for annular flow regime, Ishii's model is applied to the simulation.

The drift flux model is given as Eq. (4.67), and the correlations of bubbly and slug flow regimes depend on C_0 , and $\bar{u}_{g,j}$.

$$(\alpha) = \frac{(\beta)}{C_0 + \bar{u}_{g,j} / (\bar{j})} \quad (4.67)$$

$$\beta = \frac{1}{1 + [(1-x)\rho_g / x\rho_l]} \quad (4.68)$$

$$j = u_{jo} \left[1 + x \left(\frac{\rho_l}{\rho_g} - 1 \right) \right] \quad (4.69)$$

(a) Bubbly Flow [Zuber Findlay, 1965]

$$C_0 = 1.0, \quad \bar{u}_{g,j} = 153(1-\alpha)^2 \left[\frac{\sigma g (\rho_l - \rho_g)}{\rho_l} \right]^{1/4} \quad (4.70)$$

(b) Slug Flow [Zuber Findlay, 1965]

$$C_0 = 1.2, \quad \bar{u}_{g,j} = 0.35 \left[\frac{Dg(\rho_l - \rho_g)}{\rho_l} \right]^{1/2} \quad (4.71)$$

(c) Annular Flow [Ishii, 1982]

$$C_0 = 1.0, \quad \bar{u}_{s,j} = 123 \left(\frac{\mu_l j_l}{\rho_g D} \right)^{1/2} \frac{(\rho_l - \rho_g)}{\rho_l} \quad (4.72)$$

4.7 Geometrical Considerations

The five TLTs studied have different geometries and in the simulation, the geometrical differences of TLTs was considered individually. The details is given in Appendix L.

4.8 Effectiveness

It is logical to define the effectiveness in order to predict the performance of TLT system, The effectiveness, ε , is defined as the ratio of the actual heat transfer rate for a heat exchanger to the maximum possible heat transfer rate [Mills, 1995].

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} \quad (4.73)$$

The maximum heat transfer in an exchanger of any exchanger configuration (Parallel or Counter Flow) is defined as

$$\dot{Q}_{\max} = C_{\min} (t_{h,m} - t_{c,m}) \quad (4.74)$$

where

$$C_{\min} = \min[C_h, C_c] \quad (4.75)$$

$$C_h = \dot{m}_h c_p \quad (4.76)$$

$$C_c = \dot{m}_c c_p \quad (4.77)$$

Then the effectiveness of any exchanger configuration is

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_h(t_{h,in} - t_{h,out})}{C_{\min}(t_{h,in} - t_{c,in})} = \frac{C_c(t_{c,out} - t_{c,in})}{C_{\min}(t_{h,in} - t_{c,in})} \quad (4.78)$$

4.9 Fin Efficiency

The heat transfer from a prime surface can be increased by attaching fins or extended surfaces. Use of a finned cooling surface is particularly effective in the case where the heat transfer coefficient between the wall and one fluid is substantially greater than the heat transfer coefficient to the other fluid. Thus, for example, fins on the outside of pipes are very effective when liquid flows inside the pipe and vapor flows on the outside. Fins are also used on the liquid such as in the case of condensing vapors having low heat transfer coefficients.

The design method of for the cooling fins of various shapes is based on the same principles as those used for calculating the cooling capacity of rods of constant cross-section. The greater the distance from the base of the fins, the smaller the heat flow, so that for a constant cross section, the temperature gradient decreases along the fin length. Theoretically, if the heat transfer coefficient is constant, a fin bounded by a pair of parabolas is the most efficient one [Hewitt, 1994]. In practice, a fin of such shape is too costly to manufacture, thus it is usual to design fins having trapezoidal sections. The fins used in the present study is basically of thin rectangular shape, attached to the cylindrical base of pipes.

The fin efficiency of circular or rectangular fins attached to a TLT involves Bessel functions, which can usually be represented in tabular and graphical forms. For the analysis and simulation of

a TLT, approximate, but reasonably accurate empirical expressions presented by McQuiston and Tree [Hewitt, 1994] were used. The equations for the fin efficiency are given in Appendix I.

4.10 Thermal Properties

As stated in the reference [Lee et al., 1996], the thermal properties of the working fluids, which are used in the computer simulation, such as the density, viscosity and pressure versus saturation temperature, were calculated using the generally accepted method and approximations given in Reid [1966]. These calculations were necessary because of the lack of published data, especially for FC-72 and FC-87. Results of calculations involving FC-72 and FC-87 when compared with a few known thermal properties showed about $\pm 10\%$ difference [Lee et al., 1966]. The thermal properties of other working fluids are formulated into equations using a data regression for the temperature range of $-50\text{ }^{\circ}\text{C}$ to $300\text{ }^{\circ}\text{C}$. The analysis of these equations for the thermal properties of each working fluid showed the difference of about $\pm 5\%$. All equations related to thermal properties are shown in Appendix M.

4.11 Operating Limit

In a TLT, heat is transported upward by the vapor generated in the heating section. The vapor flows up through the connecting transport tube and condenses in the cooling section. The condensate is then returned to the heating section by gravity. However, when the heat-transport rate is increased greatly, a limiting point may be reached where a sharp rise in the wall temperature or sharp deterioration in the heat transfer coefficients takes place in the heating section. This performance limit can be classified into two types.

The first type of the performance limit occurs at very low liquid fill charges. There is a

required minimum quantity of working fluid for a TLT to have a continuous circulation of vapor and condensate in accordance with the heat transport rate. If the quantity of working fluid is less than the required minimum, the returning liquid may not reach the heating section in time, resulting in a dry-out of the heating section, or the wall temperature slowly rises. Thus, this type is usually called the dry-out limit.

The second type occurs when the heating surface is covered by vapors. Vapor bubbles are generated in the liquid pool of the evaporator section, and this nucleate boiling becomes more intense with increasing heat flux. At a certain critical radial heat flux, individual vapor bubbles are combined to form rather quickly a vapor film at the wall. This vapor film insulates the evaporator surface from the evaporating liquid. Owing to the poor thermal conductivity of the vapor, only part of the heat input to the wall is transfer to the fluid and there is a sudden increase of the evaporator wall temperature. This boiling limit is called burn-out limit. This limit should prevail for relatively large liquid fill charges. The two limits are included in the simulation code.

4.12 Simulation Interface Windows

The interface windows in the simulation code developed for TLT are based on that used for the TCT of Rhi [1996]. The simulation code developed for a TLT consists of four windows (interfaces generated); the main window, the thermosyphon design window, the processing window and the toolbox window. Each interface window has been designed for easy use of the code. Selected interface windows are given in Appendix N.

4.13 Instability of Two-Phase Closed Loop Thermosyphon

4.13.1 Nature of Instability

The fluid flow in a TLT is created by buoyancy forces as a result of the differences in fluid density induced by the temperature variations between the hot section (evaporator) and the cold section (condenser). Hence, the fluid in the vicinity of the heat sink is cooled and becomes more dense, thus tends to fall downward. Conversely, the fluid in the vicinity of the heat source is heated causing it to become lighter and rise upward. As a result of this motion, the temperature distribution is altered and the total buoyancy force is correspondingly changed. Nevertheless, as long as this force continues to act in the same direction, the fluid velocity continues to increase. This process does not go on indefinitely, however, since a retarding force due to viscous friction is developed. This force acts in the direction opposite to the direction of the fluid motion, and its magnitude increases as the fluid velocity increases. When the velocity has become large enough so that the frictional force becomes equal to the buoyancy force, the net force acting on the fluid in the direction of flow is zero. Hence, there is no further acceleration of the fluid, and the flow rate remains constant thereafter under steady applied thermal conditions at the heat source and heat sink. The fluid entering the heated flow channel always has the constant temperature while the velocity of the stream is fixed by the density difference between the hot section and the cold section. Under certain conditions this system may be unstable; that is, a small deviation from the equilibrium temperature distribution or the equilibrium velocity may be propagated in space or time with increasing amplitude. This description illustrates the normally expected operation of a loop. Nevertheless, the mechanisms described above are adequate for the purposes of an initial discussion of the operation of a TLT.

Due to the simplicity of the one-dimensional equations of conservation of mass, momentum and energy for a single-phase fluid, the corresponding flow and thermal behavior of a TLT are easily

described analytically provided that steady state conditions apply.

A TLT mainly consists of the evaporating section, the transport section, the condensing section and the returning section of the condensed working fluid. The buoyancy force is the main driving force of TLTs. The void distributions in turn are functions of the local heating and cooling rates and the fluid velocities within the loop. As shown in Fig 4.4, the pump head is the buoyancy head for a TLT. The oscillatory behaviour of the flow is caused by the system seeking a stable operating point. In doing so, the system tends to both over and undershoots the equilibrium point. The frictional shear stress results in a unstable state at a buoyancy force domain of transport section [Hsu, 1976]. In a TLT flow system, especially, in the transport section, flows the annular flow phenomena is dominant. The flow pattern can be changed to bubbly, slug, and annular according to the flow rate. The annular flow pattern can change to dispersed annular flow with an interfacial instability between the liquid film and the vapor flow, and finally to just single phase flow of vapor.

4.13.2 Two-Phase Annular Flow

As stated in Chapter 1, the two-phase flow in TLT system is dominated by annular flow over 90 %. Annular two-phase flow can be characterized the flow in which the interface between the phases is not smooth but covered with a complex pattern of waves. So far it has been reported that interfacial waves exist over the whole range of conditions. The waves observed vary widely in wavelength and in amplitude, some amplitudes being several times greater than the mean film thickness, and this waviness of the interface dominates a number of important phenomena in an annular flow. Although the effect of waves to the pressure drop is not as yet clearly understood, it is clear that the pressure drop is inevitably strongly dependent on the waves.

Experimental and theoretical studies of wave behaviour and their influence on other

phenomena in annular flow are still at a very early stage of development. And, therefore, in this present study, the Kelvin-Helmholtz interfacial instability approach is applied to the two-phase annular flow region of the transportation section of the TLT systems as *an attempt* to obtain same quantitative results.

4.13.3 Analysis of Instability

The major point of the Kelvin-Helmholtz instability applied to a TLT is that the outlet velocity of the fluid from the evaporator section must be higher than the critical velocity to sustain stable annular flow in the transport section. The outlet velocity of the fluid can be prevented to rise by the effect of surface tension or liquid entrainment etc.

Therefore, the major parameters concerning the analysis are the outlet velocities of the two-phase flow from the evaporator, u_g and u_l . The density and velocity conditions vary markedly along the evaporator section of a TLT. The density is highest at the evaporator inlet and the flow velocity is highest at the evaporator outlet. The effect of a small perturbation in the inlet velocity is to change the whole distribution.

One method of representing the system, which has the attraction of a simple physical interpretation, is to imagine a feedback loop in which a change in the applied pressure drop produces changes in velocity and density which are initially resisted only by the inertial effects. When the velocity and density change, they produce feedback effects via friction, hydrostatic, and momentum effects. This scheme has other problems of interpretation, however, particularly at zero frequency since the inertial terms are then of zero amplitude.

The effect of a change in the pressure drop is considered to increase the single-phase velocity after allowing for its inertia and friction, and the repercussions on the pressure drop resulting from

movement of the boiling boundary, two-phase region and superheat region are regarded as the feedbacks.

4.13.4 Kelvin-Helmholtz Instability in a Vertical Pipe [Hewitt, 1970]

The Kelvin-Helmholtz instability theory is well described in Ref. [Hewitt, 1970]. This section will summarize briefly the theory and propose an idea to be pursued. The Kelvin-Helmholtz instability was developed to define the interface phenomenon between two inviscid fluids of different densities. The literature applied to two-phase flow is very rare and it appears that none applied for any TLT systems. The instability resulting from the pressure drop in a TLT must be from the annular flow state because if an oscillatory flow rate is to be possible, the driving force can not be generated in the evaporator section but in the vertical riser above the evaporator. Any kind of forces and state which can prevent the flow can induce a pressure drop and flow unstable.

The annular flow state in a pipe is expressed as a mixed state between the vapor flow and the liquid flow. Consider the basic flow of two incompressible inviscid fluids of different velocities and densities separated by a vertical interface in a TLT transport section on which a regular train of waves of wavelength, λ_w , are moving with velocity, c , as shown in Fig. 4.5. To illustrate the mechanisms and concepts of this kind of instability, it is needed to work through a classic problem that demands not much of mathematics. For a mathematical solution, a problem can be simplified as following;

- By considering the action of the two phases separately, the first simplification is to evaluate the forces exerted by the gas phase on the wavy boundary and then the effects of these on the liquid phase are assessed.

- A second simplification is to consider the phenomenon from a reference frame moving with the wave. This reduces the problem to the equivalent one of flow past a stationary boundary.

Figure 4.5 shows the streamlines of a flow with a mean velocity of, $\bar{u}_g$, in the vapor phase and, $\bar{u}_l$, in the liquid. It can be noted that the amount of distortion to the streamlines decreases with increasing distance from the interface. As the vapor flows around the curves of the streamlines, centrifugal forces are set up and these must be balanced by a pressure gradient in the direction normal to the streamline. Since the vapor velocity is roughly the same at all points, the magnitude of the pressure gradient will decrease with increasing distance from the interface in accordance with the decrease in the curvature of the streamlines, ultimately falling to zero at infinity, where the pressure and velocity fields are undisturbed.

In the annular flow in a TLT, the liquid flows in a thin film enclosing the vapor phase in a cylindrical core. It is interesting to consider the effect of this change of geometry has on the stability problem, assuming the mean velocities of the two phases and the shape of the interface remain unchanged.

Considering a vertical upward co-current flow, the surface tension is the only force opposing to normal stress. The wave will grow in amplitude when the sum of the local liquid and vapor normal stresses exceed the surface tension stress. This implies that the interface is stable when the surface tension stress exactly balances the effects of the normal stresses, and is unstable in the opposite case. For the vapor phase, the curvature of the streamlines will fall off more rapidly with the distance from the interface because there can be no distortion of the centre line streamline because of the assumption of axial symmetry. Consequently, the periodic normal stress exerted on the interface will be less, although this will be partially offset by the velocity variations resulting from the changes in

the flow cross section. A similar effect occurs on the liquid side since the tube wall now becomes the position of the undisturbed streamline. Also it can be deduced that the thinner the liquid film, the greater the vapor velocity needed to cause an increase in amplitude of a given size of wave.

The unstable feature of the TLT system is when the annular flow is disturbed because of the unbalance in the normal stress between the liquid and the vapor interface. This unbalance will lead to cut off the flow stream or some chaotic flow situation and also lead to velocity and pressure perturbation. Finally this unbalance will induce a large pressure drop. When the annular flow in the riser is stable, the pressure drop is just induced from the pipe wall friction.

Considering Kelvin-Helmholtz instability in an annular flow, the problem of interfacial stability can thus be considered as the determination of these values of the flow and fluid parameters which make c (wave velocity) positive for a given value of the wave number k (wave number). Simply in a pipe flow, the wave disturbance can be assumed to be axisymmetric. The Kelvin-Helmholtz instability in an annular flow is mainly the relationship between the liquid flow and the vapor flow. Therefore, the behaviour of the vapor phase will be considered first followed by the liquid phase.

The disturbance to the interface is accompanied by velocity and pressure fluctuations in the vapor. Since the flow is axisymmetric, the velocities can be related to a Stokes stream functions of the form,

$$\psi = \bar{\psi}(r) + \delta\psi(r) \quad (4.79)$$

The pressure can be expressed in a similar form as,

$$p = \bar{p}(r) + \delta p(r) \quad (4.80)$$

where the first terms are the steady state stream function and pressure respectively, and the second

terms is the perturbations induced by the wavy interface. This perturbation can be expressed as exponential forms of $\delta\psi(r) = \hat{\psi}(r)e^{iky}$, and $\delta p(r) = \hat{p}(r)e^{iky}$. If these relationship is applied to the equations of motion in a cylindrical coordinate, the normal stress exerted by the vapor phase on the interface is,

$$\pi_{ng} = -k\hat{m}\rho_g(\bar{u}_g - c)^2 \left[\frac{I_0(k\bar{r}_i) - 1}{I_1(k\bar{r}_i)} \right] \quad (4.81)$$

The similar procedure can be applied to the liquid film. In this case, the liquid film is assumed to be thin enough, and then we can get the normal stress in the liquid film interface as,

$$\pi_{nl} = k\hat{m}\rho_l(\bar{u}_l - c)^2 \left[\frac{\cosh k\bar{m} - 1}{\sinh k\bar{m}} \right] \quad (4.82)$$

The instability of the interface will occur due to the unbalance between π_{ng} and π_{nl} . This instability directly leads to the unstable state of the TLT. Therefore, we need to find the stable state between the interfaces. The stable state of interface can be found by relating the surface tension, as

$$\pi_{nl} - \pi_{ng} = \sigma k^2 \hat{m} e^{iky} \quad (4.83)$$

Inserting Eq. (4.81) and (4.82) into (4.83), we obtain

$$k\hat{m}\rho_l(\bar{u}_l - c)^2 \left[\frac{\cosh k\bar{m} - 1}{\sinh k\bar{m}} \right] + k\hat{m}\rho_g(\bar{u}_g - c)^2 \left[\frac{I_0(k\bar{r}_i) - 1}{I_1(k\bar{r}_i)} \right] = \sigma k^2 \hat{m} e^{iky} \quad (4.84)$$

Simplifying,

$$B_1 = \left[\frac{\cosh k\bar{m} - 1}{\sinh k\bar{m}} \right] \quad (4.85)$$

$$B_2 = \left[\frac{I_0(k\bar{r}_i) - 1}{I_1(k\bar{r}_i)} \right] \quad (4.86)$$

Then we obtain the following characteristic equation.

$$\rho_l(\bar{u}_l - c)^2 B_1 + \rho_g(\bar{u}_g - c)^2 B_2 = \sigma k \quad (4.87)$$

$$(B_1 \rho_l + B_2 \rho_g) c^2 - 2(B_1 \rho_l \bar{u}_l + B_2 \rho_g \bar{u}_g) c + (B_1 \rho_l \bar{u}_l^2 + B_2 \rho_g \bar{u}_g^2 - \sigma k) = 0 \quad (4.88)$$

The solution of Eq (4.88) is,

$$c = \frac{2(B_1 \rho_l \bar{u}_l + B_2 \rho_g \bar{u}_g) \pm \sqrt{4\sigma k(B_1 \rho_l + B_2 \rho_g) - 4B_1 B_2 \rho_l \rho_g (\bar{u}_g - \bar{u}_l)^2}}{2(B_1 \rho_l + B_2 \rho_g)} \quad (4.89)$$

For a stable annular flow,

$$4\sigma k(B_1 \rho_l + B_2 \rho_g) - 4B_1 B_2 \rho_l \rho_g (\bar{u}_g - \bar{u}_l)^2 \leq 0 \quad (4.90)$$

Then,

$$4B_1 B_2 \rho_l \rho_g (\bar{u}_g - \bar{u}_l)^2 \geq 4\sigma k(B_1 \rho_l + B_2 \rho_g) \quad (4.91)$$

Therefore, we can obtain the stable velocity state expression as,

$$(\bar{u}_g - \bar{u}_l) \geq \sqrt{\frac{4\sigma k(B_1 \rho_l + B_2 \rho_g)}{4B_1 B_2 \rho_l \rho_g}} \quad (4.92)$$

Again, this equation can be simplified for a long wave and a normal annular flow ($B_2 = 1$) as

$$(\bar{u}_g - \bar{u}_l) \geq \sqrt{\frac{2\sigma}{\bar{m}\rho_g} \left(\frac{1}{2} k\bar{m} + \frac{\rho_g}{\rho_l} \right)} \quad (4.93)$$

The stability relationship in a vertical annular flow is now reduced to a simple formula. But this relationship cannot explain the whole system instability because the TLT system instability can be explained by the whole system phenomena as described in the early section of this Chapter.

The simple relationship above could not reach the exact instability criteria and can not satisfy different geometrical effects and the effects of various working fluid. In the simulation, the instability criteria is compared with experimental results by the relationship as shown in Eq. (4.93) with constant A_1 .

$$(\bar{u}_g - \bar{u}_l) \geq A_1 \sqrt{\frac{2\sigma}{\bar{m}\rho_g} \left(\frac{1}{2} k_s \bar{m} + \frac{\rho_g}{\rho_l} \right)} \quad (4.94)$$

The empirical constant A_1 , obtained from the SSL is given in Table 4.5. The constant A_1 is obviously different for different geometries and working fluids. The stable length, k_s , is defined as;

$$k_s = \sqrt{\frac{(\rho_l - \rho_g)g}{\sigma}} \quad (4.95)$$

and then Eq. (4.93) becomes

$$(\bar{u}_g - \bar{u}_l) \geq A_1 \sqrt{\frac{2\sigma}{\bar{m}\rho_g} \left(\frac{1}{2} \left(\frac{(\rho_l - \rho_g)g}{\sigma} \right)^{1/2} k_s + \frac{\rho_g}{\rho_l} \right)} \quad (4.96)$$

In the simulation, this instability criteria is a part of the computer program and liquid film thickness, entrainment and other parameters of Eq. (4.96) are calculated. This criteria is related to other two-phase flow parameters.

CHAPTER 5

RESULTS AND DISCUSSION

This chapter is divided into two-sections, one for the experimental result, Section 5.1, and the other the comparison between the experiment and the simulation, Section 5.2.

5.1 Experimental Results

The experimental test results in terms of the overall heat transfer performance of TLTs are shown in tabular and graphical formats. Figures 5.1 (a) to (f), 5.2, 5.3, 5.4 (a) and (b) and Tables 5.1 to 11 show the overall performance and summarized description of the TLTs tested in the present study as illustrated in the following.

TLT	Figures	Tables	Application
SSL	Figs. 5.1 (a) & (b), 5.3 and 5.4 (a) & (b)	Tables 5.1 to 5.7	MCM cooling in ATM switching systems
MSL I	Fig. 5.1 (c)	Tables 5.8 and 5.9	Refrigerator and Freezer
MSL II	Fig. 5.1 (d)	Tables 5.8 and 5.9	Basic TLT
LSL I	Fig. 5.1 (e)	Tables 5.10 and 5.11	Waste heat recovery system
LSL II	Figs. 5.1 (f), and 5.2	Tables 5.10 and 5.11	Waste heat recovery system

The heat transfer characteristics of the present TLTs are evaluated in terms of the overall temperature difference of the system, Δt_{h-c} , which is one of two appropriate parameters to identify the heat transfer phenomena involved, because the heat flux, q , is simply ($U_T \times \Delta t_{h-c}$). The other parameter is, of course, the overall heat transfer coefficient, U_T .

Figs. 5.1 (a) and (b) indicate that the SSL system of the present design would satisfy all the design objectives imposed, including the design target cooling heat flux of over 8 W/m² with the overall temperature difference of 50 °C as a MCM cooling system. Also Figs. 5.1 (c) to (f) show the overall performance of MSL I, MSL II, LSL I and LSL II, respectively.

It can be seen in Figs. 5.1 (a) and (b) that the choice of working fluid for TLT does not affect the overall heat transfer characteristics greatly within the design goal of 8 W/cm² with Δt_{h-c} of 50 °C. This was not the case for the two-phase closed thermosyphon as was seen by [Rhi, 1996]. However, the choice of working fluid must be considered in terms of the system vapor pressure and saturation temperature of the working fluid because the system stability depends strongly on the working fluid. This phenomena will be discussed again later. The figures for the SSL demonstrate that acetone seems to be the best working fluid for the present application. The worst seems to be FC-72. All other working fluids exhibit nearly the same heat transfer performance. However, it was noticed that water as a working fluid for the present application is accompanied by a large pressure fluctuation in the loop, consequently inducing a high temperature fluctuation in the evaporator section.

The MSL I is the heat transfer system employing two-phase loop thermosyphons for the heat extraction from confined spaces. As shown in Fig 5.1 (c), the experimental results indicate that the heat extracting rates of about 7 W/°C m² and 11 W/°C m² (the areas based on the outside heat transfer areas of the evaporator sections) at the average ambient air temperatures of the confined spaces of

-17 °C, and 3 °C, respectively, can be attainable under the forced convection heat transfer at the finned evaporator sections. R134a was used as the working fluid.

Figures 5.1 (d) to (f) show the effect of Δt_{h-c} on the heat transfer rate of MSL II, LSL I and LSL II, respectively. As shown in Fig. 5.1 (d), MSL II can perform the heat transfer rate of 1,500 W with 60 °C of Δt_{h-c} . For LSL II, experimental tests were carried out with two different condenser geometries, i.e, Counter and Parallel, as described in Chapter 3. Fig. 5.1 (f) shows that the difference between the counter flow and the parallel flow is rather insignificant. In a system which transfers heat between two different fluid media, the heat transfer performance may be explained by the system effectiveness and Fig. 5.2 shows the system effectiveness versus Δt_{h-c} of LSL II for both condenser geometries. It can be seen in the figure that the system effectiveness is placed within a range of 25 - 35 % and that LSL II with counter flow shows better performance than the parallel flow system in terms of the effectiveness.

The overall temperature difference of the system, Δt_{h-c} , depends, not unlike that of TCT, on the dimensionless volume of the working fluid, V^* , the evaporator-condenser length ratio, L^* , the air flow velocity at the condenser section, u_{max} , the saturation temperature of the system, t_{sat} , the heat flux, q , and the choice of the working fluid. The effect of each of these parameters will be discussed here.

The effects of various parameters for Medium and Large scale loops have restrictions due to the difficulties in changing configurations. Therefore, the range of parameters to be discussed will be very limited except for SSL.

5.1.1 Effect of number of condenser sections (SSL)

When the results of the small scale TLT system, SSL with one condenser, Fig. 5.1 (a), are compared with those with two condensers, Fig. 5.1 (b), it can be seen that there is an increase of about 15 to 20% in the performance of the latter to that of the former. This is expected because one of the major thermal resistances in the system occurs at the interface between the fins of the condenser and the ambient air which is the ultimate heat sink. However, the design of the system with one condenser section is more adequate for the present project objective.

5.1.2 Effect of Different Evaporator Design (SSL)

As mentioned in Section 3.1.2, for the small scale loop, SSL, three (3) different designs were adopted for the evaporator sections of the system; i.e., Evaporators A, B and C. In Fig. 5.3, the heat transfer performance characteristics of the SSL cooling system with these three different evaporators are presented. The figure indicate that the evaporators B and C are superior to Evaporator A.

However, a close examination of the evaporator A disclosed that it resulted mainly because of an increase in the thermal contact resistance between the heater surface and the evaporator due to the buckling deformation of the relatively thin wall of the heat transfer surface of the evaporator, induced by the vacuum inside.

The results indicate that the design of Evaporator C should be adopted for the performance as well as for the easiness of its manufacturing process.

5.1.3 Effect of Cooling Mode; Forced Convection vs. Natural Convection (SSL)

A series of experiment was carried out to establish the maximum cooling heat flux of the SSL system under the free convection mode and the results are presented in Figs. 5.4 (a) and (b).

It was seen in Figs. 5.1 (a) and (b) (Forced Convection Mode) and Figs. 5.4 (a) and (b) (Natural Convection Mode) that with acetone as the working fluid, the SSL can provide, with the overall temperature difference of 50 °C, the cooling heat fluxes of over 10 W/m² under the forced convection mode and more than 5 W/m² under the natural convection mode. This implies that the present small scale TLT system may easily satisfy most of cooling requirement of many electronic component without any forced cooling using fans.

5.1.4 Effect of Air Flow Velocity, $u_{\max}$ on Condenser Sections (SSL, LSL I and LSL II)

Figures 5.5 (a) to (f) show the effect of the air velocity in the finned condenser section on the heat transfer capacity of the TLT systems (SSL, LSL I and II). The velocity of air on the condenser section is a factor which affects the heat transfer capability of the system.

It is seen that the total thermal resistance was reduced because of the increased convection on the condenser section. However, it can also be seen in Figs. 5.5 (a) and (b) for the SSL that an increase in the air velocity greater than 2 - 3 m/s would not significantly increase the overall heat transfer capacity of the assembly. The reason for this phenomenon is illustrated in Fig. 5.5 (c) which shows that the air flow velocity on the condenser section has little effect on the forced convection heat transfer coefficient, h_{cond} , once the air velocity is larger than 2 - 3 m/s. Even when the heat transfer capacities of the working fluids were different, a similar tendency was observed. The effect of air velocity for one and two condensers is shown in Fig. 5.5 (d) in which the same trend can be

noticed. In Figs. 5.5 (e) to (g), the similar trends for LSL I and II with SSL are also seen. Figs 5.6 (a) and (b) show the effect of the air flow velocity in the condenser section on the system effectiveness. It is obvious that the system effectiveness increases with the increase of the air velocity as seen in Figs. 5.6 (a) and (b). Regardless of the system geometries, the velocity of air on the condenser section is a factor which affects the heat transfer capability of the system. However, there is a limit above which the effect is negligible.

5.1.5 Effect of Coolant Temperature (MSL II and LSL I)

Figs. 5.7 (a) and (b) show the effect of increasing coolant temperature thereby reducing the overall temperature difference across the loops for a constant heat loads.

The coolant temperature on the condenser section also is a factor which affects the heat transfer capability of the system. The effect of water coolant temperature, t_c , in the condenser section on the heat transfer capacity is shown in Fig. 5.7 (a) for MSL II.

It is seen that an increase in t_c is accompanied with the increased evaporator surface temperature. This implies that the heat transfer capability of the system would decrease with increasing water coolant temperature as expected.

On the other hand, air cooling system, LSL I, does show very little effect of cooling air temperature unlike the air flow velocity as shown in Fig. 5.7 (b) and this is because of the relatively small heat transfer coefficient of the cooling air medium.

5.1.6 Effects of Number, Arrangement and Length of Condensers (SSL)

It is expected that an increase in the heat transfer area of the condenser should reduce the overall thermal resistance of the system, thus increase the overall heat transfer capability. The experimental apparatus of the SSL was modified to accommodate two condenser sections to see if an increase in the condenser section area of the present design would further improve the heat transfer performance of the proposed system under the conditions of the forced and free convection, respectively, at the condenser outer surfaces. The results are quite encouraging as shown in Fig. 5.8 at the air velocity of 4.5 to 5 m/s. However, it can be deduced that there must be a limit to the number of condensers that can be increased for the system. This is one parameter that can be easily predicted by the analysis (simulation) which is to be discussed in Section 5.2.

In Fig. 5.9, the effect of the arrangement of two condensers on the heat transfer rate is shown. The heat transfer performance of the case for the condensers arranged “side-by-side” is higher than that of the case for “in-row”. This is because in “side-by-side” arrangement, all incoming air faced by the both condensers is at the ambient temperature, whereas in “in-row” arrangement, the incoming air faced by the second condenser is at a higher temperature because it has already picked up heat from the first condenser.

Figure 5.10 shows the effect of the length of the condenser (the condensing area exposed to the forced convection) on Δt_{h-c} . It can be seen that an increase in the condenser area resulted in smaller Δt_{h-c} , which is obvious, and this is the reason why an additional condenser (condenser No. 3) was added in the study. The evaporator C was always used in the study involving two condensers.

5.1.7 Effect of Saturation Temperature, t_{sat} (SSL, MSL II, LSL I and LSL II)

Figs. 5.11 (a) to (d) show the effect of the saturation temperature of the working fluid on the system performance for SSL, MSL II, LSL I and LSL II. In defining the heat transfer coefficients, h_{tp} and h_{cond} , it is essential to use t_{sat} . It is seen in Two-Phase Closed Thermosyphons for the case of Ref. [Rhi, 1996] that the higher t_{sat} , the smaller Δt_{h-c} (i.e., the higher U_T) for a given working fluid. In the experiment, the temperature of the transportation zone can be assumed to be the saturation temperature [Lee and Mitel, 1971].

However, for the SSL, the range of the working temperature of the present study is already prescribed as a design parameter, that is, $\Delta t_{h-c} = 50$ °C with t_c at 25 °C, the range of t_c is quite restricted and no special effort was made to study the effect of t_{sat} on Q at a given condition. However, from the previous study [Lee et al., 1996], it can be safe to expect that the overall heat transfer coefficient of the test thermosyphon, U_T , increases considerably with an increase in the saturation temperature, t_{sat} , in all working fluids because the boiling heat transfer coefficient (or overall heat transfer coefficient) always increases with increasing system pressure. Figs. 5.11 (a) to (d) obtained from the experiments carried out for MSL II, LSL I and II indicate this trend. The thermal properties for working fluid in TLTs are a function of saturation temperature and pressure. Increasing the heat input leads to the increase of the saturation temperature and pressure.

5.1.8 Effect of Quantity of Working Fluid (SSL and MSL II)

The quantity of the working fluid in a TLT would directly affect the heat transfer performance of the system. Figures 5.12 (a) to (g) show the effect of working fluid amount for SSL and MSL II with different working fluids. An important constraint in the operation of a TLT is the effect of V_{WF} or V_{WFR} .

In the present study, the quantity of the working fluid inside a TLT is defined as the ratio of the volume of working fluid at the ambient condition to the inside volume of the evaporator section:

$$V^* = \frac{V_l}{V_e} \quad (5.1)$$

For a two-phase closed thermosyphon, the estimation of the optimum amount of a working fluid is usually based on an assumption that the two-phase flow heat transfer in a TCT (two-phase closed thermosyphon) takes place in two modes, i.e., the heat transfer in the evaporator is that of the pool boiling, whereas the evaporative heat transfer is taken place from the falling condensate liquid film above the pool of the working fluid. There are several analytical studies on the amount of working fluids in heat pipes or TCTs based on very simplified assumptions but they all grossly underestimate the real situations [Lee et al., 1996].

However, for TLTs, this issue is not well documented in the literature. The amount of a working fluid should be sufficient enough to transfer the heat in the evaporator section as well as the amount of vapor and the returning condensate in the transporting loop and the condenser section. In the small scale loop (SSL) study, instead of the dimensionless parameter, V^* , the dimensional volume of working fluids, V_{wf} , was used because the objective of the present study was very specific. Figs. 5.12 (a) to 5.12 (e) illustrate the effect of the amount of the working fluid charged in TLTs on the total temperature difference or on the heat transfer rate for 3 working fluids, i.e., acetone, FC87 and R11. The study showed that for the system with one condenser, the optimum value of the filling charge is in the range from 30 to 40 ml and for the system with two condensers, about 50 ml. It was noticed from the experiment study that all other parameters have little or no effect. At the same time,

it was seen that there exists a minimum value of the filling charge which depends mainly on heat flux, the type of working fluid and the internal volume of the system.

For MSL II, the similar effect with the SSL was observed as seen in Fig. 5.12 (f) and (g).

Fig. 5.12 (f) shows the similar trend with that of the SSL and it can be seen that for the MSL II system, the optimum value of the filling charge is about 50 % of total loop volume.

5.1.9 Effect of Inclination of Condenser Section; Free Convection Mode (SSL)

In Figs. 5.13 (a) to 5.13 (c), the effect of the inclination of the condenser with respect to the horizontal plane for the free convection cooling is shown where the horizontal plane is taken as the inclination angle of zero.

Since the fins are attached to the condenser pipes normal to the vertical axis, air flow induced for free convection is hindered, reducing the effect of convection if the inclination of the condenser section is vertical (90°) as seen in the figures. However, the reality is much complex. In Fig. 5.13 (b), for $Q < 60$ W, the effect is opposite and we can not explain why this is so for the moment.

Figure 5.13 (c) shows the effect of the inclination of the condenser in the inclination range of $8 - 90^\circ$ from the horizontal plane. It is seen that the heat transfer performance is decreased with increasing the angle from the horizontal plane to the vertical plane.

5.1.10 Effect of Contact Resistance (SSL)

When the filler material (or attaching medium) between the heater and the evaporator surface is properly selected and packed, the effect of the contact resistance of the filler can be correctly simulated using the property data of the filler provided by the manufacturer of the filler materials.

It is extremely important that this resistance must be as small as possible to obtain the high heat transfer of the cooling system using TLTs.

In the present study, the results shown in Fig. 5.14 were obtained under the same pressure force applied to the filler by tightening the four screws in the corners of nylon boxes of the heater assembly (Plate 3.5). It is interesting to see that the aging of the filler improves significantly; the thermal resistance of the filler decreases with time, and this may be due to moisture migration with time into the filler.

5.1.11 Effect of Non-Condensable Gas (SSL)

It has been well known that the performance of a Two-Phase Closed Thermosyphon (TCT) will deteriorate in the presence of non-condensable gases which evolve from dissolved gases in the working fluid and the absorbed gas in the thermosyphon structure. The dissolved non-condensable gases in the working fluid are swept along with the vapour flow and accumulate at the condenser end. This will form a gas plug that represents a barrier to the flowing vapor and may eventually shut down the operation of a TCT. To eliminate the sources of the non-condensable gases, the working fluid must be systematically purged of the dissolved gases. However, the presence of a non-condensable gas in a TCT can be utilized in a positive manner to control the system temperature.

The effect of the presence of non-condensable gases in a Two-Phase Loop Thermosyphon (TLT) is not well known and unlike in a TCT, it can not be used in a positive manner to control the heat transfer rate. Usually the presence of the non-condensable gas results from the contact between air and the working fluid during the manufacturing, storage, or charging processes of a TCT/TLT system. For an example, up to 6 ml of air can be naturally dissolved under the atmospheric pressure

and room temperature in 100 ml of water as shown in Table 3.5. Therefore, the dissolved air in TCT/TLTs must be eliminated under vacuum before it is charged with a working fluid.

This contamination problem is especially serious for the new fluorinert liquids of FC-72 and FC-87, due to its extremely high solubility of air (as much as 10 times greater than for water) as pointed out in Table 3.5 of Section 3.1.4.

In the present experimental study, the test TLT assembly had one additional thermocouple just below the NUPRO valve connected to the condenser section as shown in Figs. 3.1, 3.2 and Plate 3.6. This thermocouple identifies the presence of non-condensable gases which are then removed from the TLT.

As was the case of the TCT system [Lee et al., 1996], a small amount of air inside the TLT system could significantly decrease the heat transfer performance as seen in Fig. 5.15. Therefore, it can not be cautioned enough to stress that the presence of non-condensable gases in the system must be avoided.

5.1.12 Total and Apparent Mass Flow Rate of the Working Fluid (SSL)

To supplement the analysis (computer simulation), a novel experimental device was introduced to measure the **total** mass flow rate, m_T , of the working fluid. The **apparent** mass flow rate, m_f , is the theoretical mass flow rate calculated from an energy balance, $Q = m_f h_{fg}$. In the loop of a TLT, the working fluid may still have a quality less than 1 and consequently, $m_T > m_f$. To test this hypothesis, we have modified our test TLT apparatus slightly as shown in Fig. 5.16 (a). This modification made it possible to measure the approximate value of m_T very close to the reality.

A typical result is shown in Fig. 5.16 (b) which indicates that the ratio between the total and apparent mass flow rates strongly depends on the total heat transfer rate. From the study, it is concluded that a great care must be exercised in the analysis how to calculate the mass flow rate which must satisfy both the mass and energy balance equations. Knowing m_T , the average quality of the working fluid in the two-phase region can be obtained.

5.1.13 Flow Visualization Study (SSL)

Photographic observations of flow regimes and temperature fluctuations were obtained and the results are presented in Fig. 5.17. Photographic observations were made by shadow-graphic technique. From the figures, slug and annular flow patterns are obviously recognized. The figures with water show that the slug flow pattern is observed even at high heat flux. In case of other working fluids, the flow pattern changed from the slug flow to the annular flow with increasing heat flux. Flow pattern is directly involved with the system stability. From the figures, it was observed that the SSL system is operating in annular flow state at heat flux over 60 W. In the present experimental study, the flow patterns observed did not fall within the generally accepted flow pattern map as shown in Figs A.1 of Appendix A.. This implies that the flow patterns of a TLT depends on the TLT geometries and flow state such as flow rate, pipe diameter, heat flux etc.

5.1.14 TLT Transient Temperature Variation from System Startup (SSL, LSL II)

Figs. 5.18 (a) and (b) show the transient behaviour of the systems from the startup (SSL) and with a heat flux change (LSL II). For SSL, the temperature was observed from a thermocouple installed in the evaporator. As shown in Fig. 5.18 (a), the temperature is increased from the starting

point to the peak point in which the boiling starts and this period must be for bubble formation. As boiling begins, the periodic temperature oscillations occur. The basic source of this oscillatory phenomena is the process of bubbles formation, growth and broken in the evaporator. Summarizing this oscillatory phenomena in brief, the divorcing velocity of bubbles formed on the interior surface of the evaporator, and intensifies the heat transfer process in the evaporator. The increased vapor flow induces to thin the thickness of the condensed film in the condenser and further increases the coefficient of condense heat transfer. On the contrary, the fluctuation of fluid flow leads to increase the hydrodynamic resistance along all circulation, induces to slow the velocity of fluid circulation, and further weaken the heat and mass transfer in a TLT. Besides, the intensive fluctuation of fluid flow in the TLT can bring about the vibration of heat pipe, this probably decreases the reliability and durability of the heat pipe. In fact, the influence of fluctuation of fluid flow in a TLT loop is complex and multiple.

Fig. 5.18 (b) shows the transient behaviour of the LSL II from the abrupt change of the heat load to potentially stable state. The temperature behaviours of all thermocouples installed in the loop were observed. The system behavior for the LSL II is similar to Fig. 5.18 (a).

5.1.15 Instability of Working Fluid Flow (SSL)

The experimental investigation of the system instability is a direct extension of the present study. The instability of the working fluid flow within the loop is one of the major concern for the design of any loop-type two-phase thermosyphons. Figs. 5.19 (a) and (b) are showing the temperature fluctuation within the SSL studied as a function of heat flux and working fluids, respectively.

In Fig. 5.19 (a), it is seen that the temperature fluctuations with water as the working fluid are severe, even at high heat fluxes. In low heat flux, obviously large temperature fluctuations were observed. Severe density oscillations occurred in the heat flux of 100 W.

In Figs. 5.19 (b), the temperature fluctuations as a function of working fluids are illustrated. It can be seen that even without any insert to damp the flow fluctuation, acetone and FC-87 hardly showed any fluctuation, whereas water has shown very strong fluctuation, especially at the evaporator and at the inlet of the condenser section. This is because the saturation pressure of water as the working fluid at these temperature ranges are so low (0.1 ~ 0.2 atm. Pressure) that a slight system pressure change can easily be translated to a relatively large temperature fluctuation, and for this reason, water can not be recommended for the working fluid of the SSL of the present application.

Figure 5.20 (a) show the direct effect of Q on the height of liquid column in the condensate return line observed in the glass tube provided in the condensate returning tube located between the condenser and the evaporator as shown in Plate 3.5. It is seen that the higher heat flux, the higher column height. The liquid column height is directly involved into the system stability because the static pressure force is the main source of system operation. The fluctuation of the heights of the working fluid are shown in Figs. 5.20 (b) for R-11 and 5.20 (c) for FC-87, respectively.

Figures 5.21 (a) and (b) show the pressure distribution along the loop of MSL II and LSL I, respectively. The effects of various parameters on the pressure drop and pressure drop fluctuation are reported in Figs. 5.22 (a) to (f) for LSL II. Figure 5.22 (a) shows that the pressure drop between the evaporator outlet and the condenser inlet is decreased with increasing heat flux. The pressure drop measurement in Fig. 5.22 (a) is from the third loop of LSL II. It seems that the pressure drop

for the counter flow has slightly steeper than that of the parallel flow.

Figs. 5.22 (b) and (c) show the effect of the saturation pressure on the pressure drop of LSL II. As seen in Fig. 5.22 (b) for LSL II with the parallel flow, the pressure drop of the transport pipe is decreased with increasing P_{sat} , and the pressure drop in the condensate return line is increased. This same trend is seen in the same manner for the case of counter flow in Fig. 5.22 (c). This trend agrees well with that of Friedly et al [1979]. Friedly et al. carried out the study on the effect of pressure drop ratio on system stability. Their pressure drop ratio was defined as the ratio of the evaporator inlet pressure drop (ΔP_i) and the outlet pressure drop (ΔP_o), ie., ($\Delta P_i / \Delta P_o$). From their study, the system stability decreased with increased pressure drop ratios. This trend can also be found Figs. 5.22 (d) and (e) where it can be seen that P_{sat} is increased linearly with increasing heat flux and saturation temperature. A change in the system vapor pressure at the range of the saturation temperature of water is strongly amplified and this in turn induced the pressure difference between the evaporator and the condenser. Fig. 5.22 (f) shows an example of the pressure drop fluctuation in a given time and operation condition for the LSL II.

Loop thermosyphon systems are subject to instabilities which are undesirable for several reasons. As described in Chapter 1, parameters (pressure drop, working fluid amount, insert size, heat flux and so on) affecting the stability of TLTs have been investigated in the present experimental study. The detailed parametric effects on the system stability are presented in Section 5.1.17.

As mentioned above, Figs 5.19 (a) and (b) show the clear effects of heat flux and different working fluids on temperature fluctuation. These instabilities hamper the smooth and continuous circulation of working liquid and vapor. Sustained flow oscillations may cause the mechanical vibration of the TLT bundle.

5.1.16 Effect of Insert Orifice (SSL)

As mentioned in Section 3.1.2, for the SSL, three orifice inserts (cylindrical : $d = 5.5$ mm and 6 mm in height, $d_{\text{orifice}} = 0.51, 0.71$ and 1.2 mm, respectively. See Plate 3.7) were used to study the effect of a change in the pressure drop in the flow channel of the TLT on the flow instability and temperature fluctuation. These inserts are placed inside the condensate line using a Tee connector situated near the evaporator section (140 mm from the upper edge of the evaporator).

The results are shown in Figs. 5.23 (a) to 5.23 (d). It can be seen in Fig. 5.23 (a) for the case with water as the working fluid that decreasing the size of orifices from 3.7 mm (no insert) to 0.7 mm drastically reduced the fluctuation of the temperature, especially that of the evaporator. Fig. 5.23 (b) indicates that with the orifice of 0.7 mm, the overall temperature fluctuation can be almost eliminated as the power is increased.

It could be expected that the smaller the size of the orifice, the larger the overall pressure drop of the system because of the insert, thus reducing the total mass flow rate which would have resulted in low heat transfer performance. However, Fig. 5.23 (c) and (d) shows that this is not all true. Previous studies by others generally showed that the system instability decreased with decreasing the insert size without considering the system performance. [Lorenzini, 1981, and Imura, 1994]. In a loop system, the instability has been studied widely by many during last 50 years since Wissler studied the instability of a two-phase loop system in 1955. But the studies of instabilities have mostly limited to single phase flow or theoretical approach with limited experimental verifications.

As mentioned by Wissler [1956], the loop oscillatory behavior is mainly from the transport section due to the lack of driving force. The oscillatory nature is caused by the system seeking a stable operating point. For this nature, the system tends to both over and undershoot the equilibrium

point. The oscillatory instability phenomena is dominated by various flow parameters (pressure drop, density, void distribution, quality, heat flux, working fluid and so on)

In Figs. 5.23 (c) and (d), it is seen that the best heat transfer performance is obtained with 0.71 mm ID insert for acetone and water. However, for the case with acetone as the working fluid, the differences in the performance with different inserts or even without any insert are relatively small. With water as the working fluid, there seems to exist an optimum size of the inserts, namely 0.71 mm ID insert.

5.1.17 Effects of Other Parameters on System Instability (SSL)

Figures 5.24 and 5.25 show the effect of Q on Δt_{hc} in view of the instability. The overall temperature difference of a TLT increases with increasing heat flux. Increasing heat flux leads to increase the operation pressure and void fraction, quality, saturation temperature and consequently the driving force will be increased. Obviously, the increased driving force leads to increase the stability. Fig 5.26 which is for natural convection mode, shows the same trend with those observed in Fig. 5.24 and 5.25.

It is generally accepted that the amount of the working fluid affects the system stability. Ishii [1990] reported that increasing the amount of working fluid reduced the stability. This feature is seen in Fig. 5.27 (a) and (b). Theoretical study of Ishii [1990] show that the system stability is decreased with increasing the amount of working fluid. But the present experimental study as shown in Figs. 5.27 (a) and (b) indicates that this is not necessarily true. As seen in the figures, the best stable performance is obtained from 40 ml. This implies that the theory can not predict the optimum working state with stability combined with the performance.

Ishii's analysis [1990] predicts that the stability is decreased with increasing $u_{\max}$ and l_c . Our study shown in Figs. 5.28 and 5.29 indicates that the temperature fluctuation with increasing $u_{\max}$ and l_c is very marginal.

5.2 Comparisons between Simulation and Experimental Results

5.2.1 Introductory Remarks

This section presents a comparison between the experimental and the corresponding simulated results for five different TLTs (SSL, MSL I, MSL II, LSL I and LSL II). In all figures shown (Figs. 5.30 to 5.36) in this section, the line plots represent the simulated results.

As described in a previous study [Rhi, 1996] for two-phase closed thermosyphons (TCT), a meaningful simulation can not exist without a benchmark experimental test. This is demonstrated in Figs. 5.30 (a) to (d) for the completeness. Fig. 5.30 (a) is the experimental setup of the TCT studied and Figs. 5.30 (b) to (d) are the representative results.

It will be shown that the success of a computer simulation based on an analytical thermal resistance network method depends on the choice of empirical correlations for various thermal resistances. For the TCT mentioned above, selecting 8 empirical correlations for the boiling heat transfer coefficients (h_e), 3 for the condensation (h_c), 3 for forced convection (h_{conv}) and 3 for shape factor (S), there could be as many as 216 possible solutions for a given condition from the simulation and every one of them could be the right one. Fig. 5.30 (b) presents the simulation results with different nucleate boiling heat transfer correlations, with a fixed condensation and convection heat transfer correlation, and shows large differences among the simulation results based on different correlations.

However, since the solution was based on the thermal resistance network method, the key to check the correctness of the simulation is to compare the temperature distribution within the TCT assembly obtained by the simulation with that of experiments as shown in Figs. 5.30 (c).

Even though the agreement between the simulation and the experiment are reasonable, the simulated interior temperature distributions could be quite different from the experimental results, depending on the choice of the empirical correlations used for boiling and condensation. For the TCT studied, the use of Bier et al.'s correlation for boiling [1994] and Gross' correlation for condensation [1992] gave the best fit. But to obtain the best possible agreement between the experimental and simulation heat transfer results, some modification was needed.

To modify a given combination of the boiling and condensation heat transfer correlations, the following expression by Park [1992] was used for the heat transfer coefficient in the evaporator section and the condenser sections.

The modified heat transfer coefficients for boiling and condensation (h_{tp} and h_{cond}) are expressed, respectively, as.

$$h_e = C_e x_e^{m_e} \quad (5.2)$$

where x_e is the original correlation for boiling;

$$h_c = C_c x_c^{m_c} \quad (5.3)$$

where x_c is the original correlation for condensation.

The constants C_e and C_c and exponents m_e and m_c were then determined from the experimental data so as to correlate the data best.

For the case of the TCT mentioned above, the values obtained were $C_e = 0.6$ and $C_c = 0.8$,

respectively. [Rhi, 1996]. With the modified correlations of the boiling and the condensation heat transfer coefficients, it was then possible to compare the simulation results with the experimental results and consider the effects of the various parameters as seen in Fig. 5.30 (d).

5.2.2 Comparisons for TLTs

The basic concept for the simulation of TLT systems starts from the same concept used for the simulation of the TCT discussed above. However, the process involved for a TLT is much different from the one for a TCT. In a TLT system, the simulation starts from the flow feedback concept and ends with the same. The simulation logics for both the lumped and sectorial methods require the condition that the two-phase flow inside the loop should satisfy mass, energy, and momentum balances. And the simulation program must converge to satisfy the other condition that the total pressure drop summation along the loop must be zero ($\sum \Delta P = 0$). This is illustrated in Fig. 5.31. The crux of the simulation is to find the convergence point with a large number of iterations with changing mass flow rate.

Figures 5.32 to 5.36 present the comparison results between the experimental and the simulation results for all TLTs studied experimentally. The ultimate test of a computer simulation is to compare the simulation results with the experimental data. Only when the simulation is verified by the benchmark experimental data, its usefulness is authenticated.

Since the present simulation study is for the two-phase fluid flow in TLTs which involved a large number of empirical correlations, one must be careful to test the simulation not only for the final results but also for the crucial elements which would determine the final outcome. It was shown [Lee et al., 1996] that the success of the computer simulation for a TCT strongly depends on the

choice of empirical correlations required for various thermal resistances used in the analysis.

The most crucial element among many is the temperature distribution along the loop of the TLT. Only when this bench mark element is verified, one can be confident that the simulation is dependable for the range for the bench mark elements verified.

In the present simulation study, we used 10 empirical correlations for the flow boiling (for lumped method) and 2 correlations (for annular flow for the sectorial method) for the boiling heat transfer coefficients, (h_{fp}), 3 for the condensation (h_{cond}), and 3 for the forced convection (h_{conv}), many possible solutions for a given condition from the simulation are attainable and every one of them could be the right one as discussed above for a TCT. The present simulation study is focused on the effects of heat transfer coefficient correlations. However, if we are to consider in the simulation the effects of other two-phase flow parameters such as the pressure drop, the void fraction, etc., the solution of the simulation could be numerous without any physical significance for each.

Table. 5.12 tabulates Figures 5.32 to 5.36 for the comparison of the simulation results with those of the experimental for various parameters. For example, all the figures with the sub-designation (a) in Figs. 5.32 to 5.36 refer the figures for the simulation results of the five TLTs studied with the different two-phase heat transfer coefficient correlations but with a fixed condensation and convection heat transfer correlations.

Figures 5.32 to 5.36 with the sub-designation (a) for Q vs. Δt_{h-c} show large differences between the experimental results and those of the simulation. The largest differences can be found in the cases for SSL and MSL I, and the main reason for this may be due to the fact that the correlations used in the simulation are originally developed for large scale two-phase systems such as nuclear reactors or industrial re-boilers. The test environmental conditions of the various

empirical correlations used in the present simulation study are already described in Chapter 2. It is seen in the figures that the simulation results with the empirical correlations for the two phase flow heat transfer, h_{tp} , of Kandlikar [1989], Chen [1966] and Shah [1982] for the Lumped method and that of Rhee and Young [1974] for the Sectorial method show the best agreement with the experimental results. In these simulation, the heat transfer coefficient for condensation, h_{cond} , of Nusselt [1916] and the forced convection heat transfer coefficient, h_{conv} of Knudson [1958] were chosen as the fixed parameters. For the case of MSL I, because of the peculiar condenser geometry which is quite different from those of other TLTs, the condensation heat transfer coefficient correlation, h_{cond} , developed by Smirnov and Lukanov [1972] was used. The combinations of correlations used in the simulation are in Table 5.13.

Even though the comparison for the large scale TLTs show better agreements between the experiment and simulation than those for the cases of SSL and MSL I & II, the simulation results by the Lumped method and by the Sectorial method did not show any partiality for the method used for the simulation. However, the simulation results with the Lumped method vary widely depending on the choice of h_{tp} , but with the Sectorial method, the combination of h_{tp} , h_{cond} and h_{conv} given above placed the agreement between the simulation and experiment within an acceptable error range (less than 20 % for LSL II). Generally, it was noticed as far as the present study is concerned that the simulation by the Lumped method gave reasonable agreements for the large scale TLTs with the selected correlations (Kandlikar, Chen and Shah), whereas the simulation by the Sectorial method gave reasonable agreements for all TLTs within an acceptable error range as seen in Figs. 5.32 to 5.36 with the sub-designation (a).

Since the solution required for the simulation for a TLT is based on a thermal resistance net

work which requires a large number of empirical correlations for various heat transfer phenomena along the TLT, the primary key to check the correctness of the simulation code is to compare the temperature distribution within the TLT obtained by the simulation with that of the experiments as discussed above for the simulation of a TCT. As seen in Figs 5.32 (b) to (h), depending on the selection of the empirical correlations for boiling and condensation heat transfer coefficients, the simulated interior temperature distributions could be quite different from the experimental results. In the case of the simulation by the Lumped method, for all TLTs except MSL I, the combination of the boiling heat transfer coefficient correlation of Kandlikar [1989] and of Chen [1966] for MSL I and the condensation heat transfer coefficient by Nusselt [1916] gave the best fit among all the correlations. In the case of the Sectorial method, the heat transfer coefficient correlation for the annular flow boiling of Rhee and Young [1974] gave a rather good agreement with the experimental results. However, it was noticed that even this combination of the empirical heat transfer correlations for boiling and condensation did not give any reasonable agreement for the cases with different working fluids as was with the simulation for a TCT. Therefore, to achieve the best fit between the experimental and simulation results, the correctional method used for the TCT is adopted to modify the heat transfer correlations for boiling and condensation, h_{tp} and h_{cond} , respectively, for a given working fluid.

To modify a given combination of the boiling and condensation heat transfer correlations, h_{tp} and h_{cond} , the expression given above as Eqs. 5.2 and 5.3, were used and the values of the correction constants for modification are given in Table 5.13.

The comparisons between the simulation and the experimental results for the temperature distributions of the five TLTs studied are shown in the figures with appropriate sub-designations as

given in the third column of Table 5.12.

Figs. 5.32 to 5.36 with the sub-designation given in the fourth column of Table 5.12 for Q vs. Δt_{h-c} (*modified*), show the comparison of the experimental results on the heat transfer rate of the system as a function of the overall temperature difference between the evaporator and the ambient air. In spite of complexity involved in choosing the right combination for various empirical correlations used in the simulation, the agreement is very good. The simulation predicts correctly not only the effect of the different working fluids but also the magnitude of the heat transfer rates.

Figs. 5.32 to 5.36 with the sub-designations given in the fifth, sixth, seventh and ninth columns of Table 5.12 for V_{WF} , u_c , l_c and *effectiveness*, respectively, show that the simulation results agree reasonably well with the experimental results for the two methods of simulation used in the present study, i.e., Lumped and Sectorial models.

The attempts to simulate the system instability are shown in Figs 5.32 (o) to (q). The instability calculation with the correction factors given in Table 4.5, Chapter 4, agree rather well with the experimental results. This may be confined for the SSL studied as there are no other experimental data for other TLTs available for comparison.

The simulation to check the system instability for other TLTs was also attempted. However, the results of the simulation for the system instability for other TLTs show that all TLT systems other than SSL are unstable. This implies that either the large TLT system within the simulation range may be unstable or the simulation analysis presented in the thesis still requires some further refinement. This remains to be studied.

CHAPTER 6

CONCLUDING REMARKS

An experimental and analytical (numerical) study was carried out on the fluid flow, heat transfer and instabilities for five (5) small, medium and large scale two-phase loop thermosyphons. Experimentally, various parameters which would affect TLT operations such as Δt_{h-c} , t_{sat} , u_{max} , u_c , l_c , V_{WF} , V_{WFR} , pressure drop, effectiveness and instability were investigated. Analytically, two simulation methods based on thermal resistance net work was considered, namely, Lumped and Sectorial methods.

The five two-phase closed loop thermosyphons (TLTs) studied were specially designed and constructed for the present study, and they are one small scale loop (SSL : 5.4 mm ID & 1.2 m $\times$ 0.5 m), two medium scale loops (MSL I: 0.095 mm ID & 0.423 m in width and MSL II: 0.0202 m ID & 3 m $\times$ 2 m) and two large scale loops (Loop I: 0.0202 m ID & 1.8 m $\times$ 2 m and Loop II: 0.0202 m ID & 2.208 m $\times$ 10 m).

From the study, the following conclusions may be made:

1. Experimental results have shown that for a given loop operating condition (t_c , u_c , l_c , ΔL_{e-c} , V_{WF} and V_{WFR}), the loop performances will be affected by changes in the source and sink

temperatures. Through a careful selection of the loop charge as well as the condenser length, working fluid, and other parameters, a TLT could provide a very attractive heat transfer capability over a wide range of temperature differences.

2. For the instability of TLTs, only the SSL was used. Three different working fluids of acetone, water, and FC-87 were used in the study. It was observed that the loop instabilities are affected by the charge, heat flux, working fluid, and insert size. The experimental observation showed that an insert before the evaporator inlet of the SSL significantly reduced the temperature fluctuations observed in the loops without any such insert. For the SSL studied, the best performance was obtained when a 0.7 mm ID insert was placed. It was also noticed that as the heat flux was increased, the temperature fluctuations were reduced.
3. The simulation study by the two models, Lumped and Sectorial, on five very different TLT systems are compared with the experimental results and the limitation of the computer simulation for such two-phase heat transfer systems is noted. The present study confirms our belief that a computer simulation alone could not, at the moment, give any meaningful quantitative results unless it is accompanied with some experimental results for a system involving some empirical correlations. This implies that no computer simulation should be developed for a two-phase heat transfer system such as TLTs studied without a benchmark experimental verification. However, once verified, the computer simulation is capable of simulating some performance characteristics of such systems.
4. It was seen that the comparisons between the experiment and the simulation results without any modification to the many empirical correlations needed in the simulation do not agree well especially with the experimental results from the smaller scale loops. This could be that the

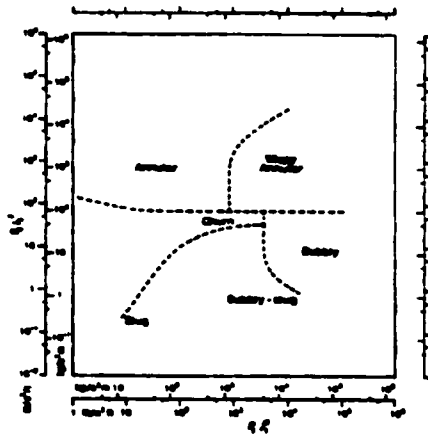
correlations used in the simulation are basically developed for large scale heat transfer systems involving two-phase flow. Any simulation with empirical correlations of unknown origin must be verified by experiment.

5. An analysis based on the Kelvin-Helmholtz instability theory for the system instability of a TLT was made. It was seen that the critical velocity accompanied with a correction factor can simulate the system instability to a certain extent and that the theory agrees rather well with the experimental results of the SSL. No comparison with experimental data for other TLTs was not possible because of the lack of the latter.

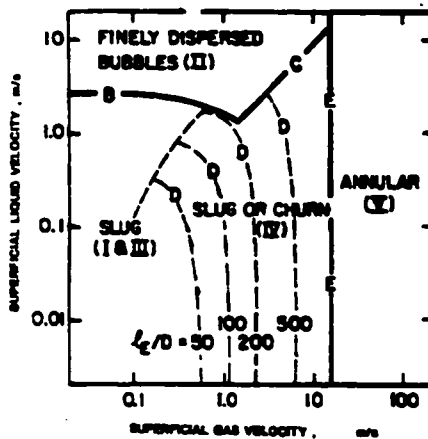
APPENDIX A

Flow Pattern Maps and Flow Pattern Transitions

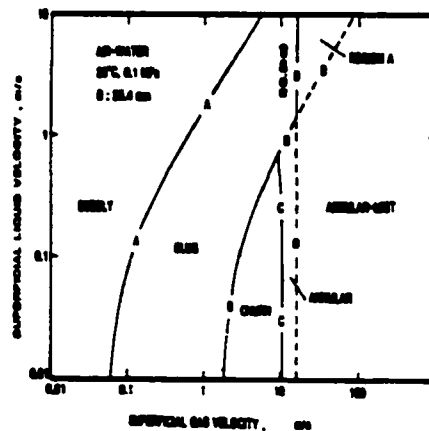
1. Flow Pattern Map



(a) Typical vertical flow pattern map by Hewitt and Roberts [1969 (Given in Collier, 1994)]



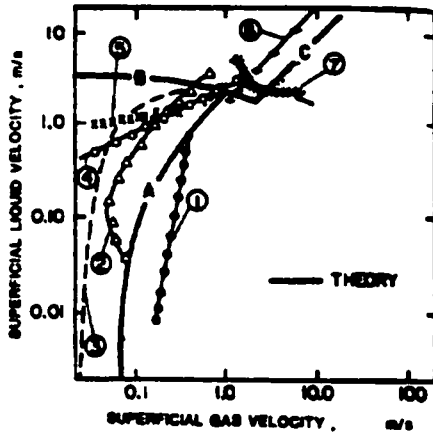
(b) Taitel and Duckler [1980]



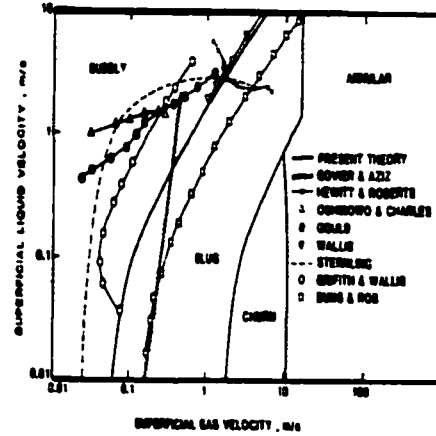
(c) Mishima and Ishii [1984]

2. Flow Regime Transition and Comparison

- Bubbly flow to Slug flow transition

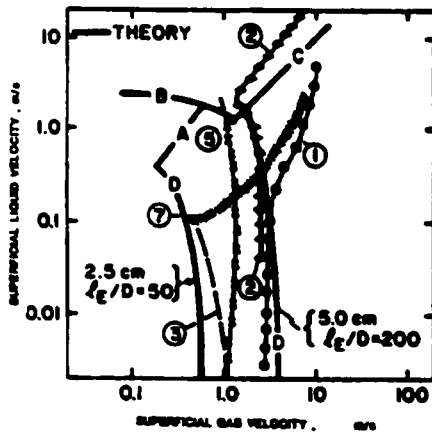


(a) Taitel and Duckler [1980]

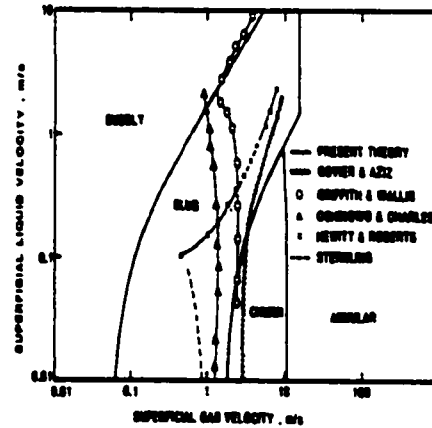


(b) Mishima and Ishii [1984]

- Slug flow to Churn flow transition

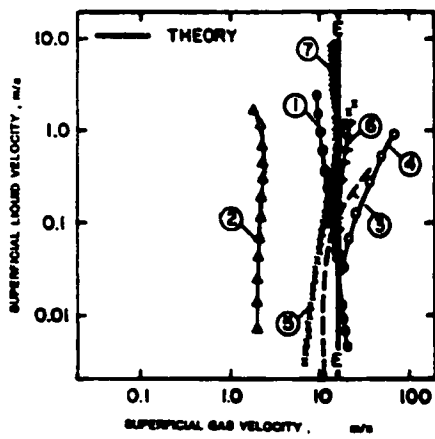


(a) Taitel and Duckler [1980]

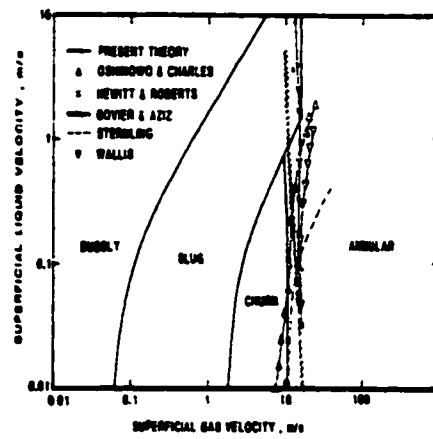


(b) Mishima and Ishii [1984]

- Churn flow to Annular flow transition



(a) Taitel and Duckler [1980]



(b) Mishima and Ishii [1984]

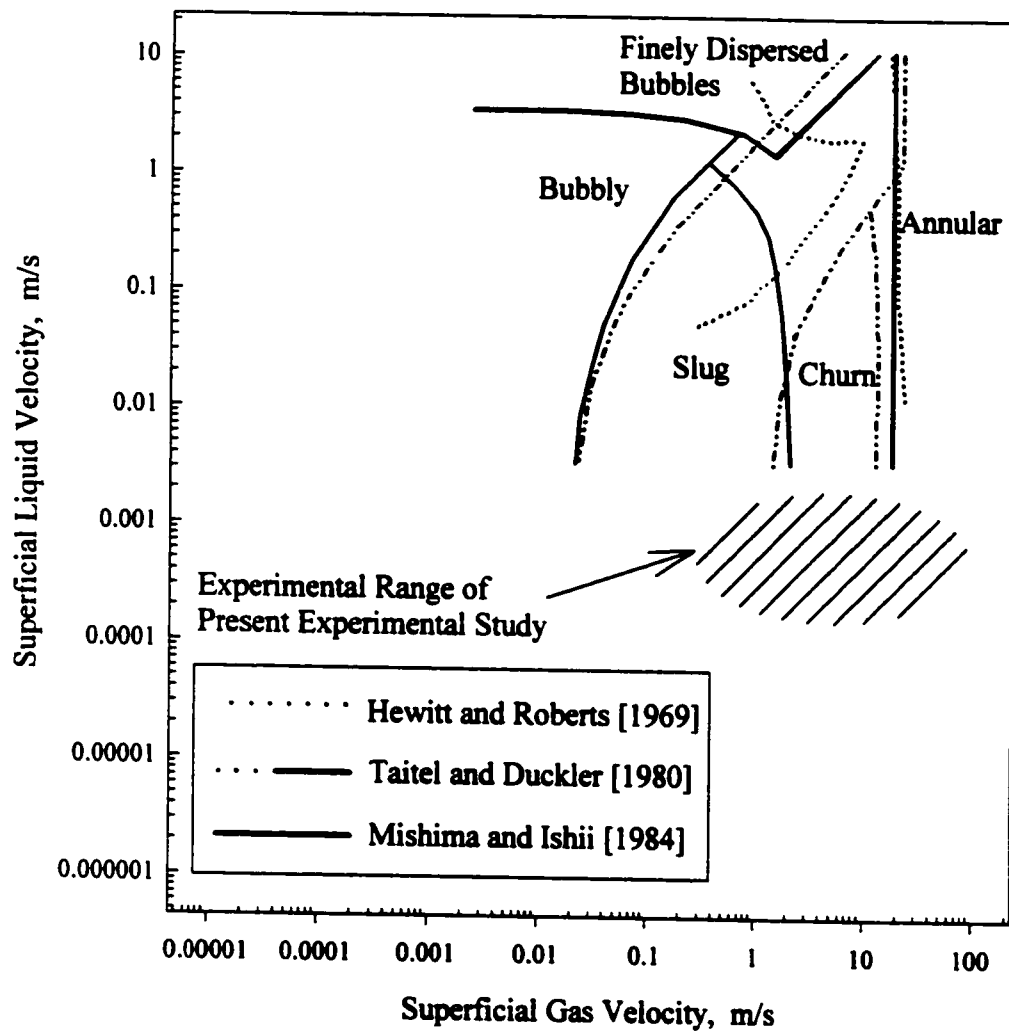


Figure A.1 Comparison of three generally accepted flow pattern maps
(Reproduced from original flow pattern maps)

APPENDIX B

Uncertainties of Two-Phase Flow Parameters

Uncertainties of following selected two-phase flow parameters are presented in tubular formats.

- Two-phase heat transfer coefficient correlation
- Void fraction correlation
- Two-phase pressure drop correlation

Table B.1 Comparison between Correlation and Experiments on Heat Transfer Coefficient (%)

Correlations Data	Dengler and Addoms [1956]	Guerrieri and Talty [1956]	Bennett et al. [1959]	Schrock and Grossman [1959]	Chen [1966]	Lavin and Young [1965]
Dengler and Addoms [1956]	30.5	62.3	20.0	20.3	14.7	187.5
Schrock and Grossman [1959]	89.5	16.4	24.9	20.0	15.1	
Sani [1960]	26.9	70.3	26.5	48.6	8.5	
Bennett et al. [1959]	17.9	61.8	11.9	14.6	10.8	225.0
Guerrieri and Talty [1956]	55.0	10.1	60.9	44.2	10.8	150.0
Wright [1961]	24.0	75.8	30.4	51.7	15.4	

**Table B.1 Comparison between Correlation and Experiments
on Heat Transfer Coefficient (%) (Continued)**

Exp. Correlation	Kilmenko [1988]	Kandikar [1989]			Hahne [1989]	Gungor and Winterton [1986]
		W	R	C		
Shah [1976]	-60 ~ 170	17.9	25.8	52.1	41.6	17.4 ~ 45.0
Bjore et al. [1982]	-35 ~ 41	24.6	54.7		7.2	37.8 ~ 114.1
Chen [1966]	-61 ~ 135	29.6	48.6		47.8	27.6 ~ 91.7

W : Water, R : Refrigerants, C : Cryogenic fluids

Table B.2 Comparison between Correlation and Experiments on Void Fraction (%)

Exp. Correlation	Friedel [1977]		Klidonas [1986]	
	Water	R-12	Water	Refrigerant
Chisholm [1973]	-0.4	-1.8	3.5 ~ 14.9	5.6
Hughmark [1962]	-10.8	-18.5		
Smith [1969]	0.5	4.1	3.1 ~ 16.0	6.4
Thom [1964]	36.5	52.2		
Bankoff [1960]			4.0 ~ 15.5	5.5
Homogeneous			4.3 ~ 27.1	11.9
CISE [1961]			3.1 ~ 21.2	5.8
Zuber [1965]			3.2 ~ 16.6	7.4
Rouhani [1969]			3.6 ~ 21.7	5.5
Armand [1946]			3.3 ~ 15.6	5.9

Table B.2 Comparison between Correlation and Experiments on Void Fraction (%)
(Continued)

Exp. Correlation	Friedel [1977]		Klidonas [1985]	
	Water	R-12	Water	Refrigerant
Zivi [1964]			4.5 ~ 20.8	7.4
Klidonas [1985]			4.2 ~ 13.6	6.0
Walley [1978]			7.8 ~ 24.9	9.4
Baroczy [1965]	-5.6	-10.6		

Table B.3 Comparison Between Pressure Drop Correlations and Experimental Data
In Straight Pipe (%)

Exp. Correlations	Dukler et al. [1964]	Friedel [1977]	Weisman and Choe [1976]	Idsinga [1977]
Homogeneous		24.8 ~ 38.9		-33.1
Lockhart and Martinelli [1949]	-5.5 ~ 38.3		19.1 ~ 418.7	155.4
Bandell and Schlunder [1974]			69.3	
Dukler et al. [1964]			7.0 ~ 370.0	
Baker [1954]			-3.8 ~ 455.9	
Hoogendoorn [1961]			-0.8 ~ -19.7	
Martinelli and Nelson [1948]				52.8
Bankoff [1961]	737 ~ 4810			-31.0
Lombardi and Pedrocchi [1972]		28.0 ~ 56.3		12.9

Table B.3 Comparison Between Pressure Drop Correlations and Experimental Data in Straight Pipe (%) (Continued)

Correlations \ Exp.	Dukler et al. [1964]	Friedel [1977]	Weisman and Choe [1976]	Idsinga [1977]
Chenoweth and Martin [1955]	-8.5 ~ 95.4		-22.0 ~ 403.5	
Baroczy [1965]		13.1 ~ 53.2	-6.1	-19.8
Chisholm and Sutherland [1969]			39.5	
Chisholm [1973]			-18.9	-10.2
Thom [1964]				-6.4
Becker [1962]				96.9
Levy' momentum exchange [1960]				47.3
Armand [1946]				112.2

Table B.4 Two-Phase Pressure Drop in Orifice (%)

Correlations \ Exp.	Friedel [1985]	Sadatomi [1988]
Chisholm [1966]	100.0	-6.5 ~ 15.1
Beattie [1973]	62.0	
James [1965]	201.0	-18.9 ~ 8.6
Lin and Hanratty [1987]	127.0	-10.4 ~ 19.9
Murdock [1962]	147.0	-11.4 ~ 8

Table B.5 Two-Phase Pressure Drop in Valves and Area Change (%)

Correlations \ Exp.	Friedel [1985]	Simpson [1985]	Grønnerud (1972)
Tremblay [1972]	4.0	-42.2	
Fairhurst [1983]	-25.0	-18.2 ~ 10.3	
Morris [1983]	70.0	-4.9 ~ 28.2	
Chisholm et al. [1980]	83.0	138.0	110.0
Simpson et al. [1985]	94.0	-21.6 ~ 25.5	
Strathclyde [1983]			-16.0
Dukler et al. [1964]			-7.0
Grønnerud [1972]			-6.0
Fairhurst [1983]			-20.0
Homogeneous	14.0	69.1	

Table B.6 Pressure drop and Void fraction Calculation with Combinations of Different Correlations (Sub-average percent error, %)

α \ Δp_f	Dukler et al. [1964]	Oliemans [1976]	Beggs and Brill [1973]
Hugmark [1962]	710 / 57.9	694 / 30.1	
Eaton et. Al [1967]	72.5 / 50.6	78.0 / 6.2	75.3 / 58.9
Dukler et al. [1969]	-50.0 / 50.9		
Beggs and Brill [1973]			470 / -0.3

Table B.7 Pressure drop and Void fraction Calculation with Combinations of Different Correlations (Overall average percent error, %)

α \ Δp_r	Dukler et al. [1964]	Oliemans [1976]	Beggs and Brill [1973]
Hugmark [1962]	306 / 26.0	301 / 13.0	
Eaton et al. [1967]	40.4 / 32.9	51.6 / -4.9	46.0 / 24.9
Dukler et al. [1969]	-20.3 / 41.7		
Beggs and Brill [1973]			214 / -1.4

- * Data sets are obtained from experimental results of "Oil Pipe Line".
- * Above table is rearranged from the results of Garry et al.[1985].
- * Detail calculation procedure is given in Garry et al.[1985].

Pressure Drop Comparison
Void Fraction Comparison

Sample View

Void Fraction Corr.	Pressure Drop Corr.	Percent Error	
		Pressure Drop	Void Fraction
Hughmark [1962]	Dukler et al. [1964]	306.0	26.0

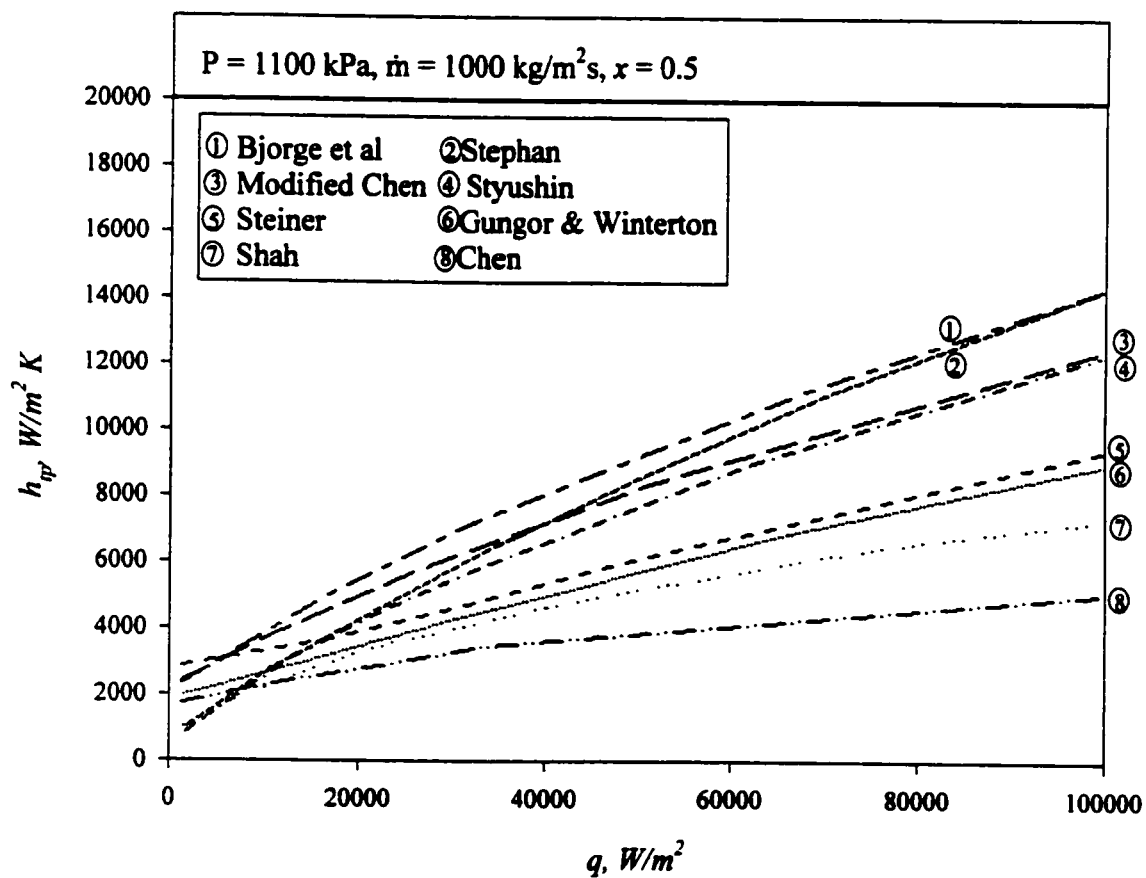
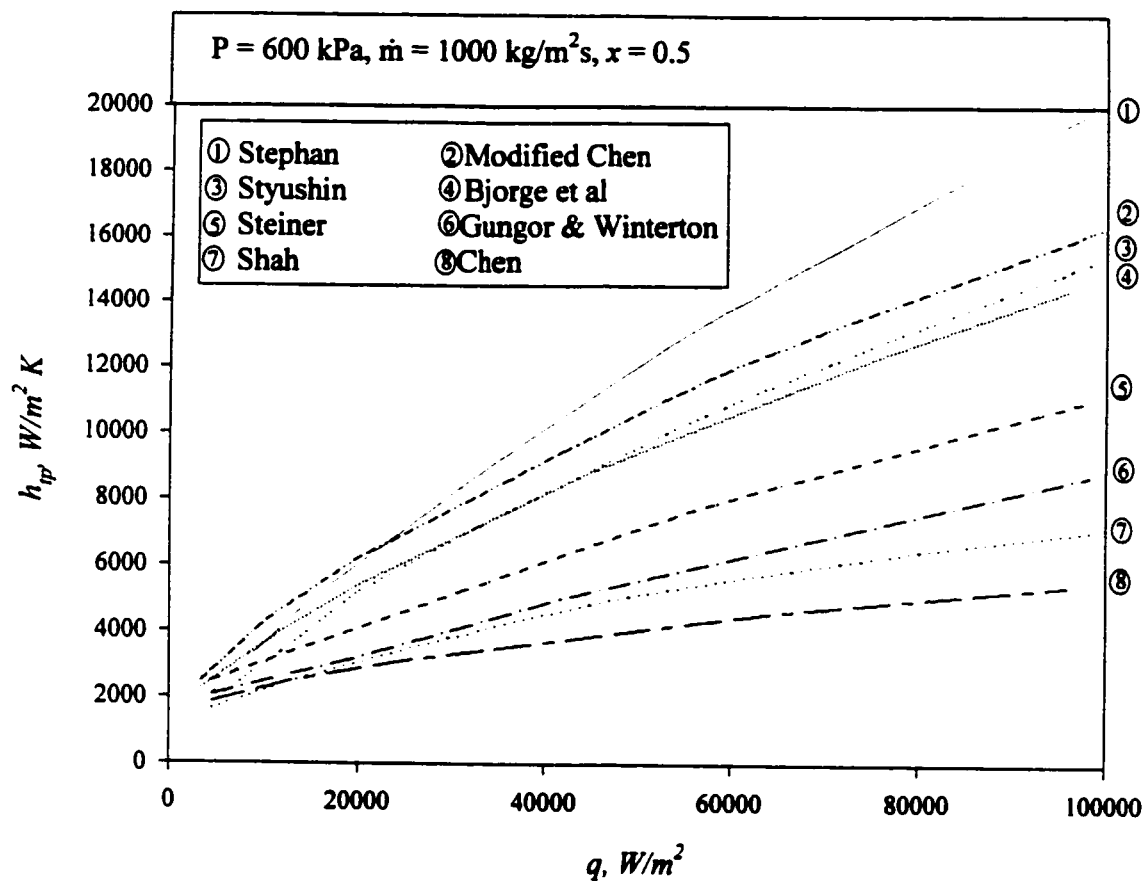


Figure B.1 Selected Graphical Comparison of different h_{wp} correlations [Hahne et al., 1989]

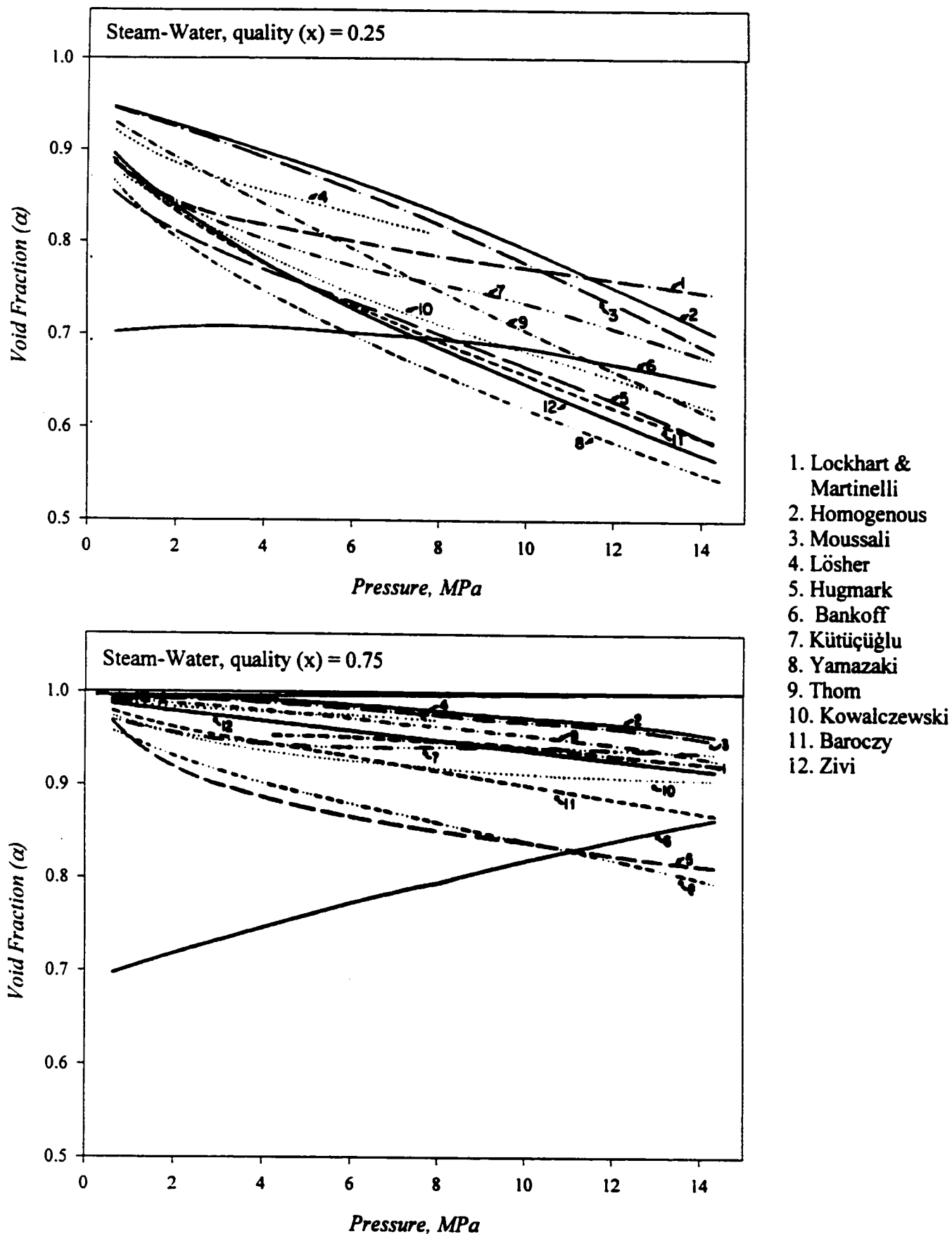


Figure B.2 Selected Graphical Comparison of Void Fraction Correlations [Isbin et al., 1979]

APPENDIX C

Two-phase Heat Transfer Coefficient Correlations

1. Lumped Models

(a) Type A - Martinelli [1949]'s Suggestion :

$$\frac{h_p}{h_{lo}} \left[\text{or } \frac{h_p}{h_l} \right] = f_n \left(\frac{1}{X_n} \right)$$

Equation to be correlated :

$$\frac{h_p}{h_l} = a \left(\frac{1}{X_n} \right)^b$$

$$h_l = 0.023 \frac{k_l}{D} \left(\frac{D \dot{m} (1-x)}{\mu_p} \right)^{0.8} \text{Pr}_l^{0.3}$$

$$h_{lo} = 0.023 \frac{k_l}{D} (\text{Re}_{lo})^{0.8} \text{Pr}_{lo}^{0.3}$$

$$\frac{1}{X_n} = \left(\frac{x}{1-x} \right)^{0.9} \left(\frac{\rho_l}{\rho_g} \right)^{0.5} \left(\frac{\mu_g}{\mu_l} \right)^{0.1}$$

Authors	a	b	h_i base	Fluid
Dengler and Addoms [1956]	3.5F	0.5	h_{io}	Water
Guerrieri and Talty [1956]	3.4E	0.45	h_i	Various Organics
Schrock and Grossman [1959]	2.5	0.75	h_{io}	Water
Wright [1961]	2.721	0.581	h_i	Water
Bennet et al. [1959]	$0.5649 \times q^{0.11}$	0.74	h_i	Water
Somerville [1962]	7.55	0.328	h_i	n-Butanol
Collier and Pulling [1962]	2.167	0.699	h_i	Water
Chaddock and Noeranger [1966]	3.0	0.667	h_i	Water
Uchida and Yamaguchi [1966]	$17.5Bo^{0.2}$	1.67	h_i	R-12
Pujol and Stenning [1969]	4.0	0.37	h_{io}	R-113
Sekoguchi [1992]	2.5	2/3	h_{io}	Water

(b) Type B - Schrock and Grossman [1959]:

$$\frac{h_p}{h_i} = a1 \left(Bo + a2 \left(\frac{1}{X_u} \right)^{b1} \right)^{b2}$$

Authors	a1	$a2 \times 10^{-3}$	$b1 \times 10^4$	b2	h_i base	Fluid
Schrock and Grossman [1959]	7.39	1.5	0.67	1	h_{io}	Water

Authors	a1	a2×10 ⁻³	b1×10 ⁴	b2	<i>h_f</i> base	Fluid
Sani [1960]	14.8	1.5	0.67	1	<i>h_f</i>	Water
Wright [1961]	6.7	3.5	0.67	1	<i>h_f</i>	Water
Somerville [1962]	24.5	1.5	0.67	1	<i>h_f</i>	n-Butanol
Chaddock and Brunemann [1967]	19.1	1.5	0.67	0.6	<i>h_{fo}</i>	Various chlorinated paraffins
Pujol and Stenning [1969]	9.0	4.45	0.37	1	<i>h_{fo}</i>	R-113
Pujol and Stenning [1969]	5.3	7.55	0.67	1	<i>h_{fo}</i>	R-113

Following correlation are mainly the modification of above correlation.

Gungor and Winterton [1986] :

$$\frac{h_p}{h_f} = \left(1 + 3000Bo^{0.86} + \left(\frac{x}{1-x} \right)^{0.75} \left(\frac{\rho_l}{\rho_g} \right)^{0.41} \right)$$

Chaddock and Noeranger [1966] :

$$\frac{h_p}{h_f} = 1.85 \left(10000Bo + 1.5 \left(\frac{1}{X_n} \right) \right)^{2/3}$$

(c) Type C - Modified Chen [1966] 's Correlations :

Mainly Chen's correlation is to describe the effect of nucleate boiling in forced convective boiling. Therefore, his correlation includes two terms such as nucleate boiling heat transfer coefficient correlation and convection heat transfer coefficient correlation with correction factors.

Chen's correlation :

Two-phase heat transfer coefficient

= the contribution of forced convective heat transfer + the contribution of bubble nucleate boiling

$$h_{tp} = h_{mic} + h_{mac}$$

$$h_{mic} = 0.00122 \left[\frac{k_l^{0.79} c_{pl}^{0.45} \rho_l^{0.49}}{\sigma^{0.5} \mu_l^{0.29} h_{fg}^{0.24} \rho_g^{0.24}} \right] \left[t_w - t_{sat}(P_l) \right]^{0.24} \left[P_{sat}(t_w) - P_l \right]^{0.75} S$$

$$h_{mac} = h_l F(1/X_n)$$

Correlations	F	S
Chen [1966]	Graphical Form, F vs (1/X _n)	Graphical Form , S vs (Re _{tp})
Butterworth [1979, (given in Hewitt, 1994)]	F(X _n) = 1 for X _n ⁻¹ ≤ 0.1 F(X _n) = 2.35(0.213 + 1/X _n) ^{0.736} for X _n ⁻¹ > 0.1	S(Re _{tp}) = (1 + 2.56 × 10 ⁻⁶ Re _{tp} ^{1.17}) ⁻¹
Bennett et al. [1980]		$S = \frac{1 - \exp\{-F(X_n)h_l X_0 / k_l\}}{F(X_n)h_l X_0 / k_l}$ $X_0 = 0.041 [\sigma / (g(\rho_l - \rho_g))]^{0.5}$
Shah [1982]	F = 1 + 1.8(1/X _n) ^{0.71} [ρ _l /ρ _g] ^{0.04} [μ _l /μ _g] ^{0.07}	
Gungor and Winterton [1986]	F = 1 + 1.37(1/X _n) ^{0.86} + 24000 Bo ^{1.16}	S = 1 / (1 + 1.15 × 10 ⁻⁶ F ² Re _l ^{1.17})
Jung et al. [1989]	F = 2.37 (0.29 + (1/X _n) ^{0.85})	
Kenning and Cooper [1989]	F = 1 + 1.8(1/X _n) ^{0.87} for 0.6 ≤ 1/X _n ≤ 40	
Wadekar [1990]	F = 3.2 × (x/(1-x)) ^{3/4} Pr ^{-3/8} × M ^{1/3} + 1	

Following correlations used basically the same concept with Chen's correlation and in some case, the correlation is modified with Chen's correlation.

Bennett and Chen [1980]

$$h_{tp} = h_{mic} + h_{mac}$$

h_{mic} is same as in Chen's correlation.

$$h_{mac} = h_l \left[\frac{(dp/dz)_{tp}}{(dp/dz)_l} \right]^{0.44} F(Pr_l) \left(\frac{\Delta T}{\Delta T_{sat}} \right)_{mac}$$

For pure components

$$\left(\frac{\Delta T}{\Delta T_{sat}} \right)_{mac} = 1$$

$$F(Pr_l) = Pr_l^{0.296}$$

Shah [1976]

$$\psi_s = \frac{h_{tp}}{h_l} = f(Co, Bo, Fr_{le})$$

$$Co = \left(\frac{1-x}{x} \right)^{0.8} \left(\frac{\rho_g}{\rho_l} \right)^{0.5}$$

h_{tp}/h_l is found from a chart according to the value of Co . Graphical Chart is given in Shah [1976]

Shah [1982]

h_c :
for vertical

$$N_s = Co$$

for horizontal

$$N_s = Co \text{ for } Fr_{le} \geq 0.04$$

$$N_s = 0.38 Fr_{le}^{-0.3} Co \text{ for } Fr_{le} < 0.04$$

where

$$Co = \left(\left(\frac{1}{x} \right) - 1 \right)^{0.8} \left(\frac{\rho_g}{\rho_l} \right)^{0.5}$$

$$h_c/h_l = 1.8 N_s^{-0.8}$$

h_{nb} :

For $N_s > 1.0$

$$h_{nb}/h_l = 230 Bo^{0.5} \text{ for } Bo > 0.0003$$

$$h_{nb}/h_l = 1 + 46 Bo^{0.5} \text{ for } Bo < 0.0003$$

For $0.1 < N_s < 1.0$

$$h_{nb}/h_l = F \times Bo^{0.5} \exp(2.74 N_s^{-0.1})$$

For $N_s < 0.1$

$$h_{nb}/h_l = F \times Bo^{0.5} \exp(2.74 N_s^{-0.15})$$

$$Fr_l = \frac{G^2}{\rho_l^2 g D} \quad \text{and} \quad Bo = \frac{q}{G h_{fx}}$$

$$F = 14.7 \text{ for } Bo > 0.0011$$

$$F = 15.43 \text{ for } Bo < 0.0011$$

h_{tp} :

$$h_{tp} = \max [h_c, h_{nb}]$$

Gungor and Winterton [1986]

$$h_{tp} = Eh_l + Sh_{nb}$$

$$E = 1 + 24,000 Bo^{1.16} + 1.37(1/X_n)^{0.86}$$

$$S = 1 / \left[1 + 0.00000115 E^2 Re_l^{1.17} \right]$$

h_l is Dittus - Boelter correlation

$$h_{nb} = 55 P_r^{0.12} (-0.4343 \ln P_r)^{-0.55} M^{-0.5} q^{0.67}$$

Kandlikar [1989]

$$h_{tp} = h_l \left[C_1 Co^{C_2} (25 Fr_{le})^{C_3} + C_3 Bo^{C_4} F_k \right]$$

$$Co = \left(\frac{1-x}{x} \right)^{0.8} \left(\frac{\rho_g}{\rho_l} \right)^{0.5}$$

	$Co < 0.65$ (Convective Region)	$Co \geq 0.65$ (Nucleate Boiling Region)
C_1	1.1360	0.6683
C_2	-0.9	-0.2
C_3	667.2	1058.0
C_4	0.7	0.7
C_5	0.3	0.3

Fluid	F_K
Water	1.00
R-11	1.30
R-12	1.50
R-13B1	1.31
R-22	2.20

Fluid	F_K
R-113	1.30
R-114	1.24
R-152a	1.10
Nitrogen	4.70
Neon	3.50

Bjorge et al [1982]

$$h = \frac{q_{tot}}{t_w - t_{sat}}$$

$$q_{tot} = q_{fc} + q_{fdb} \left[1 - \left(\frac{(t_w - t_{sat})_i}{t_w - t_{sat}} \right)^3 \right]$$

$$q_{fc} = F_B \text{Pr}_l \frac{k_l}{D} (t_w - t_{sat}) \frac{\text{Re}_l^{0.9}}{C_2}$$

$$F_B = 0.15 \left(\frac{1}{X_n} + \frac{2}{X_n^{0.32}} \right)$$

for $\text{Re}_l > 1125$

$$C_2 = 5 \text{Pr}_l + 5 \text{Pr}_l \ln(1 + 5 \text{Pr}_l) + 2.5 \ln(0.0031 \text{Re}_l^{0.812})$$

for $50 < \text{Re}_l < 1125$

$$C_2 = 5 \text{Pr}_l + 5 \text{Pr}_l \ln(0.0964 \text{Re}_l^{0.585} - 1)$$

for $\text{Re}_l \leq 1125$

$$C_2 = 0.0707 \text{Pr}_l \text{Re}_l^{0.5}$$

$$q_{fdb} = B_M \mu_l h_{fg} \left[\frac{g(\rho_l - \rho_g)}{\sigma} \right]^{0.5} \left[\frac{k_l^{0.5} \rho_l^{17/8} c_{pl}^{19/8} (t_w - t_{sat})^3}{\mu_l h_{fg}^{7/8} (\rho_l - \rho_g)^{9/8} \sigma^{5/8} t_{sat}^{1/8}} \right]$$

for water $B_M = 1.89 \times 10^{-14}$

$$(t_w - t_{sat})_i = \frac{8\sigma T_{sat} h_{fc}}{k_l h_{fg}} \left(\frac{1}{\rho_g} - \frac{1}{\rho_l} \right)$$

Steiner and Taborek [1992]

$$h_{tp} = \left[(h_{NB} F_{NB})^3 + (h_{fo} F_{TP})^3 \right]^{1/3}$$

$$h_{fo} = \frac{k_l (f/8)(Re - 1000) Pr}{D [1 + 12.7(f/8)^{1/2} (Pr^{2/3} - 1)]}$$

$$f = [0.7904 \ln(Re) - 1.64]^{-2}$$

$$F_{TP} = \left\{ \left[(1-x)^{1.5} + 1.8x^{0.6}(1-x)^{0.01} \left(\rho_l / \rho_g \right)^{0.35} \right]^{-2.2} + \left[(h_{bo} / h_{fo}) x^{0.01} (1 + 8(1-x)^{0.7}) \left(\rho_l / \rho_g \right)^{0.67} \right]^{-2} \right\}^{-0.5}$$

for water

$$h_{NB} = h_o \left[1.73 P_r^{0.27} + \left(6.1 + \frac{0.68}{1 - P_r} \right) P_r^2 \right] \left[\phi / \phi_o \right]^{0.9 - 0.3 P_r^{0.15}} \left[R_p / R_{po} \right]^{0.133}$$

for other fluids

$$h_{NB} = h_o \left[1.2 P_r^{0.27} + 2.5 P_r + \frac{P_r}{1 - P_r} \right] \left[\phi / \phi_o \right]^{0.9 - 0.3 P_r^{0.15}} \left[R_p / R_{po} \right]^{0.133}$$

$$F_{NB} = F_{PF} (\phi/\phi_o)^{nf} (D/D_o)^{-0.4} (R_p/R_{po})^{0.133} F(M)$$

$$F_{PF} = 2.816 P_r^{0.45} + \left(3.4 + \frac{1.7}{(1 - P_r^7)} \right) P_r^{3.7}$$

for cryogenics

$$nf = 0.7 - 0.13 \exp(1.105 P_r)$$

for all fluid except cryogenics

$$nf = 0.8 - 0.1 \exp(1.75 P_r)$$

$$F(M) = 0.377 + 0.199 \ln M + 0.000028427 M^2 \leq 2.5$$

where M is molecular weight

Liu and Winterton [1991]

$$h_{tp} = \sqrt{(Fh_i)^2 + (Sh_{pool})^2}$$

for laminar flow

$$h_i = 0.86 \frac{k_f}{D} \left(\frac{D_i}{D_o} \right)^{-0.16} \frac{f / 8 (\text{Re}_i - 1000) \text{Pr}_i}{\left(1 + 12.7 \sqrt{f / 8} (\text{Pr}_i^{2/3} - 1) \right)} \left(1 + \left(\frac{D_h}{L} \right)^{2/3} \right)$$

for turbulent flow

$$h_i = \frac{k_f}{D} \left[3.66 + 12 \left(\frac{D_i}{D_o} \right)^{-0.8} + \left(1 + 0.14 \left(\frac{D_i}{D_o} \right)^{0.5} \right) \frac{0.19 (\text{Re}_i \text{Pr}_i D_h / L)^{0.8}}{1 + 0.117 (\text{Re}_i \text{Pr}_i D_h / L)^{0.467}} \right]$$

$$h_{pool} = 7.42 q_w^{0.67} R_p^{-0.21 \log P/P_r} P_r^{0.12} / (-\log P_r)^{0.55}$$

$$S = (1 + 0.055 F^{0.1} \text{Re}_i^{0.16})^{-1}$$

$$F = \left(1 + x \text{Pr}_l \left(\rho_l / \rho_g - 1 \right) \right)^{0.35}$$

Kilmenko [1988]

$$Nu_p = \max[Nu_b, Nu_c]$$

$$Nu_b = C Pe^{*0.6} K_p^{0.54} \text{Pr}_l^{-0.33} \left(\frac{k_w}{k_l} \right)^{0.12}$$

where Pe^* is a modified Peclet number, K_p is dimensionless parameter.

Fluids	C	Pr
Freons	0.0076	3.8 ± 2.5
Organic fluids	0.0068	3.5 ± 1.3
Cryogenic fluids	0.0061	1.6 ± 1.0
Water	0.0049	1.2 ± 0.4

$$Nu_c = 0.087 \text{Re}_m^{0.6} \text{Pr}_l^{1/6} \left(\frac{\rho_g}{\rho_l} \right)^{0.2} \left(\frac{k_w}{k_l} \right)^{0.09}$$

(d) Type D - Modification of h_i or h_{io} :

Authors	Correlations
Coulson and Mcnelly [1956]	$h_{tp} = \left(1.3 + \frac{D}{A}\right) \left(\frac{k_l}{D}\right) \text{Re}_l^{0.23} \text{Pr}^{0.9} \text{Re}_g^{0.34} \left(\frac{\mu_g}{\mu_l}\right) \left(\frac{\rho_b}{\rho_l}\right)^{0.23}$
Groothuis and Hendel [1959]	$h_{tp} = 0.029 \left(\frac{k_l}{D}\right) \text{Re}_l^{0.87} \text{Pr}^{1/3} \left(\frac{\mu_f}{\mu_w}\right)^{0.14}$
Kvamme [Mathur, 1986]	$\frac{h_{tp}}{h_i} = 15 \left[\frac{(1-x)}{R_l} \right]^{0.8}$
Sachs and Long [1961]	$h_{tp} = 0.023 \left(\frac{k_l}{D}\right) \text{Re}_{tp}^{0.8} \text{Pr}^{1/3}$
Ananiev [1961]	$h_{tp} = h_i \left(\frac{\rho_l}{\rho_{av}}\right)^{0.5}$
Chawla [1967]	$h_{tp} = \left(\frac{k_l}{D}\right) C_1 (\text{Re}_l Fr_l)^{C_2} \left(\frac{x}{1-x}\right) \left(\frac{\rho_l}{\rho_g}\right)^{0.8} \text{Re}_l^{0.35} \text{Pr}_l^{0.45}$
Beaver and Hughmark [1968]	$h_i = 0.023 \text{Re}_l^{0.8} \text{Pr}^{0.4} \left(\frac{\mu_l}{\mu_w}\right)^{0.4} \left(\frac{1}{R_l}\right)^{0.8}$ R_l is Martinelli's liquid holdup
Sohns [1974]	$h_{tp} = 0.264 \left(\frac{k_l}{D}\right) \text{Re}_{tp}^{0.65} \text{Pr}^{1.21}$
Crain and Bell [1973]	$h_{tp} = 0.0587 \left(\frac{k_l}{D}\right) \text{Re}_l^{0.85} \text{Pr}^{0.4} \left(\frac{d}{D}\right)^{0.1} x^{-7.6}$

Authors	Correlations
Shah [1981]	$h_p = h_i \left(1 + \frac{j_g}{j_l} \right)^{0.25}$
Elamvaluthi and Srinivas [1984]	$h_p = 0.5 \left(\frac{k_l}{D} \right) \text{Re}_m^{0.7} \text{Pr}^{0.33} \left(\frac{\mu_g}{\mu_l} \right)^{0.25} \left(\frac{\mu_b}{\mu_w} \right)^{0.14}$
Doughherty et al. [1990]	$h_p = 0.166 \left(\frac{k_l}{D} \right) \text{Re}_i^{0.66} \text{Pr}^{0.33} \left(\frac{\mu_f}{\mu_w} \right)^{0.14}$
Steiner and Taborek [1992]	$h_p = h_i \left[(1-x)^{1.5} + 1.9(x)^{0.6} \left(\frac{\rho_l}{\rho_g} \right)^{0.35} \right]^{1.1}$
Mobarak et al. [1984]	for $x < 0.1$ $\text{Nu}_p = 0.8 \left(\frac{\text{Re}_p}{\text{Re}_i^{0.75}} \right)^{1.15}$ for $x > 0.1$ $\text{Nu}_p = 2.75 \left(\frac{\text{Re}_p}{\text{Re}_i^{0.75}} \right)^{0.75}$

Chawla [1970]

$$h_p = \left(\frac{k_l}{D} \right) A C_1 (\text{Re}_i \text{Fr}_i)^{C_2} \left(\frac{x}{1-x} \right) \left(\frac{\rho_l}{\rho_g} \right)^{0.8} \text{Re}_i^{0.35} \text{Pr}_i^{0.45}$$

$$A = \left[1 + \left(1 - \frac{1-x}{x \varepsilon \frac{\rho_l}{\rho_g}} \right)^{-0.5} \right]^{-1}$$

$$C_1 = 0.006, C_2 = 0.475 \quad \text{for } \text{Re}_i \text{Fr}_i < 109$$

$$C_1 = 0.015, C_2 = 0.3 \quad \text{for } \text{Re}_i \text{Fr}_i > 109, \varepsilon \text{ is determined from Graph}$$

2. Sectorial Models

(a) Bubbly Flow ($\alpha < 0.25$)

Authors	Correlations
Kutateladze [1961]	$h_{bub} = h_l \left[1 + \left(\frac{h_{pool}}{h_l} \right)^2 \right]$
Mikielewicz [1973]	$h_{bub} = h_l \left[B \left(\frac{1}{X_n} \right)^m + \left(\frac{h_{pool}}{h_l} \right)^2 \right]^{0.5}$
Rhee and Young [1974]	$h_{bub} = 0.023 Re_i^{0.8} Pr_i^{0.4} + 0.3 Re_{th}^{0.69} Pr_i^{0.69} \left(\frac{\rho_l}{\rho_g} - 1 \right)^{0.31} \left(\frac{PD}{\sigma} \right)^{0.31}$

Bohdal et al., [1999]

$$h_{bub} = A \left(\frac{k_l}{D} \right) (Re)^{2.42} (Fr)^{-0.35} (Ku)^{1.38} (Bo)^{-1.08}$$

$$A = 4.5 \times 10^{-5}$$

$$Re = \left(\frac{G(1-x)D}{\mu_l(1-\alpha)} \right)$$

$$Fr = \frac{G^2}{gD\rho_l^2}$$

$$Ku = \frac{q(1-\alpha)}{h_{fg}\rho_g v_l(1-x)}$$

$$Bo = \frac{q}{\rho_g h_{fg}}$$

$$\alpha = \frac{1}{1 + \frac{1-x}{x} \frac{\rho_g}{\rho_l} \left(\frac{x}{1-x} \frac{\rho_l}{\rho_g} \right)^n \left(\frac{\rho_l}{\rho_g} \right)^{0.12} \left(1 - \frac{\rho_g}{\rho_l} \right)^6 \left(1 + \frac{5}{0.1 + Fr} \right)^m}$$

$$n = 0.32 \left(\frac{\rho_l}{\rho_g} \right)^{0.125}$$

$$m = 6.7 \left(\frac{p}{p_{cr}} \right) \left(1 - \frac{p}{p_{cr}} \right)^4$$

(b) Slug Flow

Wadekar and Kenning [1990]

$$h_{slug} = h_p \left[\frac{L_p}{L_p + L_s} \right] + h_s \left[\frac{L_s}{L_p + L_s} \right]$$

$$h_p = 0.0038 \text{Re}_f^{0.4} \left[\frac{g k_l^3 \rho_l}{\mu_l} \right]^{0.333} \text{Pr}_l^{0.65}$$

$$h_s = 0.0023 \frac{k_l}{D} \left[\frac{\dot{m}(1-x)D}{\mu_l} \right]^{0.8} \text{Pr}_l^{0.4}$$

$$L_p = \left[\frac{80 D \dot{V}_g}{\pi u_p (D - 2\delta)^2 - 4 \dot{V}_g} \right]$$

$$L_s = 20D$$

$$\dot{V}_p = \left[1 - \frac{4\delta}{D} \right] u_p A_c$$

$$u_p = 1.2 \left[\frac{\dot{V}_g + \dot{V}_l}{A_c} \right] + 0.35 \left[\frac{gD(\rho_l - \rho_g)}{\rho_l} \right]^{0.5}$$

$$\text{Re}_l = \frac{4\dot{V}_l \rho_l}{\pi D \mu_l}$$

$$h_{slug} = h_p \left[\frac{L_p}{L_p + L_s} \right] + h_s \left[\frac{L_s}{L_p + L_s} \right]$$

Kenning and Cooper [1989]

$$j_g^* = \frac{Gx}{\left[gD\rho_g(\rho_l - \rho_g) \right]^{0.5}}$$

with Chen's correlation

$$F = 1 + 1.8 \left(\frac{1}{X_u} \right)^{0.87} \left[j_g^* \right]^{-0.5}$$

(c) Annular Flow

Authors	Correlations
Davis and David [1961]	$h_{ann} = 0.06 \left(\frac{k_l}{D} \right) \text{Re}_l^{0.87} \text{Pr}_l^{0.4} \left(\frac{\rho_l}{\rho_g} \right)^{0.28}$
Lavin and Young [1965]	$\frac{h_{ann}}{h_l} \left[\frac{\dot{m} h_{fg}}{q} \right] = C_2 \left[\frac{1+x}{1-x} \right]^{1.16}$

(c) Annular Flow (Continued)

Authors	Correlations
Rhee and Young [1974]	$\frac{h_{ann}}{h_l} = 59.03 \left(\frac{1+x}{1-x} \right)^{0.81} Fl^{0.3}$
Kenning and Cooper [1989]	with Chen's Correlation $F = 1 + 1.8 \left(\frac{1}{X_{tt}} \right)^{0.87}$

(d) Drop Flow

Lavin and Young [1965]

$$h_{drop} = \frac{k_g}{D} 0.0162 (Re)^{0.84} (Pr)^{0.33} (1-x)^{0.1}$$

APPENDIX D

Condensation Heat Transfer Coefficient Correlations

Table D.1 Condensation Heat Transfer Coefficient Correlations

#	Author	Correlation
1	Nusselt [1916]	$h_{cond} = \left[\frac{k_l^3 \rho_l (\rho_l - \rho_g) g h_{fg}}{\mu_l \Delta t_{sat} l_c} \right]^{1/4}$
2	Bromley [1952]	$h_{cond} = 0.943 \left[\frac{k_l^3 \rho_l (\rho_l - \rho_g) h_{fg} g \left(1 + 4 \frac{C_{p,l} \Delta t}{h_{fg}} \right)^2}{D \mu_l \Delta t} \right]^{0.25}$
3	Rohsenow [1956]	$h_{cond} = 0.943 \frac{k_l}{l_c} \left[\frac{l_c^3 \rho_l (\rho_l - \rho_g) g}{\mu_l \Delta t_{sat} k_l} \left[h_{fg} + 0.68 c_{p,l} \Delta t_{sat} \right] \right]^{1/4}$
4	Hassen and Jakob [1958]	$h_{cond} = 0.727 \left[\frac{\rho_l^2 h_{fg} k_l^3 \cos \beta}{\mu_l D \Delta t} \right]^{1/4}$

Table D.1 Condensation Heat Transfer Coefficient Correlations (continued)

#	Author	Correlation
5	Altman et al. [1960]	$h_{cond} = 0.057 \frac{k_i \rho_i^{0.5}}{\mu_i} Pr_i^{0.5} \tau_w^{0.5}$
6	Akers et al. [1960]	for $1000 < Re_g < 20,000$ $h_{cond} = 13.8 \frac{\mu_i C_{p_i}}{k_i} Pr_i^{0.33} \left(\frac{h_{fg}}{C_{p_i} \Delta T} \right)^{1/6} Re_g^{0.2}$ for $20,000 < Re_g < 100,000$ $h_{cond} = 0.1 \frac{\mu_i C_{p_i}}{k_i} Pr_i^{0.33} \left(\frac{h_{fg}}{C_{p_i} \Delta T} \right)^{1/6} Re_g^{0.67}$
7	Akers et al. [1961]	for $Re < 50,000$ $h_{cond} = 5.03 Pr_i^{0.33} \left(\frac{DG}{\mu_i} \right)^{1/3}$ for $Re > 50,000$ $h_{cond} = 0.0265 Pr_i^{0.33} \left(\frac{DG}{\mu_i} \right)^{0.8}$
8	Meyers and Rossen [1961]	$h_{cond} = C_4 Re^{-0.14} \left(\frac{k_i}{D} \right) \left(\frac{D^3 \rho_i^2 g}{\mu_i^2} \right)^{1/3}$ C_4 is constant depending on working fluid
9	Chato [1962]	$h_{cond} = 0.357 C_5 \left[\frac{k_i^3 \rho_i (\rho_i - \rho_g) h_{fg} g (1 + 0.68 (C_{p_i} \Delta t / h_{fg}))}{\mu_i D \Delta t} \right]^{0.25}$ C_5 is a function of angle as a graphical format. [Chato, 1962]
10	Soliman et al. [1968]	$h_{cond} = 0.036 \frac{k_i \rho_i^{0.5}}{\mu_i} Pr_i^{0.65} \tau_w^{0.5}$

Table D.1 Condensation Heat Transfer Coefficient Correlations (continued)

#	Author	Correlation
11	Traviss et al. [1973]	$h_{cond} = 0.15 \frac{Pr_i Re_i^{0.9} k_i}{F_T D} \left[\frac{1}{X_n} + \frac{2.85}{X_n^{0.476}} \right]$ $Re_i = \frac{G(1-x)D}{\mu_i}$ for $Re_i > 1125$ $F_T = 5Pr_i + 5 \ln \{1 + 5Pr_i\} + 2.5 \ln (0.0031 Re_i^{0.812})$ for $50 < Re_i < 1125$ $F_T = 5Pr_i + 5 \ln \{1 + Pr_i (0.0964 Re_i^{0.585} - 1)\}$ for $Re_i < 50$ $F_T = 0.707 Pr_i Re_i^{0.5}$
12	Ali and McDonald [1977a]	$h_{cond} = B_1 \left[\frac{k_i^3 \rho_i^2 g h_{fg}}{\mu_i D (t_{sat} - t_w)} \right]^{1/4}$ for $0^\circ < \beta < 40^\circ$; $B_1 = 0.727$ for $40^\circ < \beta < 90^\circ$; $B_1 = 0.727 \left[\cos \left(\frac{\beta - 40}{50} \right) 90 \right]^{0.25}$
13	Chen [1985]	$h_{cond} = \frac{k_i g^{1/3}}{v_i^{2/3}} \left[\left(0.31 Re_x^{-1.32} + \frac{Re_x^{2.4} Pr_i^{3.9}}{2.37 \times 10^{14}} \right)^{1/3} + \frac{A_D Pr_i^{1.3}}{771.6} (Re_{ier} - Re_x)^{1.4} Re_x^{0.4} \right]^{0.5}$
14	Shah [1979]	$h_{cond} = h_{lo} \left[(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{(P/P_\sigma)^{0.38}} \right]$ for $11 \leq G \leq 211 \text{ kg/m}^2\text{-s}$, $0 \leq x \leq 1$, $1 \leq Pr_i \leq 13$

Table D.2 Condensation Heat Transfer Coefficient Correlations on Horizontal Finned Surface

#	Author	Correlation
1	Beatty and Katz [1948]	$h_{cond,f} = 0.689 \left\{ \frac{k_l^3 \rho_l^3 g h_{fg}}{\mu_l \Delta t D_{eq}} \right\}^{1/4}$ $\left\{ \frac{1}{D_{eq}} \right\}^{1/4} = 1.30 \eta_f \frac{A_f}{A_{ef}} \frac{1}{\bar{L}^{1/4}} + \frac{A_u}{A_{ef}} \frac{1}{D_r^{1/4}}$ $\bar{L} = \pi (D_0^2 - D_r^2) / 4 D_0$ $A_{ef} = \eta_f A_f + A_u$
2	Shekrladze and Gomelaury [1966]	$h_{cond,f} = a \left\{ \frac{\rho_l (\rho_l - \rho_g) g h_{fg} k_l^3}{\mu_l \Delta t D_r} \right\}^{1/4}$
3	Smirnov and Lukanov [1972]	$h_{cond,f} = 0.689 \left\{ \frac{k_l^3 \rho_l^3 g h_{fg}}{\mu_l \Delta t D_{eq}} \right\}^{1/4}$ $\left\{ \frac{1}{D_{eq}} \right\}^{1/4} = 1.30 \eta_f \frac{A_{fs}}{A_{ef}} \frac{1}{\bar{L}^{1/4}} + \eta_f \frac{A_{fs}}{A_{ef}} \frac{1}{D_0^{1/4}} + \frac{A_u}{A_{ef}} \frac{1}{D_r^{1/4}}$ $\bar{L} = \pi (D_0^2 - D_r^2) / 4 D_0$ $A_{ef} = \eta_f A_{fs} + \eta_f A_{fn} + A_u$
4	Rudy [1982]	$h_{cond,f} = h_{bk} \left\{ \frac{\phi_f}{\pi} \right\}$ $\phi_f = \cos^{-1} \left\{ \left[\frac{2 \sigma \cos \theta}{\rho_l g b R_0} \right] - 1 \right\}$

Table D.2 Condensation Heat Transfer Coefficient Correlations on Horizontal Finned Surface
(continued)

#	Author	Correlation
5	Webb et al. [1985]	$h_{cond,f} = \frac{\phi_f}{\pi} \left(\eta_f h_f \frac{A_f}{A_{ef}} + h_u \frac{A_u}{A_{ef}} \right) + \left(1 - \frac{\phi_f}{\pi} \right) h_b$ $h_f = 2.149 \frac{k_l}{S_m} \left\{ \frac{\rho_l \sigma h_{fg} \theta_m S_m (\zeta + 1)}{\mu_l k_l \Delta t (\zeta + 2)^3} \right\}^{1/4}$ h_u is heat transfer coefficient on the unfinned region. h_b is heat transfer coefficient on the finned region.
6	Honda and Nozu [1987]	$h_{cond,f} = \frac{k_l}{D_o} \frac{Nu_{\Delta u} \eta_u (1 - \bar{t}_{wu}) + Nu_{\Delta f} \eta_f (1 - \bar{t}_{wf})}{(1 - \bar{t}_{wu}) \bar{\phi}_f + (1 - \bar{t}_{wf}) (1 - \bar{\phi}_{wf})}$ $\bar{\phi} = \phi_f / \pi \quad \text{and} \quad \bar{t}_c = (t_w - t_c) / (t_{sat} - t_c)$ $\bar{t}_{wf}$ and $\bar{t}_{wu}$ are determined from heat conduction equation.

APPENDIX E

Void Fraction Correlations

1. Lumped Models

$$\frac{1-\alpha}{\alpha} = A \left(\frac{1-x}{x} \right)^p \left(\frac{\rho_g}{\rho_l} \right)^q \left(\frac{\mu_l}{\mu_g} \right)^r$$

Authors	A	p	q	r
Homogeneous	1	1	1	0
Zivi [1964]	1	1	0.67	0
Lockhart and Martinelli [1949]	0.28	0.64	0.36	0.07
Thom [1964]	1	1	0.89	0.18
Baroczy [1965]	1	0.74	0.65	0.13
Turner and Wallace [as given by Butterworth, 1975]	1	0.72	0.4	0.08

Table E.1 Void Fraction Correlations based on Lumped Model

Table E.1 Void Fraction Correlations based on Lumped Model (Continued)

Authors	Correlations
Kowalczewski [1964 (as given by Isbin, 1979)]	for water $\alpha = \beta - k^* (1 - \beta)^{0.5} Fr_l^{-0.045} \left(1 - \frac{P}{P_{crit}} \right)^{1.0}$ $k^* = 0.8$
Kütüçüglü [1969 (as given by Isbin, 1979)]	for water $\alpha = \beta - (1 - \beta)^{0.5} Fr_l^{0.2} \left(1 - \frac{P}{P_c} \right)^{2.0}$
Premoli [1970 (as given by Isbin, 1979)]	$\alpha = \frac{1}{1 + \left(S \frac{1 - x \rho_g}{x \rho_l} \right)}$ $S = 1 + E_1 \left(\frac{y}{1 + y E_2} - y E_2 \right)^{0.5}$ $y = \frac{\beta}{1 - \beta}$ $E_1 = 1578 Re^{-0.19} \left(\frac{\rho_l}{\rho_g} \right)^{0.22}$ $E_1 = 0.0273 We Re^{-0.51} \left(\frac{\rho_l}{\rho_g} \right)^{-0.08}$

Table E.1 Void Fraction Correlations based on Lumped Model (Continued)

Authors	Correlations
Chisholm [1973]	$\frac{1}{1-\alpha} = \frac{x/\rho_g + (1-x)/\rho_l}{C_A K (1-x)/\rho_l}$ $C_A = \frac{1 + \frac{1-x}{x} \frac{\rho_g}{\rho_l}}{1 + (1-x)^{0.5} \left[\left(1 + \frac{x}{1-x} \frac{\rho_l}{\rho_g} \right)^{0.5} \left(\frac{1-x}{x} \frac{\rho_g}{\rho_l} \right) \right]}$ $K = \left(\frac{\rho_l}{\rho_H} \right)^{0.5}$
Löcher [1973 (as given by Friedel, 1977)]	$\alpha = \beta - \left(\frac{P}{P_{crit}} \right)^{-0.22} \beta^{1.39} (1-\beta)^{0.8} Fr_l^{-0.25} \left(1 - \frac{P}{P_{crit}} \right)^{3.4}$
Moussalli [1974]	$\alpha = \alpha_H \left[1 - \frac{\left(30.4 \frac{x}{1-x} \frac{\rho_g}{\rho_l} \right) + 11}{60 \left(1 + 1.6 \frac{x}{1-x} \frac{\rho_g}{\rho_l} \right) \left(1 + 3.2 \frac{x}{1-x} \frac{\rho_g}{\rho_l} \right)} \right]$
Madsen [1974]	$\alpha = \frac{1}{1 + \left(\frac{\rho_l}{\rho_g} \right)^{0.5} \left(\frac{1-x}{x} \right)^m}$ $m = \frac{0.5 \log \left(\frac{\rho_l}{\rho_g} \right) - \log \left[\frac{\alpha_H}{1-\alpha_H} \right]}{\log \left(\frac{\rho_l}{\rho_g} \right) - \log \left[\frac{\alpha_H}{1-\alpha_H} \right]}$

Table E.1 Void Fraction Correlations based on Lumped Model (Continued)

Authors	Correlations
Yamazaki [1976 (as given by Isbin, 1979)]	$\frac{\alpha}{(1-\alpha)(1-h\alpha)} = \frac{\rho_l}{\rho_g} \frac{x}{1-x}$
Friedel [1975 (as given by Isbin, 1979)]	$\alpha = \beta - 0.539(1-\beta)^{0.5} Fr_l^{-0.321} \left(1 - \frac{P}{P_c}\right)^{1.016}$ valid for $0.12 \leq P/P_{crit} \leq 0.57$ $1 \leq Fr_l \leq 88$

2. Sectorial Models

(a) Bubbly Flow

Authors	Correlations
Armand [1946]	$\frac{R_g}{R_l} = \frac{1}{0.2 + 1.2(Q_l / Q_g)}$
Bankoff [1960]	$\alpha = (0.71 + 0.015P)\alpha_H$
Hughmark [1962]	$\alpha = K\alpha_H$ K is determined by z $z = \frac{Re^{1/6} Fr^{1/8}}{y_L^{1/4}}$ $Re = \frac{D_p G}{(1-\alpha)\mu_l + \alpha\mu_g}, \quad Fr = \frac{G^2 \left[\frac{x}{\rho_g} + \frac{1-x}{\rho_l} \right]^2}{gD_p}$ $y_L = (\dot{m}_l / \rho_l) (\dot{m}_l / \rho_l + \dot{m}_g / \rho_g)$
Wallis [1969]	$(\alpha) = \frac{(\beta)}{C_0 + \bar{u}_{g,j} / (\bar{j})}$ $\beta = \frac{1}{1 + \left[(1-x)\rho_g / x\rho_f \right]}$ $j = u_{fo} \left[1 + x \left(\frac{\rho_f}{\rho_g} - 1 \right) \right]$ $C_0 = 1.0, \quad \bar{u}_{g,j} = 1.53(1-\alpha)^2 \left[\frac{\sigma g (\rho_f - \rho_g)}{\rho_f} \right]^{1/4}$

(a) Bubbly Flow (Continued)

Authors	Correlations
Rouhani [1969]	$\alpha = \frac{x}{\rho_g} \left\{ C_0 \left(\frac{x}{\rho_g} + \frac{1-x}{\rho_l} \right) + \frac{\bar{u}_g}{G} \right\}^{-1}$ $C_0 = 1.0 + 0.12(1-x)$ $\bar{u}_g = 1.18 \left[\frac{g\sigma(\rho_l - \rho_g)}{\rho_l^2} \right]^{0.25}$

(b) Slug Flow

Authors	Correlations
Armand [1946]	$\frac{R_g}{R_l} = \frac{1}{0.2 + 1.2(Q_l / Q_g)}$
Wallis [1969]	$C_0 = 12, \quad \bar{u}_{g,s} = 0.35 \left[\frac{Dg(\rho_f - \rho_g)}{\rho_f} \right]^{1/2}$
Bonnecaze et al [1969]	$\eta = 1 - \frac{(1-\lambda)}{1.2 + 0.35 \left(1 - \frac{\rho_g}{\rho_l} \right) / \delta \sqrt{\text{Re}}}$

(c) Annular Flow

Authors	Correlations
Smith [1970]	$\alpha = \frac{1}{1 + \frac{1-x}{x} \frac{\rho_g}{\rho_l} \left[K - (1-K) \left(\frac{\frac{\rho_g}{\rho_l} + \frac{1-x}{x}}{1 + \frac{1-x}{x}} \right)^{0.5} \right]}$
Ishii [1975]	$(\alpha) = \frac{(\beta)}{C_0 + \bar{u}_{g,J} / (\bar{j})}$ $C_0 = 1.0, \quad \bar{u}_{g,J} = 23 \left(\frac{\mu_l j_l}{\rho_g D} \right)^{1/2} \frac{(\rho_l - \rho_g)}{\rho_l}$
Tandon et al [1985]	for $50 < Re_l < 1125$ $\alpha = \left[1 - 1.928 Re_l^{-0.315} F(X_n)^{-1} + 0.9293 Re_l^{-0.63} F(X_n)^{-2} \right]$ for $1125 < Re_l$ $\alpha = \left[1 - 0.38 Re_l^{-0.088} F(X_n)^{-1} + 0.0361 Re_l^{-0.176} F(X_n)^{-2} \right]$ $F(X_n) = 0.15 \left[X_n^{-1} + 2.85 X_n^{-0.476} \right]$
Chen and Spedding [1981]	$\bar{R}_g = \frac{1}{1 + \chi^{2/3}} \quad \text{and} \quad \bar{R}_l = \frac{\chi^{2/3}}{1 + \chi^{2/3}}$

(c) Annular Flow (Continued)

Author	Correlations
Klidonas and Wallis [1985]	$\alpha = \frac{1}{\left\{ 1 + \frac{b}{a} E_f + \frac{b}{a} (1 - E_f) \left(\frac{a + b E_f}{1 + b E_f} \right)^{0.5} \right\}}$ $a = \frac{\rho_l}{\rho_g} \quad \text{and} \quad b = \frac{1 - x}{x}$ $E_f = 1 - \beta \exp \left\{ -0.370 - 30 \left(\frac{x G \mu_g}{\rho_g \sigma} \right) \left(\frac{\rho_l}{\rho_g} \right)^{-0.5} \right\}$

APPENDIX F

Pressure Drop Correlations

1. Lumped Models

Author	Correlation
Homogeneous Model [1942]	$\phi_{lo}^2 = \left(1 + x \left(\frac{\rho_l}{\rho_g} - 1 \right) \right) \left(1 + x \left(\frac{\mu_l}{\mu_g} - 1 \right) \right)^{-1/4}$
Martinelli-Nelson [1948]	$\phi_{lo}^2 = f(x, p)$ ϕ_{lo}^2 is defined as a function of quality and pressure in graphical format.
Lockhart and Martinelli [1949]	Correlation is given below.
Levy [1952]	for viscous-liquid/viscous-gas $\phi_l^2 = \left(\frac{Y}{Y-1} \right)^2, \quad Y = 1 + X_w \frac{\mu_g}{\mu_l} + X_w \sqrt{1 + \left(X_w \frac{\mu_g}{\mu_l} \right)^2}$ $X_w = \left(\frac{1-x}{x} \frac{\rho_g}{\rho_l} \frac{\mu_l}{\mu_g} \right)^{0.5}$

Lockhart and Martinelli's Frictional Pressure Drop [1949]

$$\phi^2 = \frac{(dp/dz)_f}{(dp/dz)_l}$$

Liquid (turbulent or viscous; laminar) Gliquid (turbulent or viscous; laminar)	tt	vt	tv	vv
$Re_{tp} = 4w_l/\pi D_p \mu_l$	> 2000	< 1000	> 2000	< 1000
$Re_{gp} = 4w_g/\pi D_p \mu_g$	> 2000	> 2000	< 2000	< 2000
r	0.2	1.0	0.2	1.0
s	0.2	0.2	1.0	1.0
C_l	0.046	16	0.046	16
C_g	0.046	0.046	16	16
χ	$\frac{w_l}{w_g} \left(\frac{\rho_g}{\rho_l} \right)^{0.355} \left(\frac{\mu_l}{\mu_g} \right)^{0.111}$	$Re_{tp}^{0.25} \left(\frac{C_l}{C_g} \right) \left(\frac{w_l}{w_g} \right) \left(\frac{\rho_g}{\rho_l} \right) \left(\frac{\mu_l}{\mu_g} \right)$	$Re_{gp}^{0.25} \left(\frac{C_l}{C_g} \right) \left(\frac{w_l}{w_g} \right) \left(\frac{\rho_g}{\rho_l} \right) \left(\frac{\mu_l}{\mu_g} \right)$	$\left(\frac{w_l}{w_g} \right) \left(\frac{\rho_g}{\rho_l} \right) \left(\frac{\mu_l}{\mu_g} \right)$
$X^2 = (dp/dz)_f/(dp/dz)_g$	$X_u = \chi_u^{0.9}$	$X_v = \sqrt{\chi_v}$	$X_v = \sqrt{\chi_v}$	$X_w = \sqrt{\chi_w}$
$\phi_l^2 = (dp/dz)_f/(dp/dz)_l$	$\phi_{l,u}^2$	$\phi_{l,v}^2$	$\phi_{l,v}^2$	$\phi_{l,vv}^2$
$\phi_g^2 = (dp/dz)_f/(dp/dz)_g$	$\phi_{g,u}^2$	$\phi_{g,v}^2$	$\phi_{g,v}^2$	$\phi_{g,vv}^2$

Author	Correlation
Levy [1952]	for viscous-liquid/viscous-gas $\phi_l^2 = \frac{1}{(1 - \alpha)^2}$ for turbulent-liquid/turbulent-gas $\phi_l^2 = \left(\frac{7}{120} \right)^{7/4} \frac{1}{(1 - \alpha^{1/2}) \left(0.125 - \frac{1}{15} (1 - \alpha^{1/2}) \right)^{7/4}}$
Jakob et al. [1956]	$dp = \frac{\Delta p_o}{L_b} \left[\frac{(0.795 \times 10^5) (x/L_b)^{1.93-0.37 \log p} L^{1.93-0.37 \log p}}{p^{1.04}} + 6 \right] dL$ Correlation include total pressure drop (frictional, acceleration, gravitational).
Chenoweth and Martin [1955]	$\phi_{lo}^2 = f(\beta, \Gamma^2)$ ϕ_{lo}^2 is defined as a function of liquid fraction and property index. Γ^2 is physical property parameter.
Chisholm and Laird [1958]	$\phi_l^2 = 1 + \frac{C}{X} + \frac{1}{X^2}$
Thom [1964]	$\phi_{lo}^2 = f(x, p)$ ϕ_{lo}^2 is defined as a function of quality and pressure in graphical format.
Broczy [1966]	ϕ_{lo}^2 is defined as a function of quality, mass flow rate and property index in graphical format.
Chisholm [1967]	$\phi_l^2 = 1 + \frac{C}{X} + \frac{1}{X^2}$ where $C = \left(\frac{\rho_l}{\rho_g} \right)^{1/2} + \left(\frac{\rho_g}{\rho_l} \right)^{1/2}$

Author	Correlation
Grønnerud [1972]	$\left(\frac{dp}{dz}\right)_w = f_{Fr} \left(x + 4 \left(x^{1.8} - x^{10} \sqrt{f_{Fr}} \right) \right)$ $f_{Fr} = 1.0, \text{ for } Fr_{lo} \geq 1.0$ $f_{Fr} = Fr_{lo}^{0.3} + 0.0055 \left(\ln(1 / Fr_{lo}) \right)^2, \text{ for } Fr_{lo} < 1.0$
Chisholm [1978]	$\phi_{lo}^2 = 1 + (\Gamma^2 - 1) \left[B x^{(2-n)/2} (1-x)^{(2-n)/2} + x^{(2-n)} \right]$ $\Gamma^2 = \frac{CT - 2^{2-n} + 2}{\Gamma^2 - 1}$
Friedel [1979]	$\phi_{lo} = \frac{(dp_F / dz)}{(dp / dz)_{lo}}, \quad \phi_{lo} = E + \frac{3.24 FH}{Fr^{0.045} We^{0.035}}$ $-\left(\frac{dp_F}{dz}\right)_{lo} = \frac{2 f_{lo} \dot{m}^2}{D \rho_l}, \quad f_{lo} = 0.079 Re_{lo}^{-1/4}$ $E = (1-x)^2 + x^2 \left[\frac{\rho_l f_{go}}{\rho_g f_{lo}} \right], \quad F = x^{0.78} (1-x)^{0.24}$ $H = \left(\frac{\rho_l}{\rho_g} \right)^{0.91} \left(\frac{\mu_g}{\mu_l} \right)^{0.19} \left(1 - \frac{\mu_g}{\mu_l} \right)^{0.7}$ $We = \frac{\dot{m}^2 D}{\rho_{TP} \sigma}$
Chen and Spedding [1981]	$\phi_g^2 = 4050 Re_{sg}^{-0.91} Re_{sl}^{0.44}$

2. Sectorial Models

(a) Bubbly Flow

Authors	Correlation
Baker [1954]	$\phi_g = 7.34 \frac{X^{0.75}}{G_l^{0.1}}$
Hoogendoon and Buitelaar [1961]	$\phi_{lo}^2 = \frac{\lambda_{l20}}{\lambda_0}$ λ_{l20}/λ_0 is presented as a graphical format to a base of gas to liquid density ratio. Author recommends that the correlation can be used in bubbly and slug flow pattern based on their graphical chart.
Govier and Aziz [1972]	$Re_{GA} = \frac{Re_l}{1 - \alpha}$ $\phi_l^2 = \frac{1}{(1 - \alpha)^{2-n}}$
Beattie [1973]	$f = \frac{\rho_l D(dp/dz)}{2G^2 \left\{ 1 + X \left(\frac{\rho_l}{\rho_g - 1} \right) \right\}}$ $Re = \frac{DG}{\mu_l} \frac{1 + X(\rho_l/\rho_g - 1)}{1 + X \left(\frac{(35\mu_g + 2\mu_l)\rho_l}{(\mu_g + \mu_l)\rho_g} - 1 \right)}$ $\phi_{lo}^2 = \left\{ 1 + X \left(\frac{\rho_l}{\rho_g} - 1 \right) \right\}^{0.8} \left\{ 1 + X \left(\frac{(35\mu_g + 2\mu_l)\rho_l}{(\mu_g + \mu_l)\rho_g} - 1 \right) \right\}^{0.2}$

(b) Slug Flow

Authors	Correlation
Baker [1954]	$\phi_s^2 = \frac{1920 X^{1.63}}{G_l}$
Bonnecaze et al. [1969]	$\left(\frac{dp}{dx}\right)_{slug} = \left(\frac{\eta - \eta_F}{1 - \eta_F}\right) \left(\frac{2\rho_l f_l u_{NS}^2}{g_c D}\right)$ $u_{NS} = (1 - \eta_F)u_T + u_F \eta_F$ $\eta = 1 - \frac{(1 - \lambda)}{120 + 0.35(1 - \rho_g/\rho_l)/(\delta\sqrt{Fr})}$ $\eta_F = \frac{C_1 - 1}{C_1}$

(c) Annular Flow

Author	Correlation
Chien and Ibele [1962]	$\phi_g^2 = 3.885 \times 10^{-6} \text{Re}_g^{0.710} \text{Re}_l^{0.725}$
Beattie [1973]	$f = \frac{\rho_l D(dp/dz)}{2G^2 \left\{1 + X \left(\frac{\rho_l}{\rho_g - 1}\right)\right\}}$ $\text{Re} = \frac{DG}{\mu_l} \frac{1 + X(\rho_l/\rho_g - 1)}{1 + X \left(\frac{\rho_l \mu_g}{\rho_g \mu_l} - 1\right)}$ $\phi_{lo}^2 = \left\{1 + X \left(\frac{\rho_l}{\rho_g} - 1\right)\right\}^{0.8} \left\{1 + X \left(\frac{\rho_l \mu_g}{\rho_g \mu_l} - 1\right)\right\}^{0.2}$

(c) Annular Flow (Continued)

Author	Correlation
Hughmark [1973]	$\left(\frac{dp}{dz}\right)_f = 0.005 \left(1 + 149 \frac{y}{R}\right) \frac{2u_g^2 \rho_g}{D_c}$ y/R is defined from their empirical relations.

Chisholm [1985]

$$\psi = \frac{\phi_{lo}^2 - 1}{\Gamma^2 - 1}$$

$$\psi = Bx^{\frac{2-n}{2}}(1-x)^{\frac{2-n}{2}} + x^{2-n}$$

n is Blasius exponent

$$B = \frac{G}{G_{GM}} \left[\frac{G_{GM}}{G - G_{GM}} \right]^{0.9} = \frac{1}{X_M} \left[\frac{X_M}{1 - X_M} \right]^{0.9}$$

$$G = 173E_p \left[E_{\max}^{0.4} \frac{\alpha}{1 - 1/K} \frac{1}{X_M(1 - X_M)^{0.1}} \right]^{0.9091}$$

$$E_p = \frac{\rho_R^{0.605} \mu_l^{0.09091} \sigma^{0.4545}}{(\rho_l - \rho_g)^{0.1515} d^{0.5454}}$$

$$X_M = \frac{G_{GM}}{G}$$

$$G = 173E_p \left[E_{\max}^{0.4} \frac{\alpha}{1 - 1/K} \left(\frac{X_M}{1 - X_M} \right)^{0.1} \right]^{0.9091}$$

$$\frac{\varepsilon}{D} = \frac{659F}{(1+1400F)^{1/2}} = \frac{\delta}{D}$$

$$F = \frac{\gamma(\text{Re}_l)}{(\text{Re}_s)^{0.9}} \frac{\mu_s}{\mu_l} \sqrt{\frac{\rho_l}{\rho_s}}$$

$$\gamma = \left\{ \left[0.707(\text{Re}_l)^{0.5} \right]^{2.5} + \left[0.0379(\text{Re}_s)^{0.9} \right]^{2.5} \right\}^{0.4}$$

$$\rho_c = \left[\frac{\dot{m}_g + e\dot{m}_l}{\dot{m}_g} \right] \rho_g$$

$$-\left(\frac{dp}{dz}\right)_{a-f} = \frac{\rho_{gc} f_{a-f} V_{gc}^2}{2D}$$

$$V_{gc} = \frac{e\dot{m}_l + \dot{m}_g}{\rho_{gc} A_{gc}}$$

$$A_{gc} = \frac{\pi D_{gc}^2}{4}$$

$$D_{gc} = D - 2\delta$$

$$E = 1 - \exp(-0.125(\beta - 1.5))$$

$$\beta = \frac{3048 j_{sg} \mu_s}{\sigma} \left(\frac{\rho_s}{\rho_l} \right)^{0.5}$$

3. Pressure Drop Correlations due to Two-Phase Flow Obstruction Area change

Authors	Correlation
Lottes [1961]	$\Delta p_{\text{expansion}} = \frac{G^2 \sigma^2}{\rho_l g_c} \left[\frac{\rho_l}{\rho_g} x^2 \left(\frac{1}{a_1 \sigma} - \frac{1}{a_4} \right) + \right.$ $\left. (1-x)^2 \left(\frac{1}{\sigma(1-a_1)} - \frac{1}{(1-a_4)} \right) \right]$
Jassen [1966]	$\Delta p_{\text{contraction}} = \frac{G^2}{2\rho_l g_c} \left[\frac{1}{\sigma^2} \left\{ \frac{\rho_l}{\rho_g} x^2 \bar{a}_1 \left(\frac{1}{a_1 \sigma} - \frac{1}{a_4} \right) + \right. \right.$ $\left. (1-x)^2 (1-\bar{a}_1) \left(\frac{1}{C^2(1-a_3)^2} - \frac{1}{(1-a_4)^2} \right) \right\}$ $- \frac{2}{\sigma^2} \left\{ \frac{\rho_l}{\rho_g} x^2 \bar{a}_1 \left(\frac{1}{C a_3} - \frac{1}{a_4} \right) + (1-x)^2 \left(\frac{1}{C(1-a_3)} - \frac{1}{a_4} \right) \right\}$ $\left. + \frac{\rho_l}{\rho_g} x^2 \bar{a}_2 \left(\frac{1}{C a_4} - \frac{1}{a_1^2} \right) + (1-x)^2 (1-\bar{a}_2) \left(\frac{1}{\sigma^2(1-a_4)^2} - \frac{1}{(1-a_1)^2} \right) \right]$
Geiger and Rohrer [1966]	$\Delta p_{\text{crit}} = \frac{G_3^2}{2g_c \bar{\rho}} (1 - \sigma^2 + K_{TPC})$ K_{TPC} is defined in Ref. Geiger and Rohrer [1966] as a complex equation.
Chisholm and Sutherland [1967]	$\phi_{lo}^2 = 1 + \frac{C}{X} + \frac{1}{X^2}$ $C = \left(1 + (C_2 - 1) \left(\frac{\rho_l - \rho_g}{\rho_l} \right)^2 \right) \left(\left(\frac{\rho_l}{\rho_g} \right)^2 + \left(\frac{\rho_g}{\rho_l} \right)^2 \right)$ for contraction, $C_2 = 1$, for expansion, $C_2 = 0.5$, for 90o bends, $C_2 = 1 + 35(D/L)$, for 180o bends, $C_2 = 1 + 20(D/L)$,

Authors	Correlation
Beattie [1973]	$K = \frac{\rho_l D (dp / dz)}{2G^2 \left\{ 1 + X \left(\frac{\rho_l}{\rho_g} - 1 \right) \right\}}$ $Re = \frac{DG}{\mu_l} \frac{1 + X(\rho_l / \rho_g - 1)}{1 + X \left(3.5 \frac{\rho_l}{\rho_g} - 1 \right)}$ $\phi_{lo}^2 = \left\{ 1 + X \left(\frac{\rho_l}{\rho_g} - 1 \right) \right\}^{0.8} \left\{ 1 + X \left(\frac{3.5 \rho_l}{\rho_g} - 1 \right) \right\}^{0.2}$
Chisholm [1983]	$\phi_{lo}^2 = 1 + (\Gamma^2 - 1) \left[Bx^{(2-n)/2} (1-x)^{(2-n)/2} + x^{(2-n)} \right]$ $B = 1 + \frac{2.2}{k_{lo}(2 + R/D)} \quad , \quad k_{lo} = f\left(\frac{z}{D}\right)$ (z/D) is evaluated from graphical chart.
Chisholm [1983]	$\phi_{lo}^2 = 1 + \frac{C_B}{X} + \frac{1}{X^2}$ $C_B = \frac{1}{\Gamma_B^2} \left[\left\{ 1 + \frac{2.2}{k_{lo}(2 + R/D)} \right\} (\Gamma_B^2 - 1) + 2^{2-n} - 2 \right]$

4. Pressure Drop Correlations for Orifice

Authors	Correlation
Baroczy [1965]	$\phi_{lo}^2 = \left[\frac{\rho_l X^2}{\rho_g R_g} + \frac{(1-x)^2}{R_l} \right]$ $R_l = \left(\frac{X_n^{111}}{X_n^{111} + 1} \right)$
Beattie [1973]	$K = \frac{\rho_l D(dp/dz)}{2G^2 \left\{ 1 + X \left(\frac{\rho_l}{\rho_g - 1} \right) \right\}}$ $Re = \frac{DG}{\mu_l} \frac{1 + X(\rho_l/\rho_g - 1)}{1 + X \left(\frac{\rho_l \mu_g}{\rho_g \mu_l} - 1 \right)}$ $\phi_{lo}^2 = \left\{ 1 + X \left(\frac{\rho_l}{\rho_g} - 1 \right) \right\}^{0.8} \left\{ 1 + X \left(\frac{\rho_l \mu_g}{\rho_g \mu_l} - 1 \right) \right\}^{0.2}$
Strathclyde [1983]	$\phi_{lo}^2 = \left(1 + x \left(\frac{\rho_l}{\rho_g} \right)^{1/6} - 1 \right) \left(1 + x \left(\frac{\rho_l}{\rho_g} \right)^{5/6} - 1 \right)$
Fairhurst [1983]	$\phi_{lo}^2 = \left(1 + \left(\frac{\rho_l}{\rho_g} \right) \left(\frac{x}{2-x} \right) \right)$ $k_{go} = k_{lo}$
Sadatomi and Sato [1988]	$\phi_{lo}^2 = \frac{1}{Y_g^2} \frac{\rho_l}{\rho_g} \frac{x^2}{a} + \frac{(1-x)^2}{(1-a)}$

APPENDIX G

Thermal Resistances

The heat transfer flow rate of the system is defined as:

$$Q = \frac{\Delta t}{\sum R}$$

Thermal Resistances in loop systems are similar with each other. Those depend on how to heat and how to cool.

1. Thermal resistance through the filler (SSL)

$$R_{fil} = \frac{\delta_{fil}}{k_{fil} A_h}$$

2. Thermal resistance through the plate (SSL)

$$R_{pl} = \frac{1}{k_{pl} S}$$

where S is shape factor as defined as in Appendix J

3. Thermal resistance due to the boiling (All TLTs)

$$R_{ev} = \frac{1}{h_{tp} \pi d L_{ev}}$$

4. Thermal resistance due to the condensation (all TLTs)

$$R_{cond} = \frac{1}{h_{cond} \pi d L_{cond}}$$

5. Thermal resistance through the tube (all TLTs)

$$R_{ct} = \frac{\ln(D/d)}{2\pi k_t L}$$

where L can be the length of evaporator or condenser.

6. Thermal resistance due to the forced convection through a finned area. (SSL, MSL I, MSL II, LSL I, LSL II) - this thermal resistance can be from hot or cold side.

$$R_{conv} = \frac{1}{h_{conv} \eta_T A_{conv}}$$

where η_T is defined as Eq. (I.1) in Appendix I.

7. Thermal resistance due to the forced convection through a bare tube (MSL II)

$$R_{conv} = \frac{1}{h_{conv} A_{conv}}$$

APPENDIX H

Correlations for Forced and Natural Convection Heat Transfer Coefficient

1. Empirical correlations for forced convection from a finned area

a. Knudsen and Katz [Holman, 1996]

$$\frac{hd}{k_f} = C \left(\frac{u_\infty D}{\nu_f} \right)^n \text{Pr}^{1/3} \quad (\text{H.1})$$

$$C = 0.683, n = 0.466 \text{ for } 40 < \text{Re} < 4000$$

b. Fand [Holman, 1996]

$$\text{Nu}_f = (0.35 + 0.65 \text{Re}_f^{0.52}) \text{Pr}_f^{0.3} \quad (\text{H.2})$$

c. Churchill and Bernstein [Holman, 1996]

$$\text{Nu} = 0.3 + \frac{0.62 \text{Re}^{1/2} \text{Pr}^{1/3}}{[1 + (0.4 / \text{Pr})^{2/3}]} \left[1 + \left(\frac{\text{Re}}{282,000} \right)^{5/8} \right]^{4/5} \quad (\text{H.3})$$

2. Empirical equations for natural convection without air flow in the condenser section [Bejan, 1993]

$$\overline{Nu} = \frac{4}{3} \left[\frac{7 Ra Pr}{5(20 + 21 Pr)} \right]^{1/4} + \frac{4(272 + 315 Pr) L_c}{35(64 + 63 Pr) D} \quad (H.4)$$

APPENDIX I

Fin Efficiency

The total fin efficiency is defined as [Hewitt, 1994]:

$$\eta_T = 1 - \frac{A_{fin}}{A_T} (1 - \eta_{fin}) \quad (I.1)$$

$$A_T = A_{fin} + A_{br} \quad (I.2)$$

$$A_{br} = \pi D (L_c - N_{fin} \delta_{fin}) \quad (I.3)$$

$$\eta_{fin} = \frac{\tanh(m\Psi)}{m\Psi} \quad (I.4)$$

$$m = \sqrt{\frac{2h_{conv}}{k_{fin} \delta_{fin}}} \quad (I.5)$$

$$h_{conv} = \frac{Nuk}{D} \quad (I.6)$$

$$\Psi = \frac{D_i}{2} \left[\left(\frac{r_e}{r_i} \right) - 1 \right] \left[1 + 0.35 \ln \left(\frac{r_e}{r_i} \right) \right] \quad (I.7)$$

a. for annular fins

$$\frac{r_e}{r_i} = \frac{r_o}{r_i} \quad (1.8)$$

b. for rectangular fins

$$\frac{r_e}{r_i} = 128 \frac{L_{horizontal}}{D_o} \sqrt{\frac{L_{horizontal}}{L_{vertical}}} - 0.2 \quad (1.9)$$

APPENDIX J

Shape Factor

For a rectangular pipe buried in semi-infinite medium having isothermal surface as shown in Fig. J.1 [Kutateladze, 1963].

$$S = \frac{\left[5.7 + \frac{b}{2a} \right]}{\ln \left(\frac{3.5h}{b^{0.25}a^{0.75}} \right)} \quad (J.1)$$

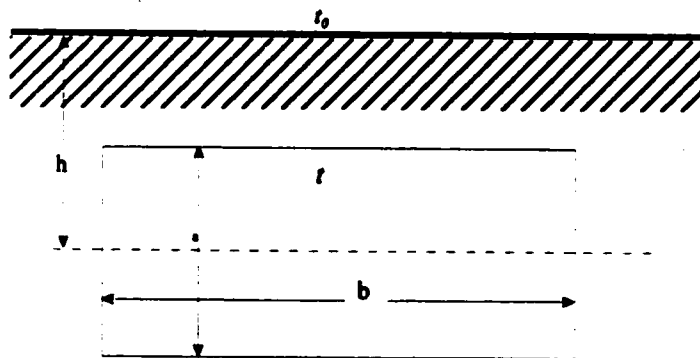


Figure J.1

APPENDIX K

Annular Flow Pressure Drop Calculation Models

The present annular flow frictional pressure drop calculation model is mainly based on Yao and Sylvester [1987]'s model. Entrainment model developed by Ishii and Mishima [1982] as applied for entrained liquid fraction calculation.

Minimum Reynolds number which the entrainment can occur. [Ishii and Grolmes, 1975]

$$(Re_f)_{\min} = \left(\frac{y^*}{0.347} \right)^{3/2} \left(\frac{\rho_f}{\rho_g} \right)^{3/4} \left(\frac{\mu_g}{\mu_f} \right)^{3/2} \quad (K.1)$$

If Reynolds number satisfied the minimum Reynold number, the entrainment is calculated by,

$$e = \tanh \left[7.25 \times 10^{-7} (j_g^*)^{2.5} (D^*)^{1.25} (Re_f)^{0.25} \right] \quad (K.2)$$

$$j_g^* = j_g \left[\frac{\rho_g^{4/3}}{\sigma g (\rho_f - \rho_g)^{1/3}} \right]^{1/4} \quad (K.3)$$

$$D^* = D \left[\frac{g(\rho_f - \rho_g)}{\sigma} \right]^{1/2} \quad (K.4)$$

$$\text{Re}_f = \frac{\rho_f j_f D}{\mu_f} \quad (\text{K.5})$$

Zigrang and Sylvester [1982] suggested the friction factor relationship for annular flow in vertical pipe as shown below.

$$\frac{1}{\sqrt{f_{a-f}}} = -2.0 \log \left[\frac{\varepsilon / D}{3.7} - \frac{5.02}{\text{Re}_{gc}} \log \left(\frac{\varepsilon / D}{3.7} + \frac{13}{\text{Re}_{gc}} \right) \right] \quad (\text{K.6})$$

Henstock and Hanratty [1976] developed the relation of the time averaged thickness of the annular liquid film to the pipe diameter.

$$\frac{\varepsilon}{D} = \frac{659F}{(1 + 1400F)^{1/2}} = \frac{\delta}{D} \quad (\text{K.7})$$

$$F = \frac{\gamma(\text{Re}_l)}{(\text{Re}_g)^{0.9}} \frac{\mu_g}{\mu_l} \sqrt{\frac{\rho_l}{\rho_g}} \quad (\text{K.8})$$

$$\gamma = \left\{ \left[0.707(\text{Re}_l)^{0.5} \right]^{2.5} + \left[0.0379(\text{Re}_g)^{0.9} \right]^{2.5} \right\}^{0.4} \quad (\text{K.9})$$

$$\text{Re}_l = \frac{\dot{m}_l D}{A \mu_l} \quad (\text{K.10})$$

$$\text{Re}_g = \frac{\dot{m}_g D}{A \mu_g} \quad (\text{K.11})$$

$$\rho_c = \left[\frac{\dot{m}_g + e \dot{m}_l}{\dot{m}_g} \right] \rho_g \quad (\text{K.12})$$

$$-\left(\frac{dp}{dz}\right)_{a-f} = \frac{\rho_{gc} f_{a-f} V_{gc}^2}{2D} \quad (\text{K.13})$$

$$V_{gc} = \frac{e\dot{m}_l + \dot{m}_g}{\rho_{gc} A_{gc}} \quad (\text{K.14})$$

$$A_{gc} = \frac{\pi D_{gc}^2}{4} \quad (\text{K.15})$$

$$D_{gc} = D - 2\delta \quad (\text{K.16})$$

APPENDIX L

Geometrical Considerations

The five TLTs studied have different geometries, and in the simulation the geometrical differences of TLTs are considered carefully and described in detail, here.

1. Small Scale Loop (SSL)

(a) Evaporator Section

The evaporator section of the SSL was designed to attain the best heat transfer performance to satisfy the design goal. As mentioned in Chapter 3, the evaporator section was designed as a square box heated only one side. It is very difficult to simulate the evaporating process which occurs in such a square box as an evaporator section. Also it was not possible to find any suitable heat transfer correlation for the particular geometry from open literatures. In the simulation, the evaporator section was dealt as a pipe equivalent for the heat transfer surface area.

(b) Transport Section

Transport section consists of vertical and horizontal pipe lines. The vertical line is connected

to the evaporator outlet and the end of horizontal line connected into the condenser inlet. In the simulation, the flow regime in the vertical line is considered as annular flow with entrainment.

(c) Condenser Section

The condenser section has three vertical pipes. To simulate the condenser section, it is assumed that the vapor flow is condensed upon entering the condenser section and a film condensation process starts. In the simulation, the heat transfer correlation for the film wise condensation are applied, and for the calculation of the pressure drop, the flow is considered as that of single phase liquid flow.

(d) Condensate Return Line

The main pressure drop in the condensate return line is hydrostatic due to the gravitational force. The pressure drop calculation for the condensate return line is carried out by the correlation for a single phase liquid flow pressure drop correlation.

2. Medium Scale Loop I (MSL I)

The MSL I has a quite different configuration from other TLTs, that is:

- simple evaporator sections, but the two-phase flow in such systems has rarely been studied.
- short transporting pipe line, and
- complex condenser section as shown in Fig. 3.9.

(a) Evaporator Section

As shown in Fig. 3.10 and 3.11, the evaporator sections consist of inclined pipe with 6 (case I) and 4 (case II) bends. The study on the evaporating process in such inclined pipes with many bends is extremely rare. In the present study, this particular geometries of the evaporator is simulated as a vertical pipe. In the simulation, it was not easy to obtain a convergence for the iteration process.

(b) Transport Section

It was the same as to the case of the SSL. The transport pipe has a 180° and a 90° bends and a short horizontal and vertical pipe line.

(c). Condenser Section

The condensation heat transfer in a radially finned tube was not widely studied and only a few studies could be found in the literature which could be used for the simulation. The details are given in Appendix D.

(d) Condensate Return Line

The condensate return line is similar with that of the SSL and the same method for the SSL was used.

3. Medium Scale Loop II (MSL II)

As shown in Fig3.12, the MSL II is a usual rectangular loop thermosyphon, the kind of TLTs experimentally well studied by many researchers.

(a) **Evaporator Section**

The evaporator section is a simple vertical pipe with electrical heating wires. The simulation for the evaporator section is made with the correlations given in the previous sections for two-phase parameters.

(b) **Transport Section**

The simulation program is the same as that with the SSL.

(c) **Condenser Section**

Unlike the SSL, the MSL II has a water cooling condenser section. The simulation program is similar to that for the SSL, but cooling medium is now water.

(d) **Condensate Return Line**

The condensate return line followed the method used for the SSL.

4. Large Scale Loop I (LSL I)

The LSL I is much difficult to simulate than other TLTs discussed, and such a system has never been simulated by others. The LSL I has multi-row evaporator and a multi-pipe row condenser section which require both pressure and mass balances in the simulation.

(a) **Evaporator Section**

As shown in Fig. 3.14, the evaporator section consists of 5 pipes with uniform electrical

heating, and in the simulation the two-phase flow phenomena of each evaporator pipe must be separately considered.

Heat Transfer - The heat transfer process in each evaporator pipe is calculated as in the previous TLTs. Although the heat input to each pipe is not exactly all the same as observed in the experimental program, in the simulation, it is assumed that each pipe has equal heat input.

Pressure Drop - The pressure drop calculation requires a large number of iteration process. At the working fluid inlet, the pressure drop due to the area change is calculated. And then the pressure drop of the first evaporator pipe is calculated, followed by the calculation of the pressure drop of the next pipe. The pressure drop of the 2nd evaporator pipe is compared with that of the 1st pipe. When the calculation procedure reaches the top of the center evaporator pipe, the mass flow rate is compared with the inlet mass flow rate. If the inlet mass flow rate is equal to the outlet rate under the acceptable error range which is specified, the simulation continues to the next.

(b) Transport Section

The transport pipe is similar to that of the MSL II which is dealt in the next section and the calculation procedure is the same as that of the MSL II. Only transport pipe line of LSL I is much longer than MSL II. The only difference is that the length of the transport pipeline of the LSL I is much larger than that of the MSL II.

(c) Condenser Section

The same iteration procedure of the evaporator discussed is applied to the condenser section in the simulation.

(d) **Condensate Return Line**

The condensate return line has a similar shape with that of the MSL II. Although the pipe size and length are different from those of the MSL II, the calculation method is the same as that of the MSL II.

5. Large Scale Loop II (LSL II)

The LSL II is much complex than other 4 TLTs discussed so far. The LSL II consists of five identical loops of the same geometry and size.

(a) **Evaporator Section**

As shown in Fig. 3.17, each evaporator section of 5 loops has 12 pipes, similar to that of the LSL I. The simulation procedure, however, requires much more iteration processes. For a consecutive loop calculation, the inlet temperature of the subsequent loop is calculated by Eq. (L.1) given below.

- **Parallel flow case**

$$t_n = \frac{Q_{n-1}}{\dot{m}c_p} + t_{n-1} \quad (\text{L.1})$$

- **Counter flow case**

For counter flow case, simulation needs more iteration to find the correct hot air temperature. First of all the inlet temperature into the first loop is assumed, then from

the energy balance, the inlet temperature into the subsequent loop is calculated. This temperature is compared with the input cold air temperature and the iteration process is continued until the convergence criteria which is prescribed is satisfied.

$$\left(t_{cold} - t_{cold,calc.} \right) < \varepsilon \quad (L.2)$$

(b) Transport Section

The feature of the transportation section is similar to that of the LSL I and the calculation method is the same as that of the LSL I.

(c) Condenser Section

The condenser section, has the same shape as that of the evaporator section. According to the direction of cold air flow, the simulation requires a large number of iterations. The calculation procedure for heat transfer and pressure drop is, however, the same as that of the evaporator section.

(d) Condensate Return Line

The calculation procedure for the condensate return line is the as that of other TLTs discussed above.

APPENDIX M

Thermal Property Equations for Working Fluids used in Simulation

This section presents saturation thermal property equations used in simulation. The development of subroutines to calculate the thermophysical properties is an important part to get a correct simulation result. Particularly, the property calculation of new refrigerants as a working fluid is not easy due to the lack of information. Basically the property calculation is based on Reid [1966] and the information from manufacturer. Each property equation was compared with other published data and selected data obtained from company. All property equations used in simulation are given below.

Predictions of Thermal Properties for Working Fluids Used in Simulation

1. Saturation Temperature (°C)

Working Fluid	Saturation Temperature
Water	$T_{sat} = (((-43.771989) \times (99.597506) + (695.33745) \times (P^{(0.27517693)})) / ((99.597506) + (P^{(0.27517693)})))$
Ethanol	$T_{sat} = (((-26.767508) \times (178.98719) + (480.54774) \times (P^{(0.33228706)})) / ((178.98719) + (P^{(0.33228706)})))$
Acetone	$T_{sat} = ((((-29.803217) \times (2629.6904) + (191.16595) \times (P^{(0.64173061)})) / (((2629.6904) + (P^{(0.64173061)}))))$
R-11	$T_{sat} = ((((-44.235406) \times 8658.3069 + (179.42845) \times (P^{(0.70968762)})) / ((8658.3069) + (P^{(0.70968762)})))$
R-113	$T_{sat} = ((((-28.557804) \times (1795.4564) + (395.38983) \times (P^{(0.5139248)})) / ((1795.4564) + (P^{(0.5139248)})))$
R-134a	$T_{sat} = ((-34.345962) + (0.00012767379) \times P + (-7.046796E-11) \times (P^2) + (2.0123185 \times 10^{-17}) \times (P^3) + (-2.1048157E-24) \times (P^4))$
FC-72	$T_{sat} = (((-5.9870742) \times (5593.8503) + (295.30201) \times (P^{(0.62817433)})) / ((5593.8503) + (P^{(0.62817433)})))$
FC-87	$T_{sat} = ((((-29.226318) \times (286.57539) + (311.90716) \times (P^{(0.39700504)})) / ((286.57539) + (P^{(0.39700504)})))$

2. Saturation Pressure (Pa)

Working Fluid	Saturation Pressure
Water	$-0.1 < T_{sat} < 100$ $P_{sat} = 671.80492 + 16.422933 \times t_{sat} + 3.6091499 \times (t_{sat}^2) + (-0.03363131) \times (t_{sat}^3) + (0.00096485833) \times (t_{sat}^4)$ $100 < t_{sat} < 220$ $P_{sat} = 2556.7524 + (-470.66592) \times t_{sat} + (17.525527) \times (t_{sat}^2) + (-0.17696422) \times (t_{sat}^3) + (0.0014749273) \times (t_{sat}^4)$ $220 < t_{sat} < 300$ $P_{sat} = 934213.11 + (-15555.808) \times t_{sat} + (107.02617) \times (t_{sat}^2) + (-0.40514253) \times (t_{sat}^3) + (0.0016818203) \times (t_{sat}^4)$ $300 < t_{sat}$ $P_{sat} = 475904720 + (-5774899.9) \times t_{sat} + 26306.286 \times (t_{sat}^2) + (-53.405785) \times (t_{sat}^3) + (0.041921397) \times (t_{sat}^4)$
Ethanol	$P_{sat} = \text{Exp}[-4.4114 + (0.08765) \times t_{sat} + (-0.00063182) \times (t_{sat}^2) + (0.0000039958) \times (t_{sat}^3) + (-0.00000001434) \times (t_{sat}^4) + (0.00000000020359) \times (t_{sat}^5)] \times 100000$
Acetone	$P_{sat} = [-2.3205 + (0.05455) \times t_{sat} + (-0.00020375) \times (t_{sat}^2) + (-0.0000018017) \times (t_{sat}^3) + (0.000000034981) \times (t_{sat}^4) + (-0.00000000013904) \times (t_{sat}^5)] \times 100000$
R-11	$P_{sat} = \text{Exp}[-0.90867 + (0.042922) \times t_{sat} + (-0.00018364) \times (t_{sat}^2) + (0.00000079287) \times (t_{sat}^3) + (-0.0000000029186) \times (t_{sat}^4) + (6.2237\text{E-}12) \times (t_{sat}^5)] \times 100000$
R-113	$P_{sat} = \text{Exp}[0.15513 + (0.0055834) \times t_{sat} + (0.00019778) \times (t_{sat}^2) + (0.0000028601) \times (t_{sat}^3) + (-0.000000057513) \times (t_{sat}^4) + (0.00000000053363) \times (t_{sat}^5)] \times 100000$
R-134a	$P_{sat} = 294043.01 + (10657.879) \times t_{sat} + (144.39577) \times (t_{sat}^2) + (0.88465742) \times (t_{sat}^3) + (0.0027531808) \times (t_{sat}^4)$
FC-72	$P_{sat} = 3824.2136 + (1634.44) \times t_{sat} + (-61.898341) \times (t_{sat}^2) + (1.9206007) \times (t_{sat}^3) + (-0.01987666) \times (t_{sat}^4) + (0.00010910463) \times (t_{sat}^5) + (-0.00000021827451) \times (t_{sat}^6)$
FC-87	$P_{sat} = \text{Exp}[20.184 \times \text{Exp}((12.637 - (5467 / (t_{sat} + 273))) - (7.098 \times \text{Log}((t_{sat} + 273) / 420.6)) + (0.361 \times (((t_{sat} + 273)^{20.6})^{0.6})))]] \times 100000$

3. Liquid Density (kg/m³)

Working Fluid	Equation for Liquid Density Calculation
Water	$0 < t_{sat} < 60$ $\rho_l = (999.79995 + 7.1545955 \times t_{sat}) / (1 + 0.0070924661 \times t_{sat} + 0.0000077679162 \times (t_{sat})^2)$ $60 < t_{sat} < 150$ $\rho_l = 1004.2453 + (-0.17474315) \times t_{sat} + (-0.0030916753) \times (t_{sat})^2 + 0.0000025570462 \times (t_{sat})^3$ $150 < t_{sat} < 250$ $\rho_l = 1034.2025 + (-0.74027352) \times t_{sat} + (0.00052567713) \times (t_{sat})^2 + (-0.0000053032682) \times (t_{sat})^3$ $250 < t_{sat} < 300$ $\rho_l = (914.85797) + (0.56156044) \times t_{sat} + (-0.0040989011) \times (t_{sat})^2$ $300 < t_{sat}$ $\rho_l = (-878107.61) + (10691.702) \times t_{sat} + (-48.697819) \times (t_{sat})^2 + (0.098436571) \times (t_{sat})^3 + (-0.000074528136) \times (t_{sat})^4$
Ethanol	$\rho_g = 10 \cdot (-0.00054 \times t_{sat} + 2.90841)$
Acetone	$\rho_g = \exp[6.7037 + (-0.0016696) \times t_{sat} + (-0.0000010073) \times (t_{sat})^2 + (0.00000012383) \times (t_{sat})^3 + (-0.0000000021854) \times (t_{sat})^4 + (8.5364e-12) \times (t_{sat})^5]$
R-11	$\rho_g = \exp[0.42772 + (-0.0014712) \times t_{sat} + (-0.0000008545) \times (t_{sat})^2 + (-0.0000000061066) \times (t_{sat})^3 + (-0.00000000072756) \times (t_{sat})^4 + (4.5734e-12) \times (t_{sat})^5] \times 1000$
R-113	$\rho_g = \exp[0.48109 + (-0.00070767) \times t_{sat} + (-0.000019487) \times (t_{sat})^2 + (-0.00000047513) \times (t_{sat})^3 + (0.000000021984) \times (t_{sat})^4 + (-0.00000000019019) \times (t_{sat})^5]$
R-134a	$\rho_g = 1292.4242 + (-3.1445223) \times t_{sat} + (-0.0051035925) \times (t_{sat})^2 + (-0.00010533391) \times (t_{sat})^3 + (-0.00000081678815) \times (t_{sat})^4$
FC-72	$\rho_g = 10^{(-0.0006765 \times t_{sat} + 3.242)}$
FC-87	$\rho_g = 10^{(3.25478 - (0.000185946 \times (t_{sat})))}$

4. Gas Density (kg/m³)

Working Fluid	Equation for Gas Density Calculation
Water	$\rho_g = \exp[(-5.3225) + (0.068366) \times (t_{sat}) + (-0.00027243) \times (t_{sat}^2) + (0.00000084522) \times (t_{sat}^3) + (-0.000000016558) \times (t_{sat}^4) + (1.5541e-12) \times (t_{sat}^5)]$
Ethanol	$\rho_g = (-3.3681) + (0.052492) \times (t_{sat}) + (0.00005163) \times (t_{sat}^2) + (-0.0000019542) \times (t_{sat}^3) + (0.0000000086893) \times (t_{sat}^4) + (-0.000000000011451) \times (t_{sat}^5)$
Acetone	$\rho_g = 0.174479 \times (10 \wedge (-0.0000119558 \times t_{sat} + 0.0137254 \times t_{sat} + 0.338383))$
R-11	$\rho_g = (2.4527) + (0.094781) \times (t_{sat}) + (0.0017144) \times (t_{sat}^2) + (0.000010868) \times (t_{sat}^3) + (-0.0000000067672) \times (t_{sat}^4) + (0.00000000039042) \times (t_{sat}^5)$
R-113	$\rho_g = (1.2915) + (0.043043) \times (t_{sat}) + (0.0014813) \times (t_{sat}^2) + (0.000020168) \times (t_{sat}^3) + (-0.00000050204) \times (t_{sat}^4) + (0.0000000046851) \times (t_{sat}^5)$
R-134a	$-60 < t_{sat} < 50$ $\rho_g = (14.398407) + (0.49886186) \times t_{sat} + (0.0070761927) \times (t_{sat}^2) + (0.000058359448) \times (t_{sat}^3) + (0.0000002689327) \times (t_{sat}^4)$ $50 < t_{sat} < 90$ $\rho_g = (1031.989) + (-62.095579) \times t_{sat} + (1.445171) \times (t_{sat}^2) + (-0.014620134) \times (t_{sat}^3) + (0.000056799734) \times (t_{sat}^4)$
FC-72	$\rho_g = 0.91634486 + (-0.2416168) \times t_{sat} + (0.06330763) \times (t_{sat}^2) + (-0.0037215772) \times (t_{sat}^3) + (0.00010469133) \times (t_{sat}^4) + (-0.0000015501574) \times (t_{sat}^5) + (0.000000012533098) \times (t_{sat}^6) + (-5.2197523 \times 10^{-11}) \times (t_{sat}^7) + (8.7673574e-14) \times (t_{sat}^8)$
FC-87	$\rho_g = [(2.4527) + (0.094781) \times (t_{sat}) + (0.0017144) \times (t_{sat}^2) + (0.000010868) \times (t_{sat}^3) + (-0.0000000067672) \times (t_{sat}^4) + (0.00000000039042) \times (t_{sat}^5)] \times 1.356$

5. Liquid Specific Heat (J/kg °C)

Working Fluid	Equation for Liquid Specific Heat Calculation
Water	$c_{p,l} = 4205 - 1.291 \times t_{sat} + 0.01416 \times (t_{sat}^2)$
Ethanol	$c_{p,l} = (10^{(0.001753 \times t_{sat} + 3.35)})$
Acetone	$c_{p,l} = ((2110.14 \times (10^{(0.000539228 \times t_{sat})})))$
R-11	$c_{p,l} = 528.512 \times (10^{(-0.000000432398 \times t_{sat} + 0.00066139)})$
R-113	$c_{p,l} = (0.00161 \times t_{sat} + 0.914) \times 1000$
R-134a	$-60 < t_{sat} < 50$ $c_{p,l} = (1334.6666) + (2.9713171) \times t_{sat} + (0.018872514) \times (t_{sat}^2) + (0.00016871375) \times (t_{sat}^3) + (0.0000025406702) \times (t_{sat}^4)$ $50 < t_{sat} < 95$ $c_{p,l} = (1467.873) + (-12.38465) \times t_{sat} + (-0.008889489) \times (t_{sat}^2) + (-0.0000090436561) \times (t_{sat}^3)$
FC-72	$c_{p,l} = ((0.00161 \times t_{sat} + 1) \times 1010)$
FC-87	$c_{p,l} = (10^{(0.001753 \times t_{sat} + 3.35)})$

6. Gas Specific Heat (J/kg °C)

Working Fluid	Equation for Gas Specific Heat Calculation
Water	$c_{p,g} = (1 / (3.1357533 + (-0.0061140756) \times (t_{sat,F} (0.95048645)))) \times 4186.8$
Ethanol	$c_{p,g} = 0.96650723 \times (1.6985631 - \exp(-0.05810284 \times t_{sat})) \times 1000$
Acetone	$c_{p,g} = ((0.64577407 + -0.0023245886 \times t_{sat}) (-1 / 1.9062419)) \times 1000$
R-11	$c_{p,g} = ((198.06152 + (-0.5254585) \times t_{sat,F}) (-1 / 2.5866964)) \times 4186.8$
R-113	$c_{p,g} = (0.16 + (0.01 / 100) \times (t_{sat,F} - 100)) \times 4186.8$
R-134a	$-60 < t_{sat} < 50$ $c_{p,g} = 883.30094 + (4.2791217) \times t_{sat} + (0.024521575) \times (t_{sat}^2) + (0.00031098309) \times (t_{sat}^3) + (0.0000029711092) \times (t_{sat}^4)$ $50 < t_{sat} < 90$ $c_{p,g} = 109617.07 + (-6575.7869) \times t_{sat} + (148.11649) \times (t_{sat}^2) + (-1.470221) \times (t_{sat}^3) + (0.005448103) \times (t_{sat}^4)$
FC-72	$c_{p,g} = (0.16 + (0.01 / 100) \times (t_{sat,F} - 100)) \times 4186.8$
FC-87	$c_{p,g} = ((198.06152 + (-0.5254585) \times t_{sat,F}) (-1 / 2.5866964)) \times 4186.8 \times 0.8$

7. Liquid Thermal Conductivity (W/m °C)

Working Fluid	Equation for Liquid Thermal Conductivity Calculation
Water	$k_l = (0.574 + 0.0016 \times t_{sat} - 0.00000565 \times (t_{sat}^2))$
Ethanol	$k_l = (0.172051 \times (10 (0.0000000587444 \times t_{sat} \times t_{sat} - 0.00033585 \times t_{sat})))$
Acetone	$k_l = (0.187401 \times (10 (-0.00000411089 \times t_{sat} \times t_{sat} - 0.000634084 \times t_{sat})))$
R-11	$k_l = -0.00000108 \times t_{sat} \times t_{sat} - 0.000224 \times t_{sat} + 0.109$
R-113	$k_l = -0.000208 \times t_{sat} + 0.0765$
R-134a	$k_l = 0.093417415 + (-0.00046113251) \times t_{sat} + (8.5029375 \times 10^{-09}) \times (t_{sat}^2)$
FC-72	$k_l = (-0.000208 \times t_{sat} + 0.0572)$
FC-87	$k_l = ((0.030635) \times ((1 + 6.667 \times (1 - ((t_{sat} + 273) / 420.6)))) (0.67)))$

8. Liquid Viscosity (kg/m-s)

Working Fluid	Equation for Liquid Viscosity Calculation
Water	$\mu_l = 0.00000236 \times \exp(1779.72 / (t_{sat} + 273))$
Ethanol	$\mu_l = 0.00181202 \times (10^{(0.0000122385 \times t_{sat} \times t_{sat} - 0.00871499 \times t_{sat})})$
Acetone	$\mu_l = 0.001 \times 0.44776 \times (10^{(-0.00514934 \times t_{sat})})$
R-11	$\mu_l = 0.001 \times (0.0000348 \times t_{sat} \times t_{sat} - 0.00727 \times t_{sat} + 0.599)$
R-113	$\mu_l = (0.0000503 \times (t_{sat}^2) - 0.0111 \times t_{sat} + 0.917) \times 0.001$
R-134a	$-60 < t_{sat} < 50$ $\mu_l = (0.00028740952) + (-0.0000035771245) \times t_{sat} + (0.00000029226428) \times (t_{sat}^2) + (-2.5537351 \times 10^{-10}) \times (t_{sat}^3) + (2.1564371 \times 10^{-12}) \times (t_{sat}^4) + (-2.4822237 \times 10^{-14}) \times (t_{sat}^5) + (1.2231847 \times 10^{-16}) \times (t_{sat}^6)$ $50 < t_{sat} < 90$ $\mu_l = (-0.0033906266) + (0.0003176791) \times t_{sat} + (-0.000011570556) \times (t_{sat}^2) + (0.00000022140318) \times (t_{sat}^3) + (-2.3630954 \times 10^{-9}) \times (t_{sat}^4) + (1.3351078 \times 10^{-11}) \times (t_{sat}^5) + (-3.1223123 \times 10^{-14}) \times (t_{sat}^6)$
FC-72	$\mu_l = 2.14693 \times (10^{10}) \times (t_{sat} + 273)^{(-5.4581)}$
FC-87	$\mu_l = (288 \times \exp(-23.2 + (835.68 / (t_{sat} + 273))))$

9. Gas Viscosity (kg/m-s)

Working Fluid	Equation for Gas Viscosity Calculation
Water	$\mu_g = \exp[4.3995 + (0.0038789) \times (t_{sat}) + (0.000021181) \times (t_{sat}^2) + (-0.00000034406) \times (t_{sat}^3) + (0.000000001673) \times (t_{sat}^4) + (-0.000000000002803) \times (t_{sat}^5)]$
Ethanol	$\mu_g = \exp[4.3995 + (0.0038789) \times (t_{sat}) + (0.000021181) \times (t_{sat}^2) + (-0.00000034406) \times (t_{sat}^3) + (0.000000001673) \times (t_{sat}^4) + (-0.000000000002803) \times (t_{sat}^5)] \times (0.0000001)$
Acetone	$\mu_g = \exp[4.3519 + (0.0026016) \times (t_{sat}) + (-0.0000035279) \times (t_{sat}^2) + (0.00000021547) \times (t_{sat}^3) + (-0.0000000035172) \times (t_{sat}^4) + (0.000000000013856) \times (t_{sat}^5)] \times (0.0000001)$
R-11	$\mu_g = 0.00056749666 + (-0.00000065442562) \times t_{sat} + (0.0000000320162) \times (t_{sat}^2) + (4.4713977 \times 10^{-11}) \times (t_{sat}^3) + (-1.7942462 \times 10^{-12}) \times (t_{sat}^4) + (1.1027002 \times 10^{-14}) \times (t_{sat}^5) + (-2.3885531 \times 10^{-17}) \times (t_{sat}^6)$
R-113	$\mu_g = 0.00082084369 + (-0.00000068223768) \times t_{sat} + (0.000000017527906) \times (t_{sat}^2)$
R-134a	$-60 < t_{sat} < 50$
	$\mu_g = (0.000010942103) + (0.0000000046102163) \times t_{sat} + (1.0659228 \times 10^{-10}) \times (t_{sat}^2) + (1.7147172 \times 10^{-12}) \times (t_{sat}^3) + (1.5056628 \times 10^{-14}) \times (t_{sat}^4)$
	$50 < t_{sat} < 90$
	$\mu_g = 0.00023003143 + (-0.000013143174) \times t_{sat} + (0.00000029426733) \times (t_{sat}^2) + (-2.8822262 \times 10^{-9}) \times (t_{sat}^3) + (1.0524118 \times 10^{-11}) \times (t_{sat}^4)$
FC-72	$\mu_g = 0.00082084369 + (-0.00000068223768) \times t_{sat} + (0.000000017527906) \times (t_{sat}^2)$
FC-87	$\mu_g = 0.00056749666 + (-0.00000065442562) \times t_{sat} + (0.0000000320162) \times (t_{sat}^2) + (4.4713977 \times 10^{-11}) \times (t_{sat}^3) + (-1.7942462 \times 10^{-12}) \times (t_{sat}^4) + (1.1027002 \times 10^{-14}) \times (t_{sat}^5) + (-2.3885531 \times 10^{-17}) \times (t_{sat}^6)$

10. Latent Heat of Evaporation (J/kg)

Working Fluid	Latent Heat of Evaporation
Water	$h_{fg} = 2504121 - 2471 \times t_{sat}$
Ethanol	$h_{fg} = (10^{(-0.00059 \times t_{sat} + 6.026)})$
Acetone	$h_{fg} = \exp[6.3491 + (-0.0027168) \times t_{sat} + (0.000029698) \times (t_{sat}^2) + (-0.000000012474) \times (t_{sat}^3) + (-0.0000000040319) \times (t_{sat}^4) + (0.00000000018757) \times (t_{sat}^5)] \times 1000$
R-11	$h_{fg} = -1.13 \times t_{sat} \times t_{sat} - 354 \times t_{sat} + 192000\#$
R-113	$h_{fg} = 10^{(-0.00088 \times t_{sat} + 5.2)}$
R-134a	$-60 < t_{sat} < 50$ $h_{fg} = 398481.95 + (582.69776) \times t_{sat} + (-1.061128) \times (t_{sat}^2) + (-0.010420707) \times (t_{sat}^3) + (-0.000044863408) \times (t_{sat}^4)$ $50 < t_{sat} < 90$ $h_{fg} = 1906959.7 + (-142592.02) \times t_{sat} + (5640.1714) \times (t_{sat}^2) + (-118.11127) \times (t_{sat}^3) + (1.3854479) \times (t_{sat}^4) + (-0.0086343197) \times (t_{sat}^5) + (0.000022316319) \times (t_{sat}^6)$
FC-72	$h_{fg} = (99199.189) + (-241.77192) \times t_{sat} + (0.29338372) \times (t_{sat}^2) + (-0.018372357) \times (t_{sat}^3) + (0.00014531867) \times (t_{sat}^4) + (-0.000000037625727) \times (t_{sat}^5) + (-2.5748707 \times 10^{-09}) \times (t_{sat}^6)$
FC-87	$h_{fg} = ((143.43 \times ((1 - (t_{sat}/420.6)) 0.38) \times 1000))$

11. Liquid Enthalpy (J/kg)

Working Fluid	Liquid Enthalpy
Water	$h_l = [-37.705033 + (1.2125648) \times t_{\text{sat-F}} + (-0.0011478494) \times (t_{\text{sat-F}}^2) + (0.000001626793) \times (t_{\text{sat-F}}^3)] \times 2326$
Ethanol	$h_l = [-106.54778 + 1.0293815 \times t_{\text{sat-F}}] \times 2326$
Acetone	$h_l = [-472.44136 + 1.5310343 \times t_{\text{sat-F}}] \times 1000$
R-11	$h_l = (200805.29) + (820.20271) \times t_{\text{sat}} + (0.26072527) \times (t_{\text{sat}}^2) + (0.0049743383) \times (t_{\text{sat}}^3)$
R-113	$h_l = (200241.84) + (919.61024) \times t_{\text{sat}} + (-0.002423251) \times (t_{\text{sat}}^2) + (0.0035283309) \times (t_{\text{sat}}^3)$
R-134a	$h_l = 200030.95 + (1330.468) \times t_{\text{sat}} + (1.4184789) \times (t_{\text{sat}}^2) + (0.0097064167) \times (t_{\text{sat}}^3) + (0.000043560917) \times (t_{\text{sat}}^4)$
FC-72	$h_l = ((200241.84) + (919.61024) \times t_{\text{sat}} + (-0.002423251) \times (t_{\text{sat}}^2) + (0.0035283309) \times (t_{\text{sat}}^3)) \times 1.2$
FC-87	$h_l = [(200805.29) + (820.20271) \times t_{\text{sat}} + (0.26072527) \times (t_{\text{sat}}^2) + (0.0049743383) \times (t_{\text{sat}}^3)] / 1.27168$

12. Gas Enthalpy (J/kg)

$$h_{fg} = h_g - h_f$$

13. Air Properties :

Thermal Properties	Equations of Thermal Property Calculation
Density (kg/m ³)	$\rho = 1 / (0.00455 \times 0.622 \times t)$
Specific Heat (kcal/kg-°C)	$90 < t \leq 260$ $c_p = 1.032 - 0.0001225 \times t$ $260 < t \leq 610$ $c_p = 1.04466 - 0.000315967 \times t + 0.000000707909 \times t^2 - 0.00000000027034 \times t^3$ $610 < t \leq 900$ $c_p = 1.00205 - 0.000162983 \times t + 0.000000569525 \times t^2 - 0.000000000268081 \times t^3$ $c_p = c_p \times 0.239006$
Thermal Conductivity (kcal/sec-m-°C)	$80 < t \leq 300$ $a = 385.859; b = 91144; c = -2686670; d = 55260400$ $300 < t \leq 600$ $a = 328.652; b = 167320; c = -30295300; d = 3058620000$ $600 < t \leq 1000$ $a = 539.544; b = -332903; c = 359756000; d = -96720200000$ $k = (t^{0.5} / (a + d / t + c / t^2 + d / t^3)) \times 0.000239006 \times 3600 \times 1.163$
Viscosity (kg/m-sec)	$\mu = (t^{0.5} / (0.671692 + 85.22974 / t - 211.475 / t^2 + 106417 / t^3)) / 10^6$

14. Thermal Conductivity of Selected Metals (W/m °C)

Material	Equations of Thermal Property Calculation
Aluminum	$k_m = 10^{(0.000125 \cdot t + 2.3044)}$
Brass	$k_m = 10^{(0.000705 \cdot t_{sat} + 2.023)}$
Copper	$k_m = 10^{(-0.000069 \cdot t_{sat} + 2.58594)}$
Stainless Steel (kg/m-sec)	$k_m = 0.69228 \times (t_{sat} + 273)^{0.53346}$

APPENDIX N

Simulation Windows

Simulation program consists of many different interface windows such as main window, input window, output window etc. Here, main and design windows are presented.

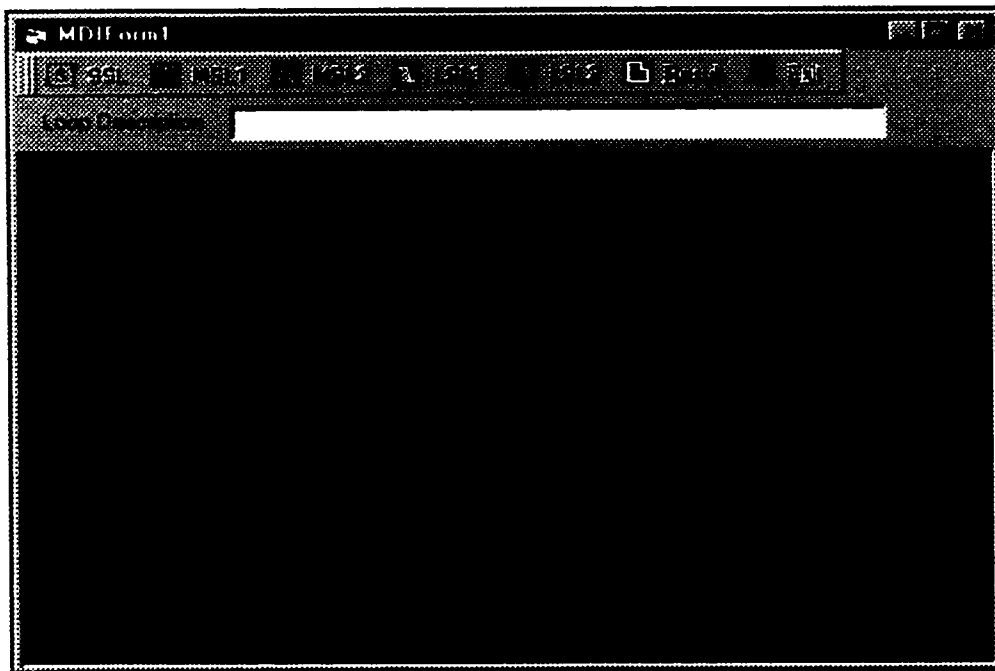


Figure N.1 Main Window

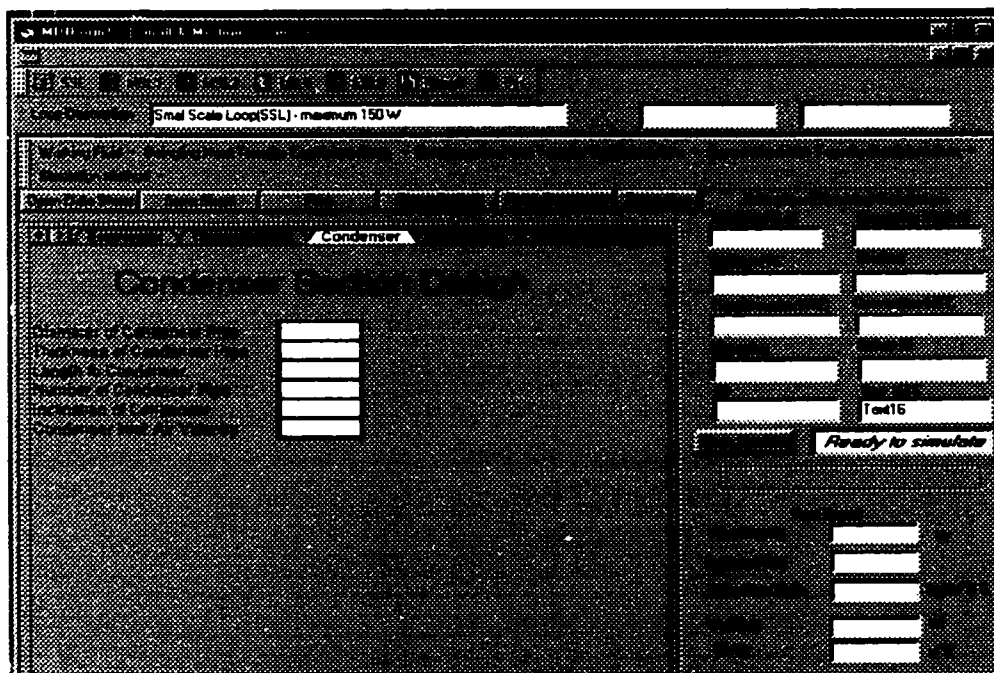


Figure N.2. Design Window

APPENDIX O

ERROR ANALYSIS

The experimental errors were calculated using the method recommended by Kline and McClintok [1953]. The evaluated uncertainties in the variables are:

L (length): $\pm 0.25 \%$

T (temperautr): $\pm 0.2 \text{ }^{\circ}\text{C}$

E (voltage): $\pm 0.5 \%$

I (current): $\pm 3.5 \%$

Using these variables, the following was calculated.

1. Power

$$Q = VI = 79$$

$$\frac{\partial Q}{\partial I} = V = 100$$

$$\frac{\partial Q}{\partial V} = I = 0.79$$

$$w_Q = \left[\left(\frac{\partial Q}{\partial V} w_v \right)^2 + \left(\frac{\partial Q}{\partial I} w_I \right)^2 \right]^{1/2}$$

$$w_Q = \left[(0.79 \times 0.2)^2 + (100 \times 0.01)^2 \right]^{1/2} = 1.012$$

$$Q = 79 \pm 1.012 \text{ W (1.13 \%)}$$

2. Area

$$A = L_e \times L_e = 0.001225 \text{ m}^2$$

$$\frac{\partial A}{\partial L_e} = 2L_e = 0.07$$

$$w_Q = \left[\left(\frac{\partial A}{\partial L_e} w_{L_e} \right)^2 \right]^{1/2}$$

$$w_A = \left[(0.07 \times 0.00000875)^2 \right]^{1/2} = 0.000000612$$

$$A = 0.001225 \pm 0.000000612 \text{ m}^2 \text{ (0.05 \%)}$$

3. Heat Flux

$$q = \frac{Q}{A} = \frac{79}{0.001225} = 64489 \text{ W / m}^2$$

$$\frac{\partial q}{\partial Q} = \frac{1}{A} = \frac{1}{0.001225} = 816.3$$

$$\frac{\partial q}{\partial A} = -\frac{Q}{A^2} = -\frac{79}{0.0000015} = -52666667$$

$$w_q = \left[\left(\frac{\partial q}{\partial A} w_A \right)^2 + \left(\frac{\partial q}{\partial Q} w_Q \right)^2 \right]^{1/2}$$

$$w_q = \left[(-5266667 \times 0.000000612)^2 + (816.3 \times 1.012)^2 \right]^{1/2} = 826.7$$

$$q = 64489 \pm 826.7 \text{ W/m}^2 (1.3 \%)$$

4. Overall heat transfer coefficient

$$U_T = \frac{q}{\Delta t_{h-c}} = \frac{64489}{43} = 1499.7$$

$$\frac{\partial U_T}{\partial \Delta t_{h-c}} = - \frac{1}{(\Delta t_{h-c})^2} = -0.00054$$

$$\frac{\partial U_T}{\partial q} = \frac{1}{(\Delta t_{h-c})} = 0.0233$$

$$w_{U_T} = \left[\left(\frac{\partial U_T}{\partial \Delta t_{h-c}} w_{\Delta t_{h-c}} \right)^2 + \left(\frac{\partial U_T}{\partial q} w_q \right)^2 \right]^{1/2}$$

$$w_q = \left[(-0.00054 \times 0.2)^2 + (0.0233 \times 836.7)^2 \right]^{1/2} = 19.3$$

$$U_T = 1499.7 \pm 19.3 \text{ W/m}^2 \text{ } ^\circ\text{C} (1.2 \%)$$

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Table 1.1 The Candidate's Role and Contribution

Two-Phase Loops	Abbreviate	Heat Transfer Surface Area (m ²)	Max. Heat Transfer Rate (W)	Application	Participation by author (See the notes below for A to D)				Institution when the experiment was carried
					A %	B %	C %	D %	
Small Scale Loop	SSL	0.001225	150	Electronics Cooling	40	70	80	100	U.O
Medium Scale Loop	MSL I	1.53	60	Heat extraction from enclosed space	10	0	80	100	S.E.I
	MSL II	0.019	1500	Basic loop	0	0	80	100	
Large Scale Loop	LSL I	0.095	7500	Waste heat recovery system	0	0	80	100	KIER
	LSL II	3.31	100000	Waste heat recovery system	10	40	80	100	KIER

A : Manufacturing

B : Experimental Measurement

C : Data Analysis

D : Simulation

U.O : University of Ottawa

KIER : Korea Institute of Energy Research, Daeduk, Korea

S.E.I : Samsung Electronics Inc., Suwon, Korea

Table 1.2 Two-phase Flow Parameters [Hewitt, 1978]

Type	Parameters
Primary Design Parameters	Heat transfer coefficient, Mass transfer coefficient, Pressure drop, Critical heat flux
Primary Design Parameters (Fault Condition)	System discharge and related parameters, Bubble growth and collapse, Dry-out under fault conditions, Rewetting
Secondary Design Parameters	Vibration, Transient momentum flux, Flow distribution, Stability, Quality, Mass flow, Liquid level
Second Order Parameters	Flow pattern, Film thickness and amplitude, Mass flow distribution, Phase concentration, Void fraction, Velocity and momentum flux distribution, Bubble and particle size, Temperature distribution, Entrainment, Contact angle
Third Order Parameters	Film thickness and interfacial wave, Fluctuations in velocity, pressure, temperature and wall shear
Scaling	Scaling law

Table 1.3 Error Range of Two-Phase Main Parameters used in Simulation

Two-Phase Parameters				Remarks	Error Range (%)
Flow Pattern	Flow Pattern Map			Determined empirically	Appendix A
	Flow Pattern Transition			Determined empirically	Appendix A
Heat Transfer Coefficients	Single-Phase	Single Phase Convection		Determined empirically	< 10 %
	Two-Phase	Boiling	Lumped	Determined empirically	Appendix B.
			Sectorial	Determined empirically	No Ref
		Condensation		Determined empirically	No Ref
Pressure Drops	Single Phase	Frictional		Determined empirically	Appendix B
		Accelerational			No Ref.
		Gravitational			No Ref.
		Area Change Model		Determined empirically	Appendix B
	Two-Phase	Frictional	Lumped	Determined empirically	Appendix B
			Sectorial	Determined empirically	No Ref.
		Accelerational		Determined empirically	No Ref.
		Gravitational		Determined empirically	No Ref.
		Area Change Model		Determined empirically	No Ref.
Other Two-Phase Parameters	Void Fraction			Determined empirically	Appendix B
	Annular Flow Entrainment			Determined empirically	No Ref.

Table 1.4 Dimensionless Groups for Scaling of Two-Phase Flow Systems

#	Two-phase Scaling [Mayer, 1977]	Het Transfer Scaling [Ahmand, 1974]	Frictional Pressure Drop Scaling [Friedel, 1977]	Two-Phase Loop Scaling [Ishii et al., 1977]
1	Weber Number	Boiling Number	Euler Number	Richardson Number
2	Laplace Constant	Reynolds Number	Dimensionless Geometry	Friction Number
3	Jakob Number	Weber-Reynolds Number	Density Ratio	Stanton Number
4	Bubble Reynolds Number	Liquid/Vapor Viscosity Ratio	Viscosity Ratio	Time Ratio Number
5	Boiling Number	Subcooling Number	Dimensionless Enthalpy	Biot Number
6	Buoyancy Number	Liquid/Vapor Density Ratio	Froude Number	Heat Source Number
7	Subcooling Number	Specific Geometric Similarity	Reynolds Number	Phase Change Number
8	Phase Change Number	Froude-Reynolds Number	Relative Roughness	Subcooling Number
9	Slip Ratio	Liquid Prandtl Number	Weber Number	Froude Number
10	Drift Number	Vapor Prandtl Number		Drift Flux Number
11	Martinelli Number	Liquid/Vapor Thermal Conductivity Ratio		Density Ratio
12		Barnett Number		Two-Phase Friction Number
13		Saturation Number		Orifice Number
14				Dimensionless Geometry
15				

Category	Subcategory	Type	Mechanism	Characteristics
Static	Simple (fundamental)	Ledinegg (flow excursion) instability	$\left(\frac{\partial \Delta p}{\partial G}\right)_m \leq \left(\frac{\partial \Delta p}{\partial G}\right)_m$	Flow undergoes a sudden, large-amplitude excursion, to a new, stable operating condition
		Thermal (boiling) crisis	Substantial decrease of heat transfer coefficient	Wall excursion with possible flow oscillation
		Flow-regime transition instability (relaxation instability)	Bubbly flow has less void but higher Δp than annular flow; condensation rate depends on flow regime	Cyclic flow regime transitions and flow rate variations
		Nonequilibrium-state instability	Transformation wave propagates along the system	Recoverable work disturbances and heating disturbances waves
	Compound	Unstable vapor formation (bumping, geysering, vapor burst) (relaxation instabilities)	Periodic adjustment of metastable condition, usually due to lack of nucleation sites	Occasional or periodic process of liquid superheat and violent vaporization with possible expulsion and refilling
		Condensation chugging	Bubble growth and condensation followed by surge of liquid (in steam discharge pipes)	Periodic interruption of vent steam flow due to condensation and surge of water up to downcomer
Dynamic	Simple (fundamental)	Acoustic oscillations	Resonance of pressure waves	High-frequency pressure oscillations (10–100 Hz) related to time required for pressure wave propagation in system
		Density wave oscillation	Delay and feedback effects in relationships among flow rate, density, pressure drops	Low-frequency oscillations (~1 Hz) related to transit time of a mass-continuity wave
	Compound	Thermal oscillations	Interaction of variable heat transfer coefficient with flow dynamics	Occurs close to film boiling
		Boiling-water reactor instability	Interaction of void/reactivity coupling with flow dynamics and heat transfer	Relevant only for a small fuel time constant and under low pressure
		Parallel channel instability	Interaction among a small number of parallel channels	Various modes of dynamic flow redistribution
	Compound as a secondary phenomenon	Condensation oscillation	Interaction of direct contact condensation interface with pool convection	Occurs with steam injection into vapor suppression pools
		Pressure drop oscillation	A flow excursion initiates a dynamic interaction between a channel and a compressible volume	Very-low-frequency periodic process (~0.1 Hz)

Table 1.5 Two-Phase Instabilities [Bergles, 1981]

Table 3.1 Description of Experimental Loops

Two-Phase Loops	Abbrev.	Maximum Heat Transfer Rate (W)	Heating Method	Cooling Method	Heat Transfer Area (m ²)		Evaporator Structure
					Evaporator Section	Condenser Section	
Small Scale Loop	SSL	150 W	Electrical	Cold Air	0.001225	0.0844	3.5 cm × 3.5 cm Square cavity
Medium Scale Loop	MSL I	60 W	Hot Air Convection	Direct Conduction	1.53 (case 1) 0.89 (case 2)	0.0114	0.00952 m OD Pipe
	MSL II	1500 W	Electrical	Water	0.019	0.0026	0.0202 m ID Pipe
Large Scale Loop	LSL I	7500 W	Electrical	Cold Air	0.095	1.2472	0.0202 m ID Pipe, (0.3 m length), 5 pipe assembly)
	LSL II	100,000 W	Hot Air Convection	Cold Air	3.31	3.01	0.0202 m ID Pipe, (0.795 m length), 5 pipe assembly), 5 Loops

Table 3.1 Description of Experimental Loops (continued)

Two-Phase Loops	Abbrev.	Condenser Structure	Application
Small Scale Loop	SSL	3.5 cm × 3.5 cm Square cavity	Electronics Cooling
Medium Scale Loop	MSL I	Direct conduction dissipater	Heat extraction from enclosed space
	MSL II	Water cooling jacket	Basic loop
Large Scale Loop	LSL I	0.0202 m ID Pipe, (0.3 m length), 5 pipe assembly with fins	Waste heat recovery system
	LSL II	0.0202 m ID Pipe, (0.785 m length), 5 pipe assembly), 5 Loops	Waste heat recovery system

Table 3.2 Characteristics of Condensers, SSL

Condenser Number	Number of tubes	O.D. mm	I.D. mm	$A_{v, cr}$ cm ²	A_{cond} cm ²	$A_{air, cr}$ cm ²	A_{fin} cm ²	A_{conv} cm ²
1	2	6.4	5.6	0.49	70	37.1	840	920
2	3	7.9	7.1	1.2	135	18.8	562	712
3	3	7.9	7.1	1.2	135	19.5	695	844
2 and 3	6	7.9	7.1	2.4	270	-	1257	1556

Table 3.3 Cooling Channels for the Condenser

# of insert	Dimension of condenser	Maximum velocity without confinement	Maximum velocity between fins
Full channel	200 mm × 35 mm	0 ~ 1.65 m/s	0 ~ 4.4 m/s
1st insert	150 mm × 35 mm	0 ~ 1.8 m/s	0 ~ 4.8 m/s
2nd insert	100 mm × 35 mm	0 ~ 2.4 m/s	0 ~ 6.4 m/s
3rd insert	50 mm × 35 mm	0 ~ 2.54 m/s	0 ~ 6.8 m/s

Table 3.4 Some Physical Properties of Working Fluids used

No	Working Fluid	Chemical Formula	T _{sat} at 1 atm., °C	Freezing Temp. at 1 atm., °C	Flammability, % in air	Toxicity (max conc. in air), mg/m ³	Compatible Material	Applied into
1	Water	H ₂ O	100	0	-	-	Stainless steel, Copper, Nickel, Titanium, Silica, Brass	SSL, MSL I, MSL II, LSL I, LSL II
2	Ethanol	C ₂ H ₅ OH	78	-114	3.3-19	1000	Stainless steel, Copper, Nickel, Titanium, Silica, Brass	SSL
3	Acetone	(CH ₃) ₂ CO	56	-93	2.6-13	200	Aluminum, Stainless steel, Copper, Nickel, Titanium, Silica, Brass	SSL
4	R113	C ₂ F ₃ Cl ₃	48	-36	-	3000	Aluminum	SSL
5	R11	CFCl ₃	24	-111	-	3000	Aluminum	SSL
6	FC72	C ₆ F ₁₄	56	-90	-	-	Aluminum	SSL
7	FC87	C ₃ F ₁₂	30	-101	-	-	Aluminum	SSL
8	R134a	CH ₂ FCF ₃	-25.5	-101	-	-	Aluminum	MSL I

Table 3.5 Solubility of Non-Condensable Gas in Working Fluids

Working Fluid	Boiling Point	Solubility, at 1 atm. <i>ml/100 ml</i> liquid
FC-72	56 °C	48 (at 25 °C)
FC-87	30 °C	54 (at 25 °C)
Water	100 °C	2.6 O ₂ (at 30 °C)
Water	100 °C	1.7 H ₂ (at 30 °C)

Table 4.1 Flow Regimes Considered in the Lumped Method

Flow Regimes	Description	References
Single Phase Convection	Basic flow regime	
Incipient Boiling	Transition	Davis and Anderson [1966]
Two-Phase Forced Convection	Basic flow regime	
Liquid Deficient Region	Transition	Rhee and Young [1975]

Table 4.2 h_m used in Simulation

Type	Original Correlation	Correlation
Type A	Martinelli [1949]	Pujol and Stenning [1968]
		Sekoguchi [1992]
Type B	Schrock and Grossman [1959]	Schrock and Grossman [1959]
		Wright [1968]
Type C	Chen [1966]	Modified Chen [1980]
		Kandlikar [1989]
		Shah [1982]
		Gungor and Winterton [1990]
Type D	Modified h_l or h_{lo} , others	Ananiev [1961]
		Crain and Bell [1973]

Table 4.3 Description of Flow Pattern Maps referred in Simulation

Author	Flow Condition	Remarks
Hewitt and Roberts [1969]	Adiabatic vertical air-water and <i>steam-water flow</i>	1. air - water flow, 31.2 mm vertical pipe, 0.14 ~ 0.54 MPa, 2. <i>steam - water flow, 12.7 mm pipe, 3.45 ~ 6.5 MPa</i>
Taitel et al. [1980]	Adiabatic unified model, 25 °C air-water flow	25 ~ 50 mm vertical pipe, 0.1 ~ 1.0 MPa,
Ishii et al. [1984]	Adiabatic Vertical, 25 °C air-water flow	10.3 ~ 50 mm, 0.1 ~ 9.7 MPa

Table 4.4 Flow Regimes and Flow Regime Transitions

Flow Regime	Remarks	Reference
Single Phase Flow	Basic Flow Pattern	
Checking the Existence of Bubbly Flow	Transition between single phase and bubbly flow	Taitel et al. [1980]
Bubbly Flow	Basic Flow Pattern	
Bubbly - Slug Flow	Transition between bubbly and slug flow	Taitel et al. [1980]
Slug Flow	Basic flow pattern	
Slug - Annular	Transition between slug and annular flow	Taitel et al. [1980], Mishima and Ishii [1980]
Annular	Basic flow pattern	
Annular - Drop	Transition between annular and drop flow	$\alpha_{AD} = 0.999$
Drop	Basic flow pattern	

Table 4.5 Constant A_f in Eq. (4.94) or (4.96)

Working Fluid	A_f
Water	0.62
Acetone	0.21
FC-87	0.21

Table 5.1. Test Conditions; Flow Patterns, Fig 5.17 (a) to (h), SSL

WF	V_{WF} <i>ml</i>	u_{max} <i>m/s</i>	Q <i>W</i>	t_v $^{\circ}C$
Acetone	40	5	20; 60; 100	32; 41; 45
Ethanol	40	5	20; 60; 100	37; 39; 47
Ethanol	40	0	20; 60; 100	41; 63; 73
FC-87	40	5	20; 60; 100	34; 44; 52
R-11	40	5	20; 60; 100	34; 43; 49
R-113	40	5	20; 60; 100	30; 36; 46
Water	40	5	20; 60; 100	34; 39; 41
Water Insert 0.5 mm	40	5	20; 60; 100	41; 50; 60

Table 5.2 Test Conditions; One Condenser , SSL

WF	Coordinates	Conditions	V_{WF} ml	t_c °C	l_c mm	u_{max} m/s	Δt_{h-c} °C	Q W
ACETONE	$Q-\Delta t_{h-c}$	one condenser	30	24-25	200	5	10-55	10-110
	$Q-\Delta t_{h-c}$	open TS, counterflow condenser	30	"	"	5	10-55	10-110
	$Q-\Delta t_{h-c}$	one condenser	30	"	"	4.5	40-50	80-115
	$Q-\Delta t_{h-c}$	natural conv.	30	"	"	0	15-60	10-80
	$Q-\Delta t_{h-c}$	insert 0.5 mm	10; 20; 25; 30; 40	"	"	5	10-50	10-120
	$Q-\Delta t_{h-c}$	inserts: 0.5; 0.7; 1.2 mm	30	"	"	5	40-50	80-120
	$\Delta t_{h-c}-d_{ins}$	inserts: 0.5; 0.7; 1.2 mm	30	"	"	5	45-50	100
	$\Delta t_{h-c}-u_{max}$	coflow and counterflow condensers	30	"	"	0 - 5.5	75-45	100
	$\Delta t_{h-c}-u_{max}$	two condensers, $l_c=400$ mm	45	"	"	0 - 5.5	60-40	100
	$t-l_{loop}$	natural conv.	30	"	"	0	-	20; 30; 40; 50; 60; 70

Table 5.2 Test Conditions; One Condenser (continued), SSL

WF	Coordinates	Conditions	V_{WF} ml	t_c , °C	l_c mm	u_{max} m/s	Δt_{h-c} °C	Q W
WATER	$Q-\Delta t_{h-c}$	one condenser	15; 30	23- 24. 5	200	5	13-53	10- 105
	$Q-\Delta t_{h-c}$	one condenser	30	"	"	2.4	20-53	10- 105
	$Q-\Delta t_{h-c}$	natural conv.	30	"	"	0	20-60	10-63
	$Q-\Delta t_{h-c}$	inserts: 0.5; 0.7; 1.2 mm	30	"	"	5	15-50	10- 110
	$\Delta t_{h-c}-u_{max}$		30	"	"	2-5	52-49	100
	$H_{fluid}-Q$	insert 0.5 mm	40	"	"	5.5	-	10- 120
	$\Delta t_{h-c}-V_{WF}$	one condenser	30; 40; 50; 60	"	"	5	49-53	100
	$\Delta t_{h-c}-V_{o\ air}$	insert 0.5 mm	40	"	"	5.5	45-70	100
R11	$Q-\Delta t_{h-c}$	one condenser	35	24	200	5.5	17-53	10- 120
	$H_{fluid}-Q$	one condenser	35	"	"	5.5	-	10- 120
	$k_{fil}-t_{fil}$	old filler	35- 40	"	"	5.5	-	-
	$k_{fil}-t_{fil}$	new filler	35- 40	"	"	5.5	-	-

Table 5.2 Test Conditions; One Condenser (continued), SSL

WF	Coordinates	Conditions	V_{WF} ml	t_c °C	l_c mm	u_{max} m/s	Δt_{h-c} °C	Q W
FC-87	$Q-\Delta t_{h-c}$	one condenser	30; 40	24	200	5	13-50	10-100
	$\Delta t_{h-c}-u_{max}$	one condenser	30	"	"	2.5-5	33-32	30
	$H_{fluid}-Q$	one condenser	40	"	"	5.5	-	20-120
	$k_{fil}-t_{fil}$	new/old filler	35-40	"	"	5.5	-	-
FC-72	$Q-\Delta t_{h-c}$	one condenser	40	22.5-25	200	5	20-60	10-100
			30	"	"	6.4	36-52	50-90
			30	"	"	5, 6.4	35-80	50-90
R-113	$Q-\Delta t_{h-c}$	one condenser	30	23-24	200	5	15-55	10-110
	$Q-\Delta t_{h-c}$	natural conv.	30	"	"	0-0.4	25-60	10-60
ETHANOL	$Q-\Delta t_{h-c}$	one condenser	30	24	200	5	15-50	10-100

Table 5.3. Test Matrix, One Condensers , SSL

W.F Coordinates	Acetone	Ethanol	FC-72	FC-87	R-11	R-113	Water
$Q-\Delta t_{h-c}$	○	○	○	○	○	○	○
$\Delta t_{h-c}-u_{max}$	○			○			○
$\Delta t_{h-c}-l_c$			○				
$\Delta t_{h-c}-d_{insert}$	○						○
$H_{fluid}-Q$				○	○		○
$t-l_{loop}$	○	○	○	○	○	○	○
$k_{fil}-t_{fil}$			○	○			
$Q-\Delta t_{h-c}$ nat. conv	○		○			○	○
$\Delta t_{h-c}-V_{WF}$	○ (insert 0.5mm)						○
$\Delta t_{h-c}-V_{o\ air}$ insert 0.5 mm							○
$Q-\Delta t_{h-c}$ two condensers	○						
$Q-\Delta t_{h-c}$ two condensers, natural convection	○						

Table 5.4. Test Conditions; Two Condensers , SSL

Loop Type	WF	V_{WF}	t_c	l_c	u_{max}	Δt_{h-c}	Q
		ml	°C,	mm	m/s	°C	W
two condensers with evaporator C	R-11	50	20.5	200	5	11-54.7	10-120
		60	20.5	200	5	15-54.1	50-150
	Ethanol	50	20.5	200	5	15-58	10-130
		50	20.5	200	0	15-62	10-80
	R-113	50	20.5	200	5	15-56	10-120
		50	20.5	200	0	20-55	10-70
	FC-72	30	20.5	200	5	13.5-54.4	10-120
		30	20.5	200	0	20-67	10-70
	FC-87	50	20.5	200	5	11-56	10-150
		50	20.5	200	5	16-57.3	10-90
	Water	50	20.5	200	5	35-48	50-105
		50	20.5	200	0	43-50	50-70
	Acetone	40	20.5	200	5	14-56.8	10-150
		40	20.5	200	0	56-63	150
		40	20.5	200	0	17-64	10-100
		50	20.5	200	5	18-56	20-150
		50	20.5	200	0	44-51	100
		60	20.5	200	5	11.7-59	10-150

Table 5.5. Test Matrix; Two Condensers , SSL

WF Coordinates	Acetone	Ethanol	FC-72	FC-87	R-11	R-113	Water
$Q-\Delta t_{h-c}$	○	○	○	○	○	○	○
$\Delta t_{h-c}-u_{max}$	○						
$H_{fluid}-Q$	○				○		
$t-l_{loop}$	○	○	○	○	○	○	○
$Q-\Delta t_{h-c}$ nat. conv	○	○	○	○		○	○
$\Delta t_{h-c}-V_{WF}$	○				○		

Table 5.6. Test Conditions for Flow Instability Investigation (two condensers), SSL

WF	V_{WF} ml	Loop Type	d_{insert}	Q, W	$t_c, ^\circ C$	u_{max}	$f(u_{max})$	$u_{max}=0$
Water	50	two condensers with evaporator C	no insert	20, 30, 40, 50, 60, 80, 100, 120	19.5- 23.5	5	○	○
			1.2	30, 60, 80, 100, 120	25	5		○
			0.7	30, 60, 80, 100, 120	20-25	5	○	○

Table 5.6. Test Conditions for Flow Instability Investigation (two condensers), SSL
(Continued)

WF	V_{WF} ml	Loop Type	d_{insert}	Q, W	$t_c, ^\circ C$	u_{max}	$f(u_{max})$	$u_{max}=0$
Acetone	50	two condensers with evaporator C	no insert	20, 30, 40, 50, 60, 80, 100, 120	18.5- 23.5	5	○	○
			0.7	30, 60, 80, 100, 120	24.5	5		
FC-87	50	two condensers with evaporator C	no insert	20, 30, 40, 50, 60, 80, 100, 120	20	5	○	○

Table 5.7. Test Conditions for Flow Instability Investigation (one condenser), SSL

W.F	V_{WF} ml	Loop Type	d_{insert}	Q, W	$t_c, ^\circ C$	u_{max}	$f(u_{max})$	$f(l_c)$	$u_{max}=0$
Water	30	one condenser with evaporator C	no insert	30, 60, 80, 100, 120, 150	25	5	○		○
	40	one condenser with evaporator C	no insert	30, 60, 80, 100, 120	25	5	○	○	○

Table 5.7. Test Conditions for Flow Instability Investigation (one condenser), SSL
(Continued)

W.F	V_{WF} ml	Loop Type	d_{insert}	Q , W	t_c , °C	u_{max}	$f(u_{max})$	$f(l_c)$	$u_{max}=0$
Water	40	one condenser with evaporator C	1.2	30, 50, 60, 80, 100, 120	25	5			
			0.7	30, 50, 60, 80, 100, 120	25	5	○	○	○
			0.5	30, 50, 60, 80, 100, 120	25	5			
	50	one condenser with evaporator C	no insert	30, 50, 60, 80, 100, 120					
Acetone	40	one condenser with evaporator C	no insert	30, 50, 60, 80, 100	25	5	○		○

Table 5.8. Test Matrix; MSL I and MSL II

WF Coordinates	MSL I		MSL II
	Case I	Case II	
$Q-\Delta t_{h-c}$	○	○	○
$H_{fluid}-Q$			○
$Q-\Delta L_{e-c}$			○
$Q-P_{sat}$			○
$Q-t_c$	○	○	○
$\Delta t_{h-c}-V_{WFR}$	○	○	○

Table 5.9. Test Conditions; MSL I and MSL II

Loop Type		WF	ΔL_{e-c}	V_{WFR}	t_c	u_h	u_c	Δt_{h-c}	Q
			m		°C,	Nm/s	Nm/s	°C	W
MSL I	Case I	R-134a		100g, 130g	-21.6, -18.3			3	50 ~ 60
	Case II			100g, 130g	-2.4, -0.7			5	50 ~ 60
MSL II		Water	0.5, 1.0, 1.5	30, 50, 70	50, 60, 70			20 ~ 100	500, 700, 1000, 1200, 1500

Table 5.10. Test Matrix; LSL I and LSL II

WF Coordinates	LSL I	LSL II
$Q-\Delta t_{h-c}$	○	○
$H_{fluid}-Q$	○	
$Q-\Delta L_{e-c}$	○	
Q-Pressure drop		○
$Q-P_{int}$	○	○
$Q-u_h$		○
$Q-u_c$		○
$Q-t_c$	○	○
$\Delta t_{h-c}-V_{WFR}$	○	

Table 5.11. Test Conditions; LSL I and LSL II

Loop Type		WF	ΔL_{e-c}	V_{WFR}	t_c	u_h	u_c	Δt_{h-c}	Q
			m	%	°C,	Nm/s	Nm/s	°C	W
LSL I		Water	1	50	10, 20, 30,		234	50- 175	2500, 5000, 7500,
LSL II	Parallel Flow	Water	1	75	10 - 45	0.63- 1.62	1.61- 3.65	151- 265	20000- 120000
	Counter Flow	Water	1	75	10 - 45	0.63- 1.62	1.61- 3.65	151- 265	20000- 120000

Table 5.12 Figures for Comparisons between Experimental and Simulation

Type		Q vs. Δt_{h-c} (Original)	Temp. Distribution	Q vs. Δt_{h-c} (Modified)	Δt_{h-c} vs. V_{WF}	Δt_{h-c} vs. u_{max} (u_c) Q vs u_c (LSL II)	Δt_{h-c} vs. I_c	Instability	Effectiveness vs. Δt_{h-c}
SSL		5.32 (a)	5.32 (b) ~ 5.32 (h)	5.32 (i) ~ 5.32 (k)	5.32 (l)	5.32(m)	5.32 (n)	5.32 (o) ~ 5.32 (q)	
MSL I	Case I	5.33 (a)	5.33 (b)	5.33 (d)					
	Case II		5.33 (c)	5.33 (d)					
MSL II		5.34 (a)	5.34 (b)	5.34 (c)	5.34 (d)				
LSL I		5.35 (a)	5.35 (b)	5.35 (c)	5.35 (d)				
	Parallel	5.36 (a)	5.36 (b)	5.36 (c)	5.36 (d)				5.36 (e)
	Counter	5.36 (f)	5.36 (g)	5.36 (h)	5.36 (i)				5.36 (j)

Table 5.13 Correction Factor of Heat Transfer Coefficients in Simulation

$$h_x = C_x x^m$$

x = evaporation or condensation (annular for sectorial method) or cond, $m = 1$

TLT	Type	W.F	C_x	C_{cond}
SSL	Lumped	Correlation	Kandlikar	Nusselt
		Water	1.3	0.4
		Acetone	1.2	0.8
		R-11	1.6	1
		R-113	1.7	1.2
		FC-72	1	1.5
		FC-87	2.3	1.9
		Ethanol	1.9	0.8
	Sectorial	Correlation	Rhee and Young	Nusselt
		Water	4	0.3
		Acetone	1.2	0.95
		R-11	1.8	0.9
		R-113	1.7	1.3
		FC-72	0.9	1.6
		FC-87	2.3	2.2
		Ethanol	2.45	0.9
MSL I	Lump	Water	Chen	Smirnov and Lukanov
			2.2	0.75
	Sectorial	Water	Rhee and Young	Smirnov and Lukanov
			1	0.3

Table 5.13 Correction Factor of Heat Transfer Coefficients in Simulation (continued)

TLT	Type	W.F	C_{α}	C_{cond}
MSL II	Lumped	Water	Kandlikar	Nusselt
			0.5	0.45
	Sectorial	Water	Rhee and Young	Nusselt
			0.24	0.3
LSL I	Lumped	Water	Kandlikar	Nusselt
			2.5	0.6
	Sectorial	Water	Rhee and Young	Nusselt
			2.5	0.4
LSL II	Lumped	Water	Kandlikar	Nusselt
			0.47	0.38
	Sectorial	Water	Rhee and Young	Nusselt
			1.4	2



Plate 3.1 Experimental Setup (SSL)

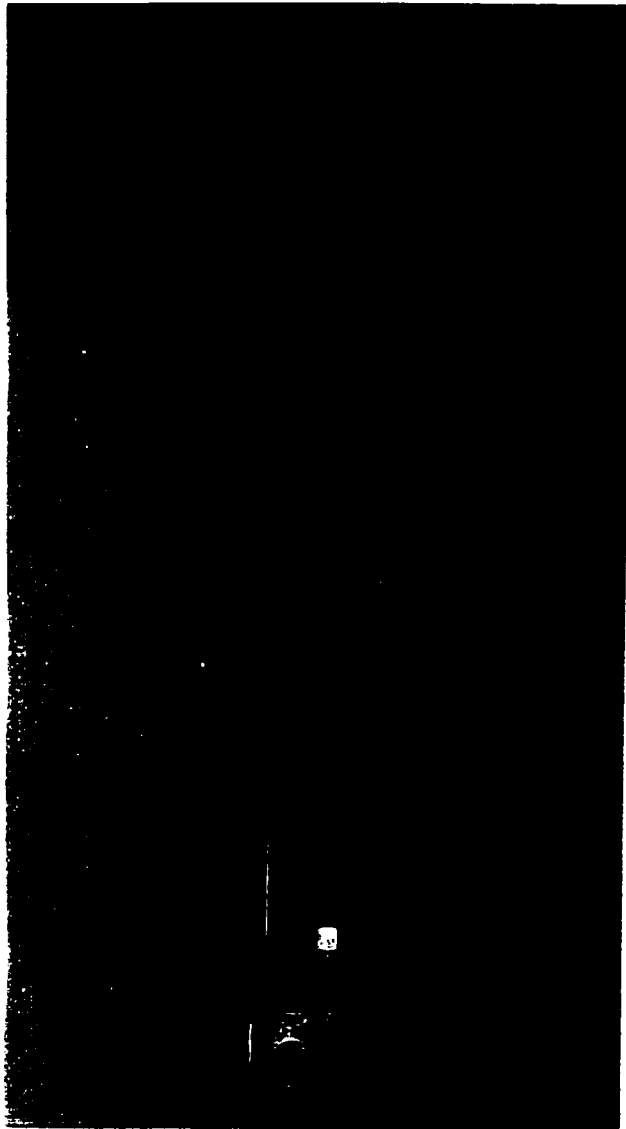


Plate 3.2 Evaporator Section "A" (SSL)

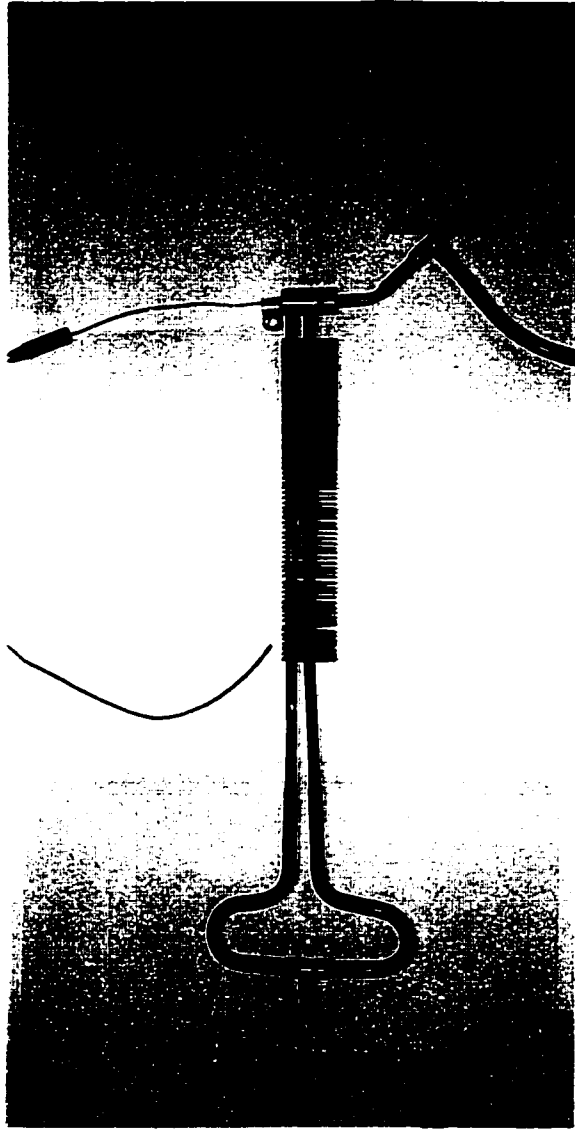


Plate 3.3 Condenser #1 (SSL)



Plate 3.4 Condenser #2 (SSL)



Plate 3.5 Transport Section (SSL)



Plate 3.6 Transport Section (Continued - SSL)



Plate 3.7 Inserts (SSL)

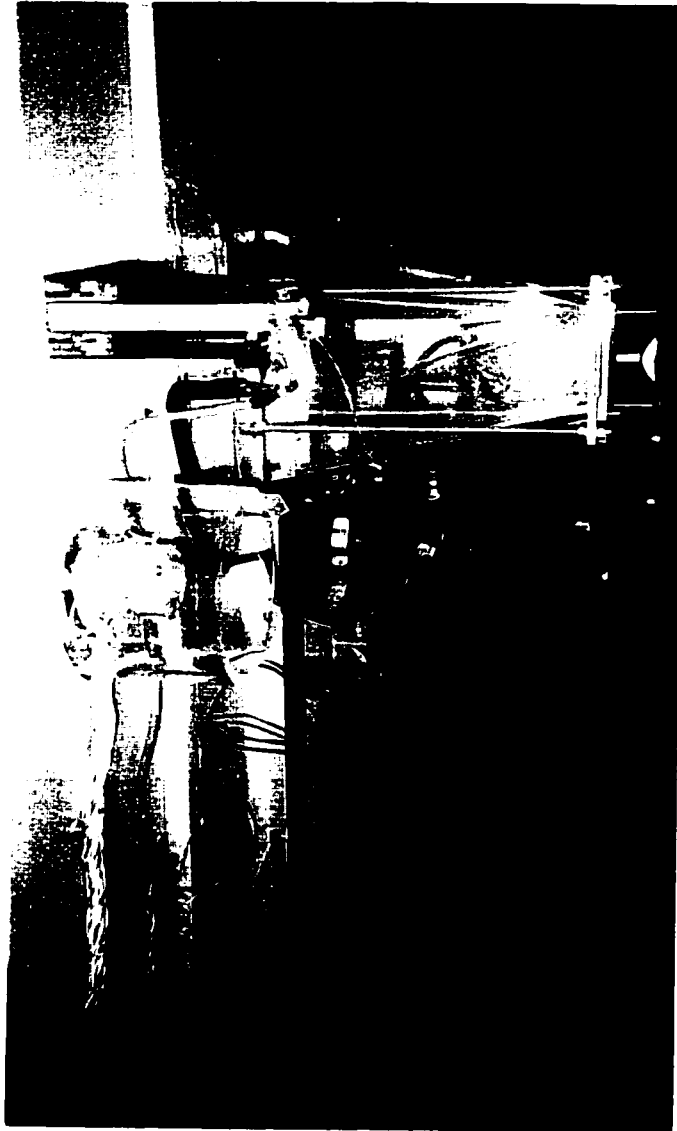
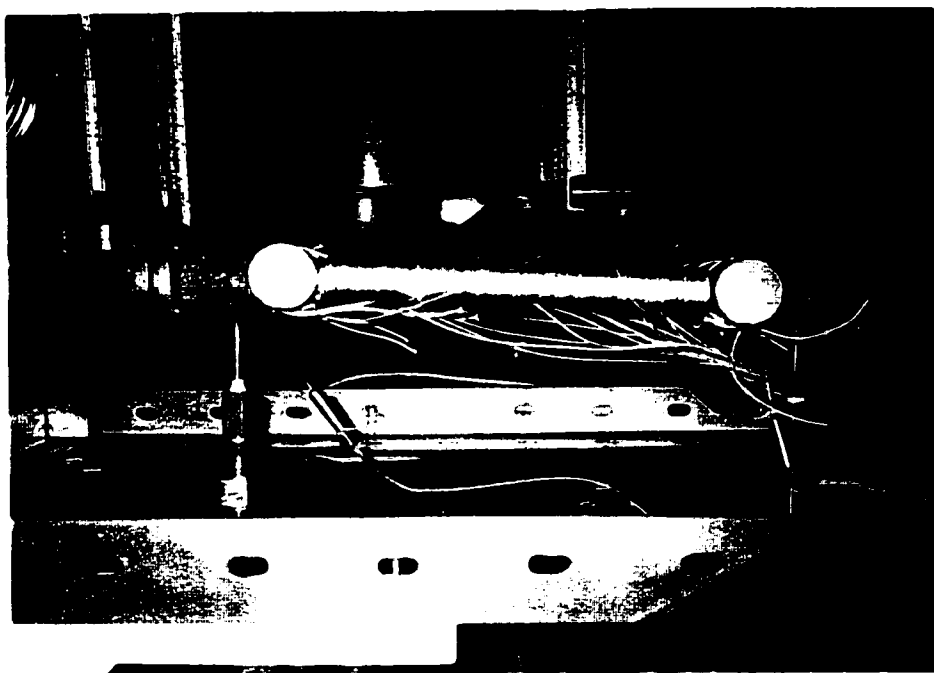


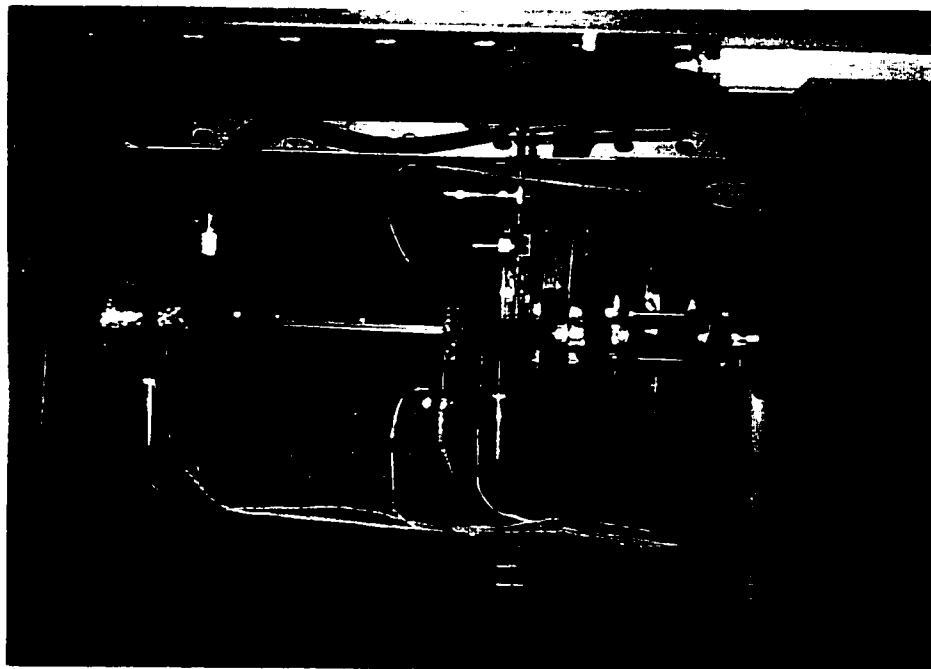
Plate 3.8 Cooling Fan Assembly (SSL)



Plate 3.9 Experimental Setup (MSL II)



(a) Evaporator Section

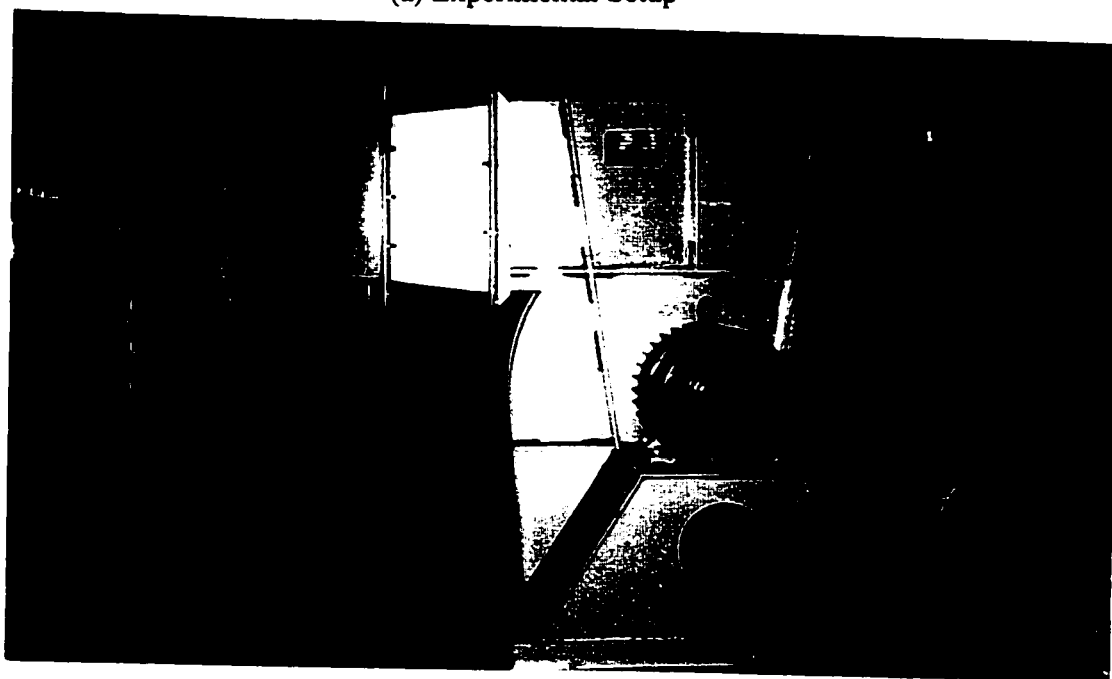


(b) Condenser Section

Plate 3.10 Evaporator and Condenser Sections (MSL II)

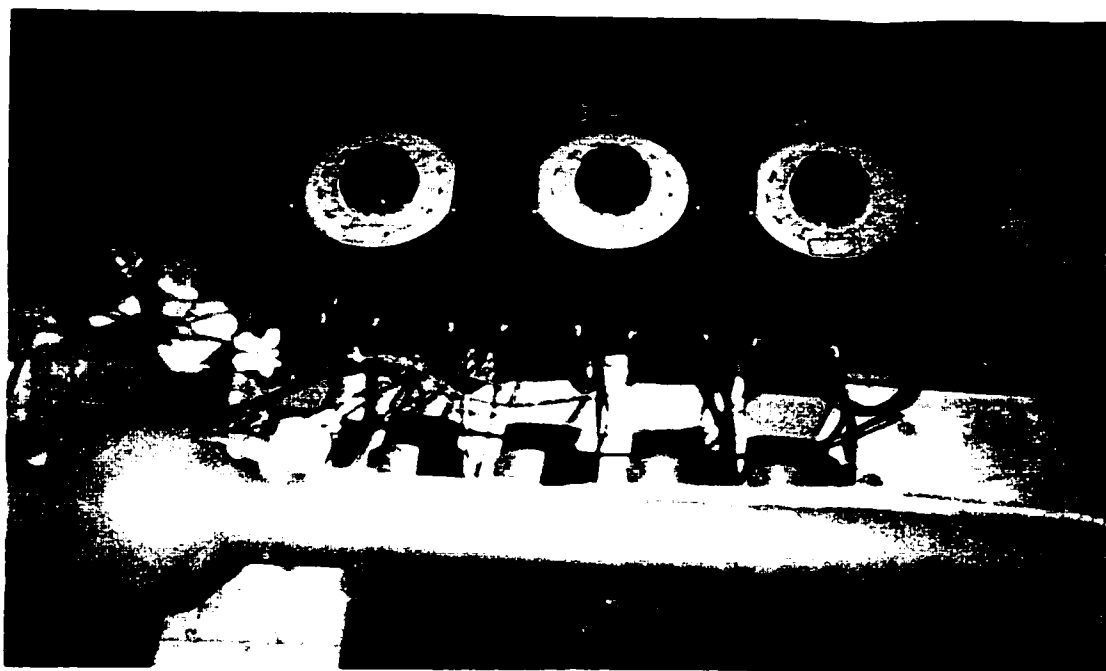


(a) Experimental Setup



(b) Burner

Plate 3.11 Experimental Setup and Burner (LSL I)

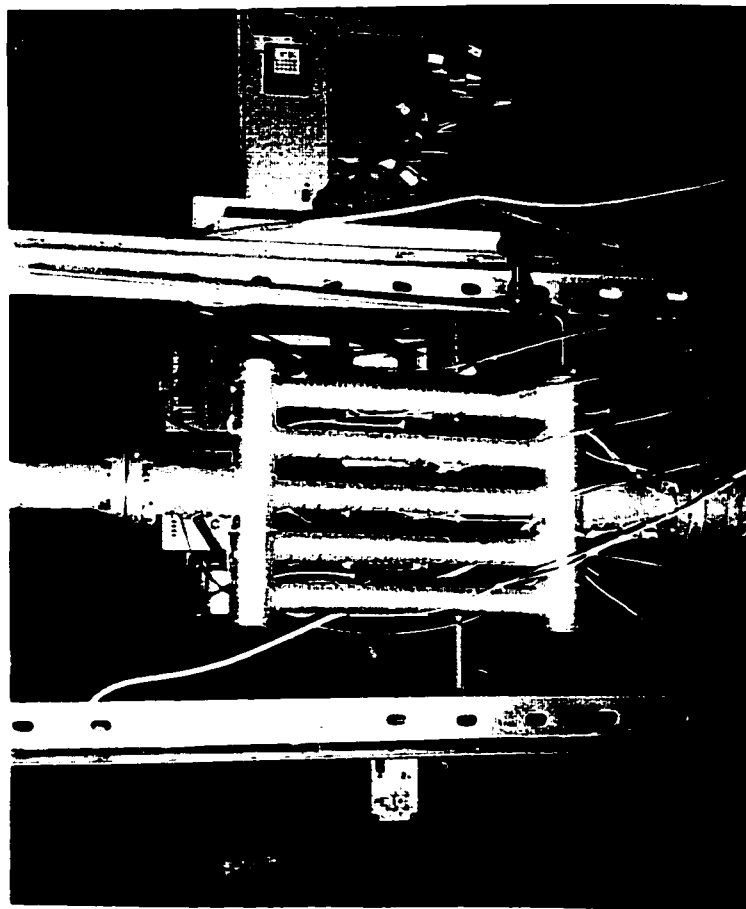


(a) Power Supply

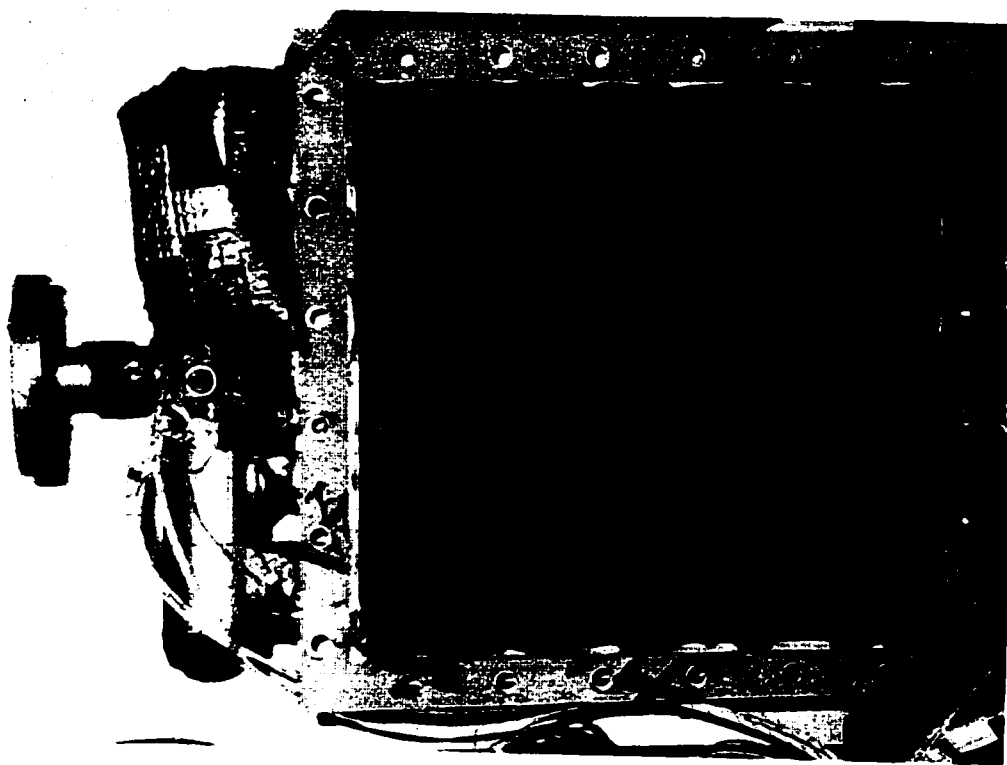


(b) Data Acquisition System

Plate 3.12 Power Supply and Data Acquisition System (LSL I)



(a) Evaporator Section

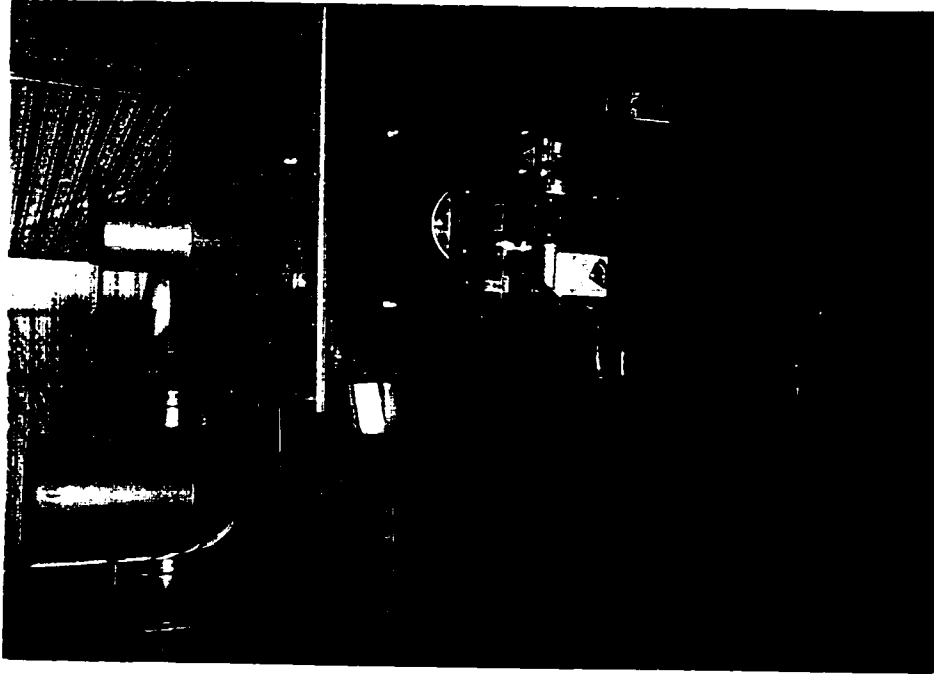


(b) Condenser Section

Plate 3.13 Evaporator and Condenser Section (LSL-1)

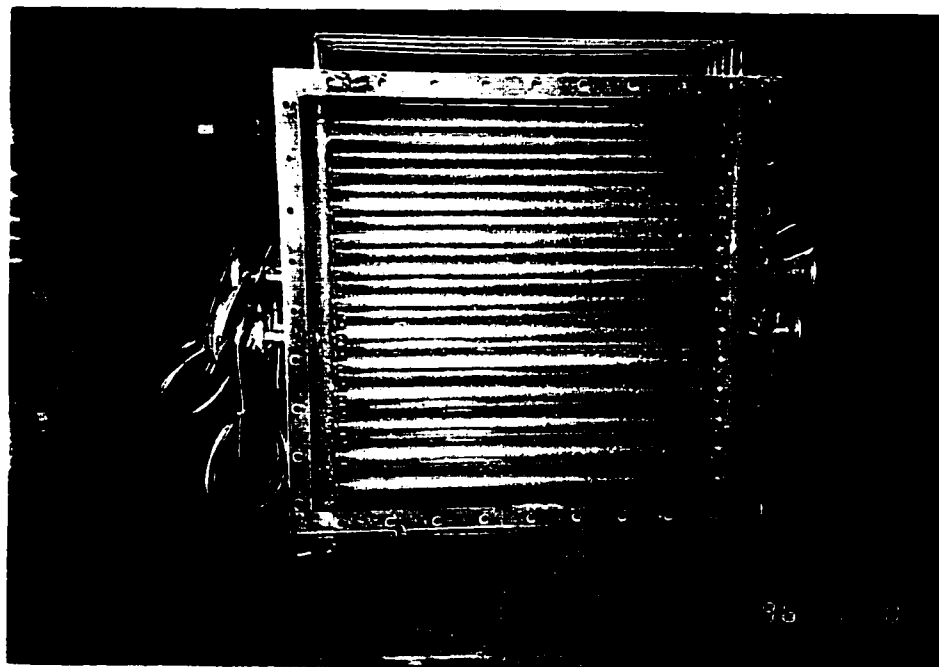


(a) Experimental Setup



(b) Burner

Plate 3.14 Experimental Setup and Burner (LSL II)



(a) Evaporator before Assembling



(b) Assembled Evaporator Section

Plate 3.15 Evaporator Section (LSL II)



(a) Condenser before Assembling



(b) Assembled Condenser Section

Plate 3.16 Condenser Section (LSL II)

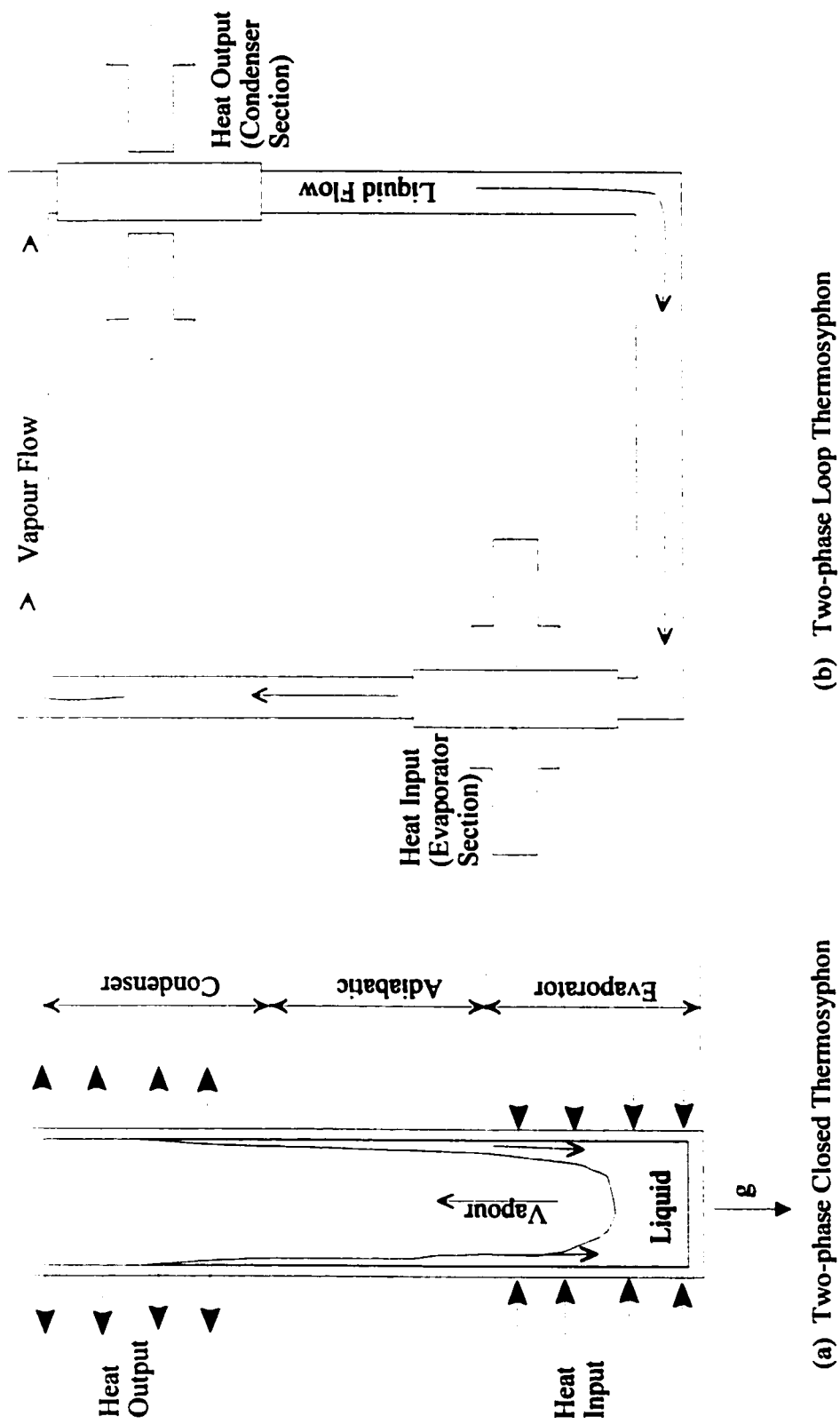


Figure 1.1 Two-Phase Closed and Loop Thermosyphons

Lumped Method

Sectorial Method

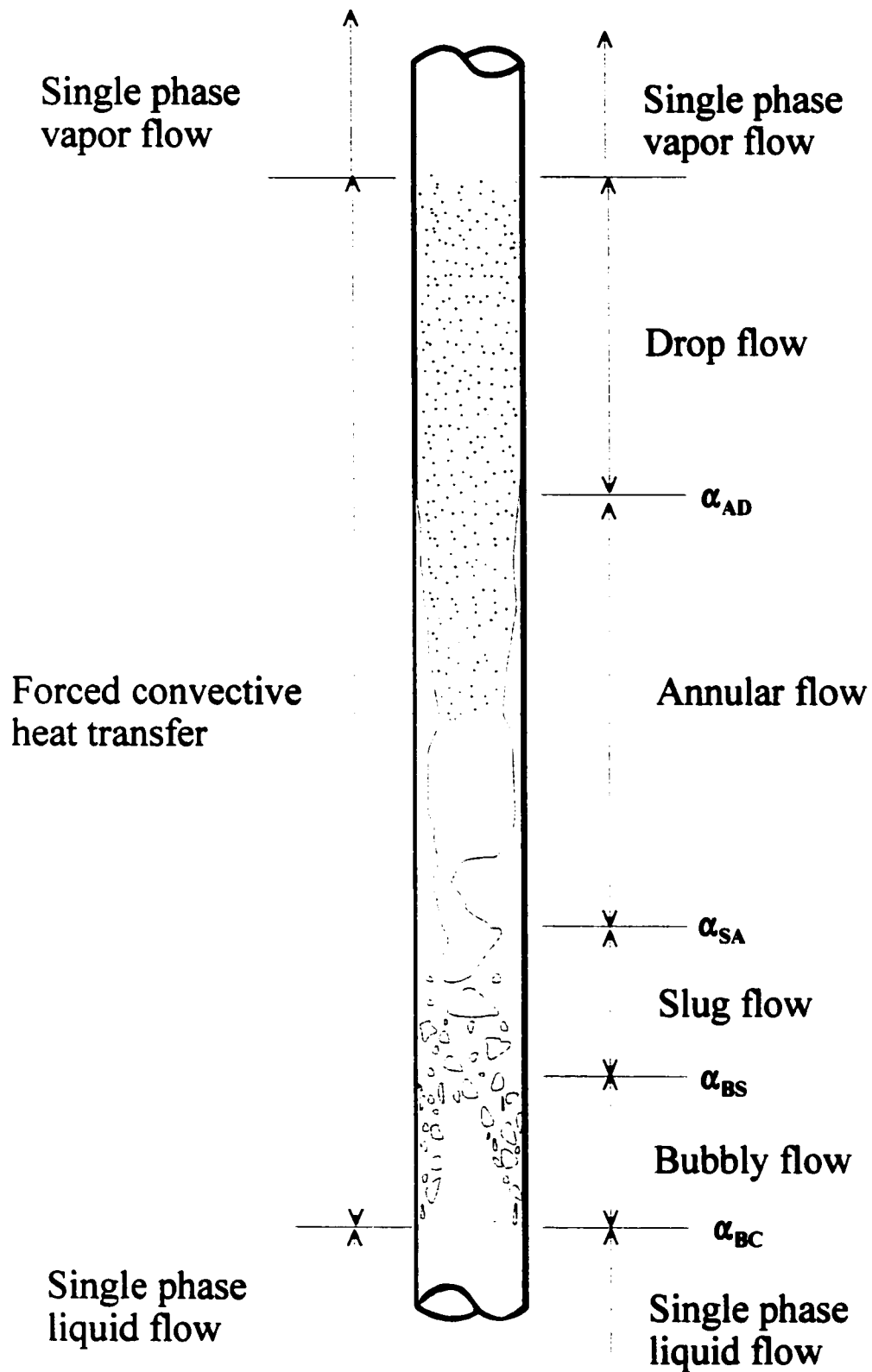


Figure 1.2 Simulation Models in Evaporator Section

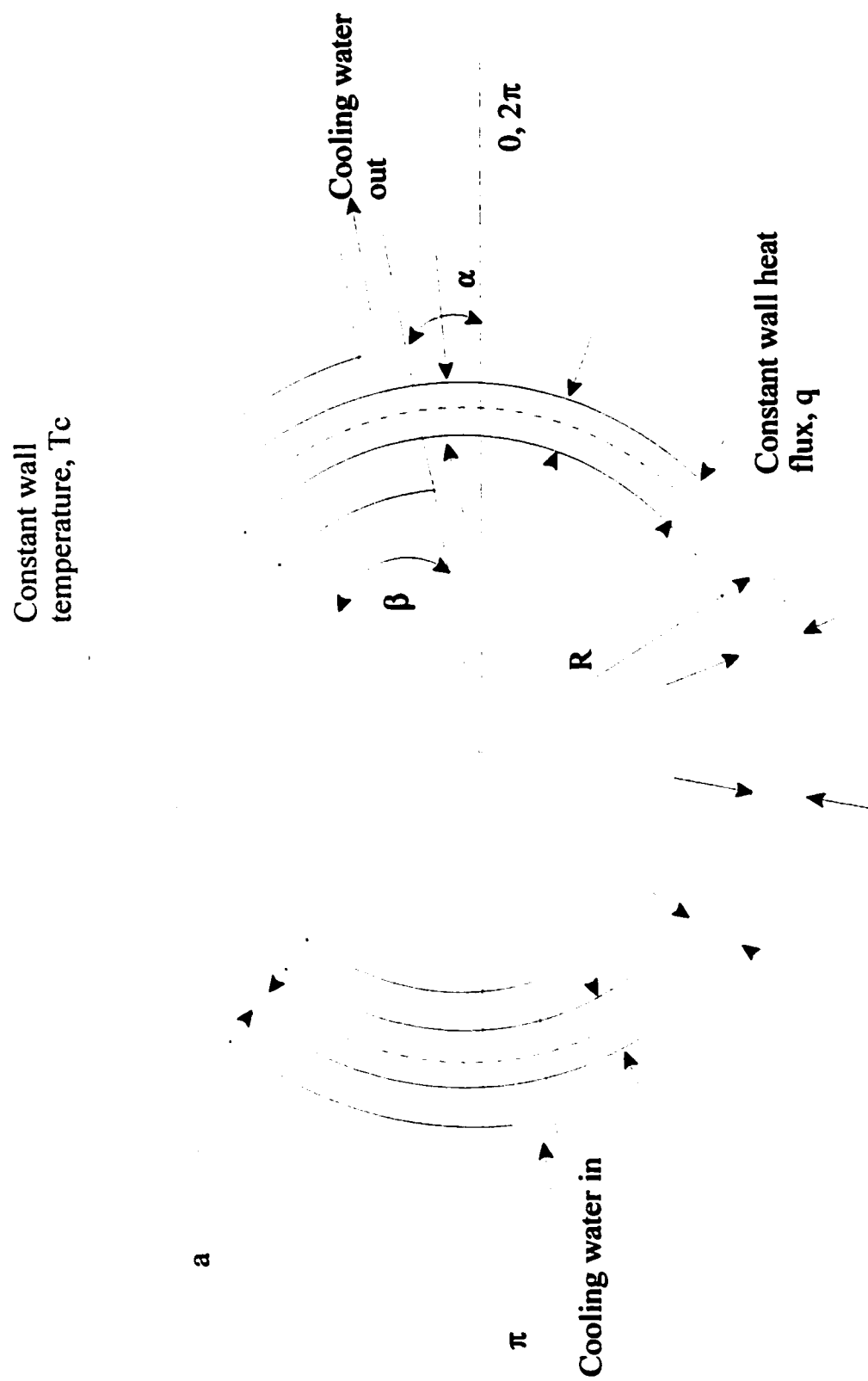
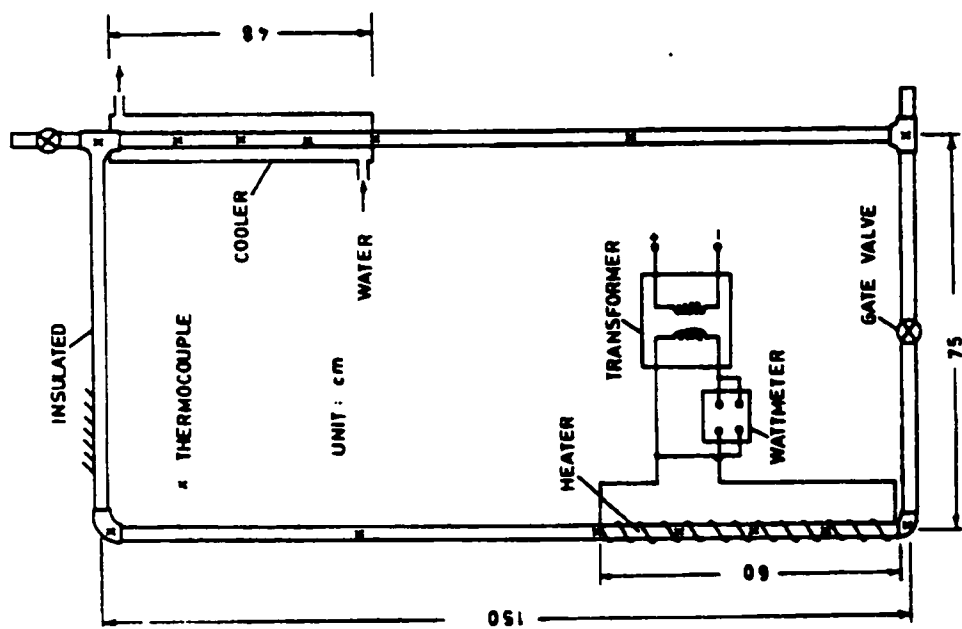
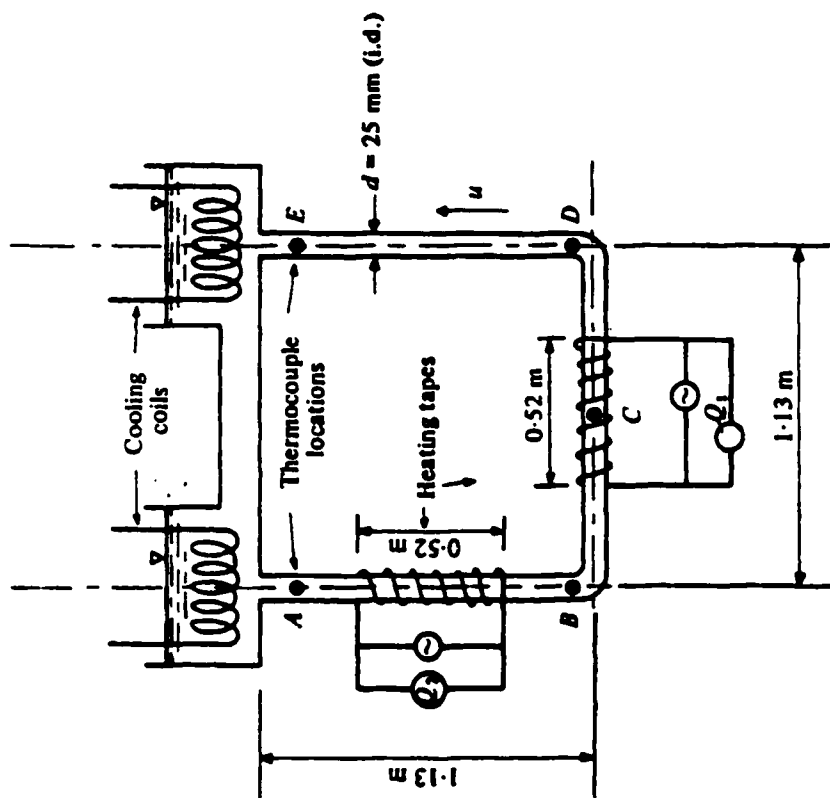


Figure 2.1 Toroidal Thermosyphon [Lavine, 1987]

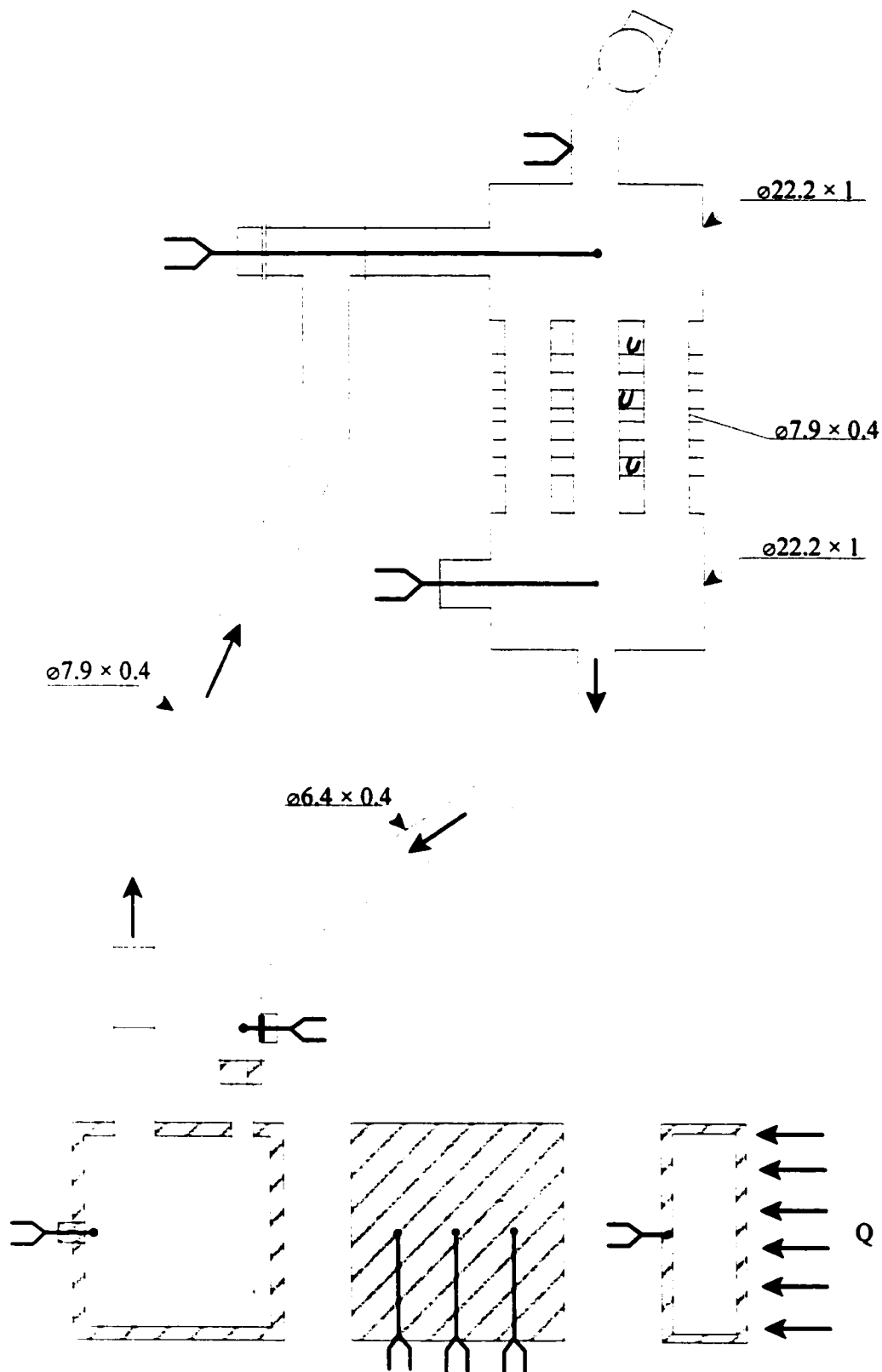


(a) Closed Loop Thermosyphon, Huang [1988]



(b) Open Loop Thermosyphon, Bau [1981b]

Figure 2.2 Closed and Open Loop Single Phase Thermosyphons



Units are in millimeters

Figure 3.1. Loop Thermosyphon Assembly (SSL)

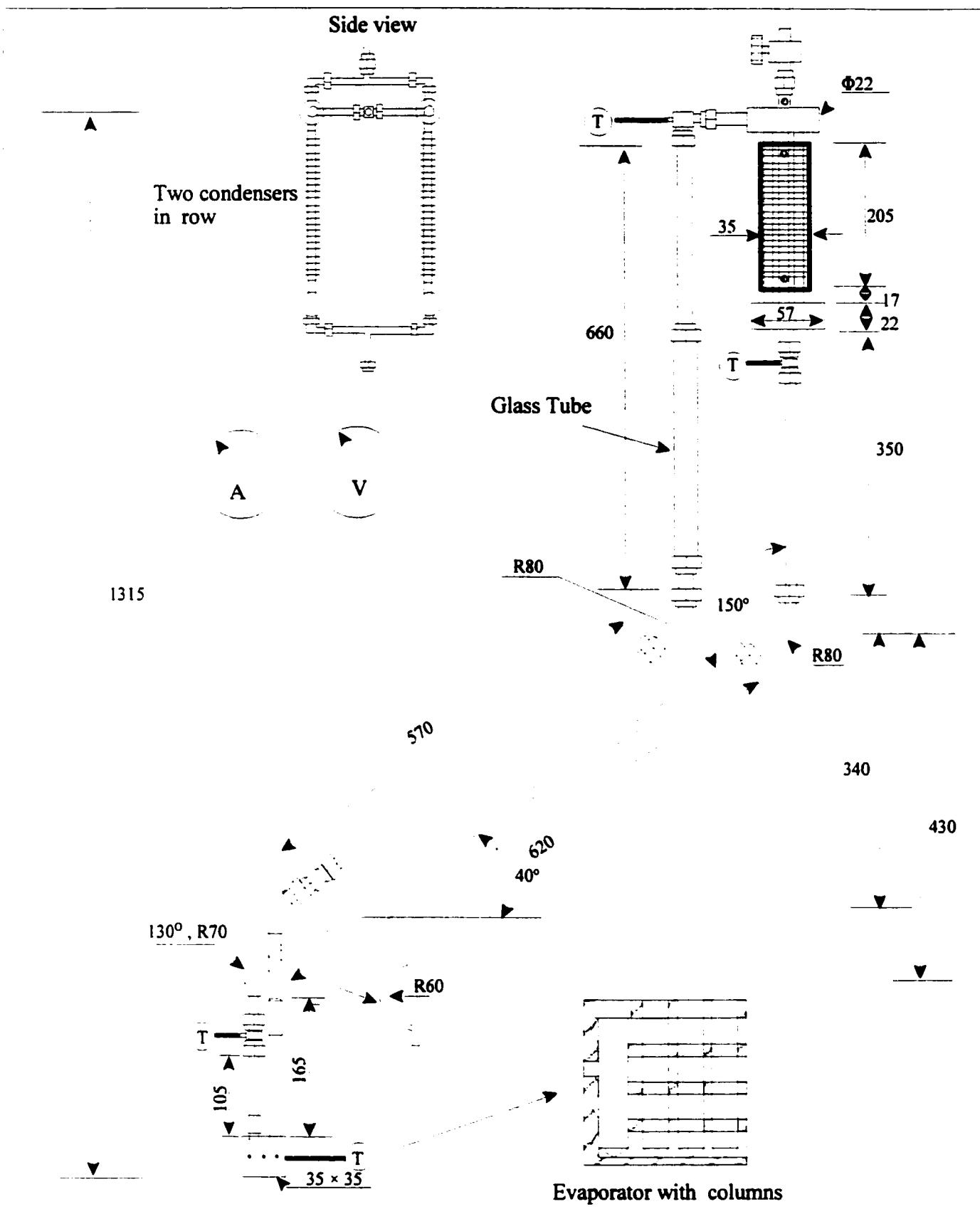


Figure 3.2. Experimental Setup (SSL)

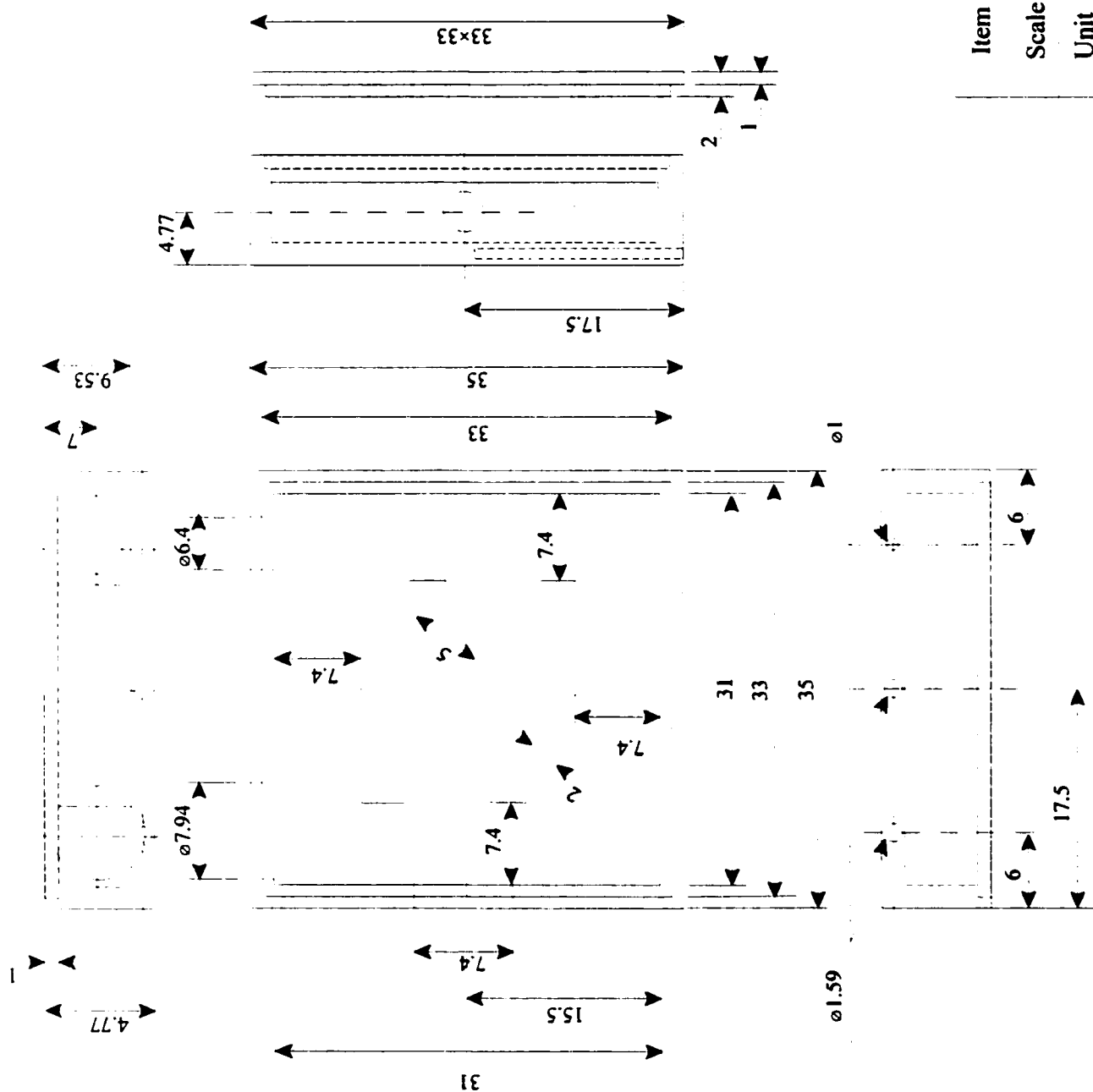


Figure 3.4. Evaporator Section, Design B (SSL)

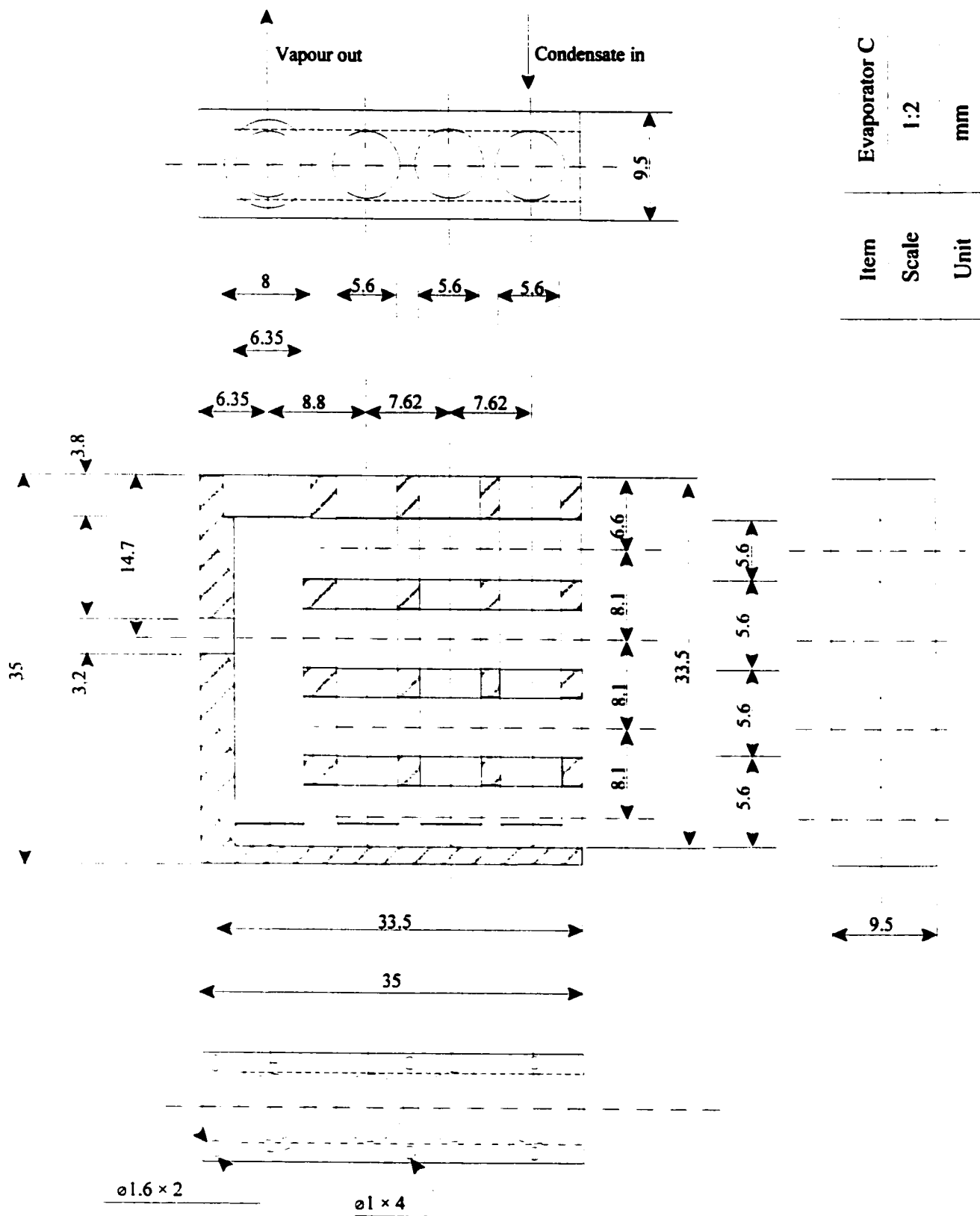


Figure 3.5. Evaporator Section, Design C (SSL)

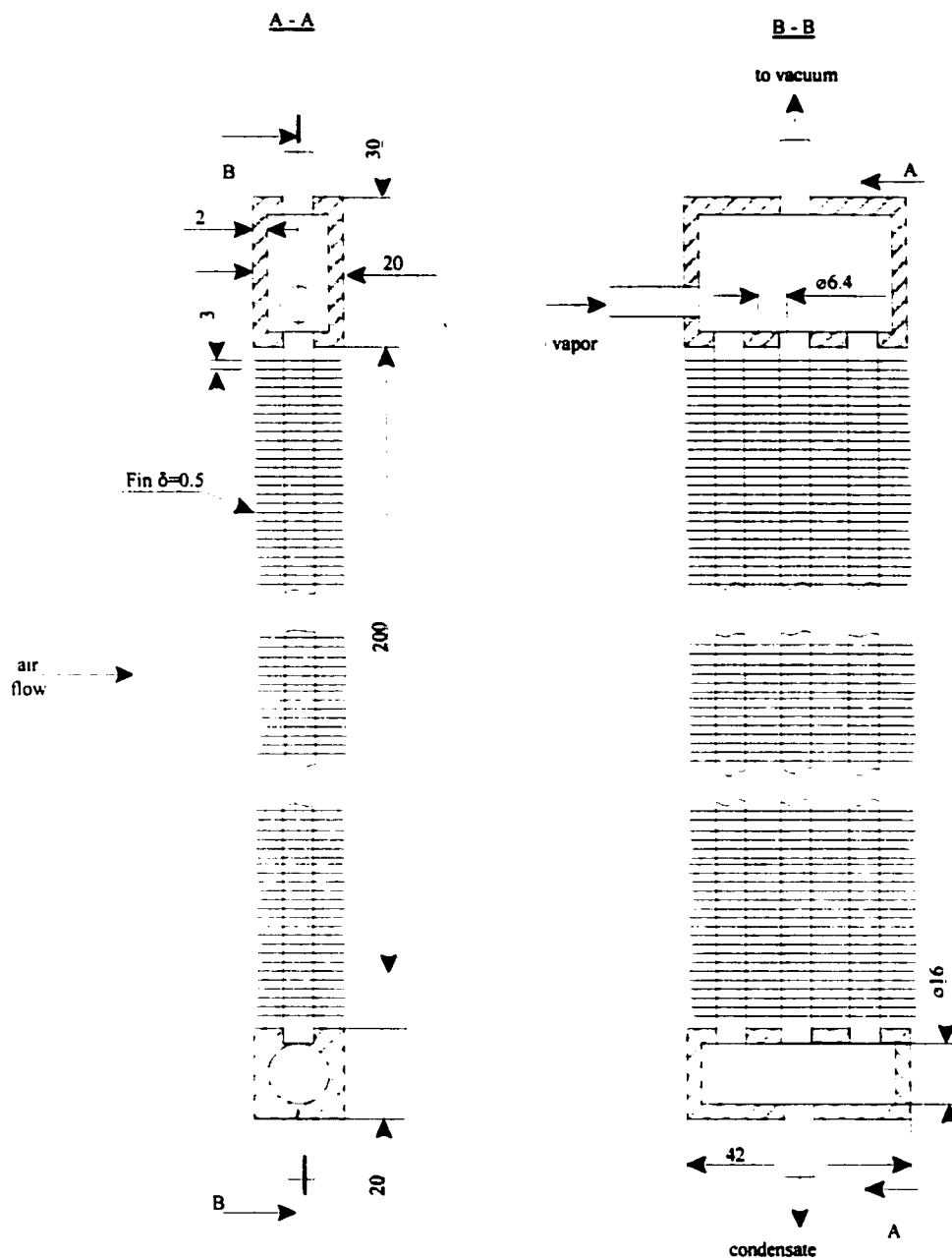


Figure 3.6. Condenser Section (SSL)

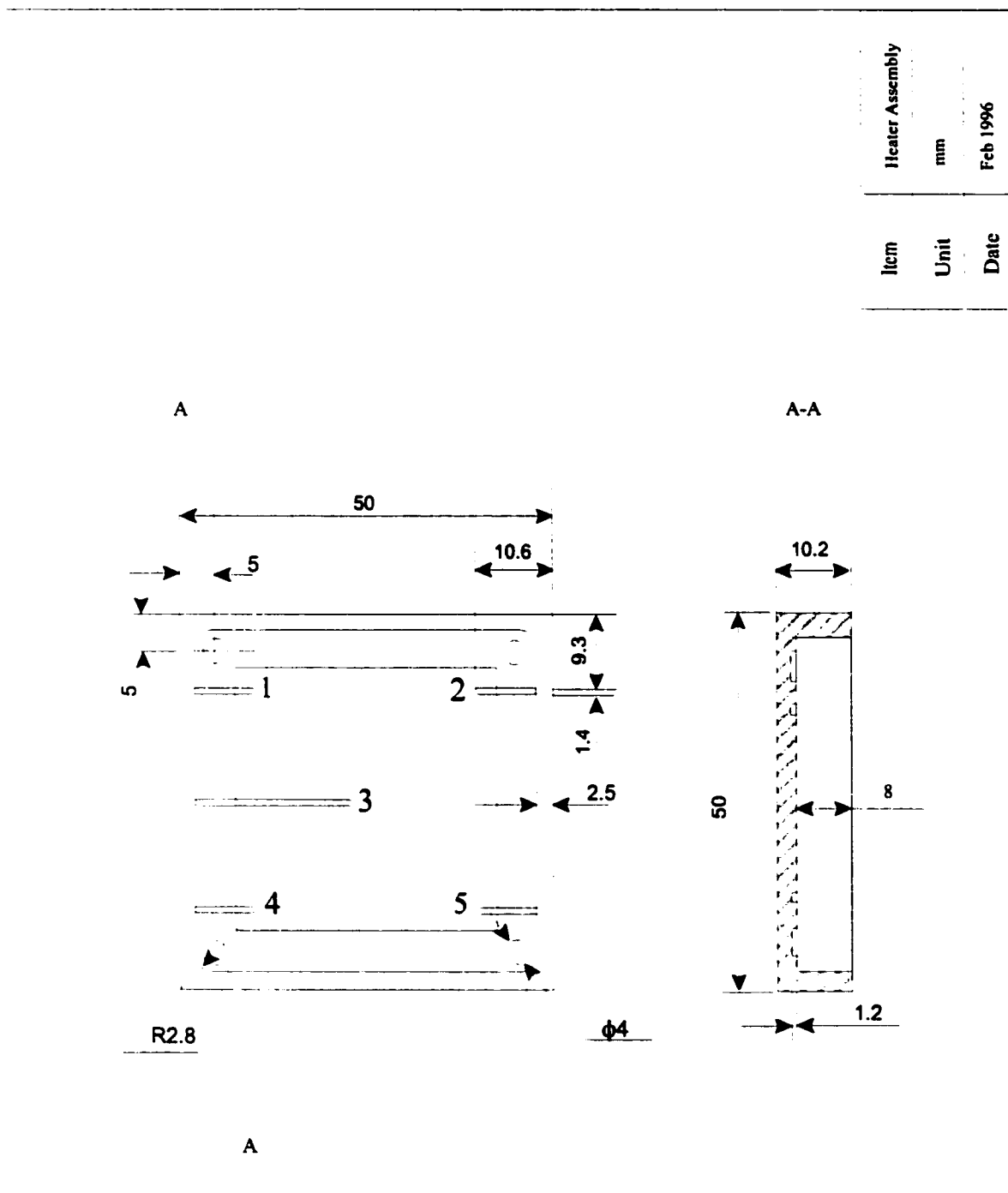


Figure 3.7. Heater Assembly (SSL)

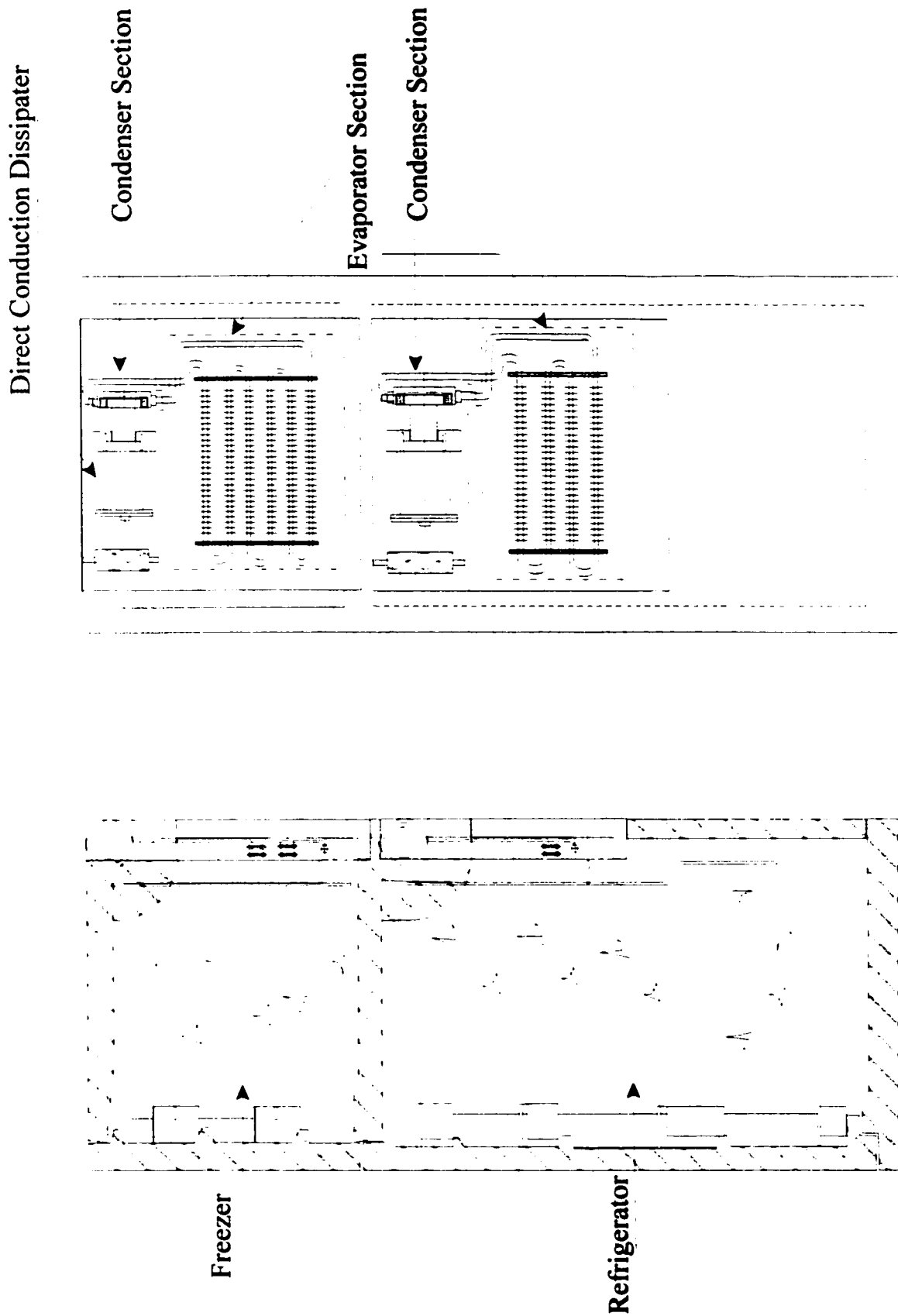


Figure 3.8 Test Assembly of Medium Scale Loop I (MSL I)

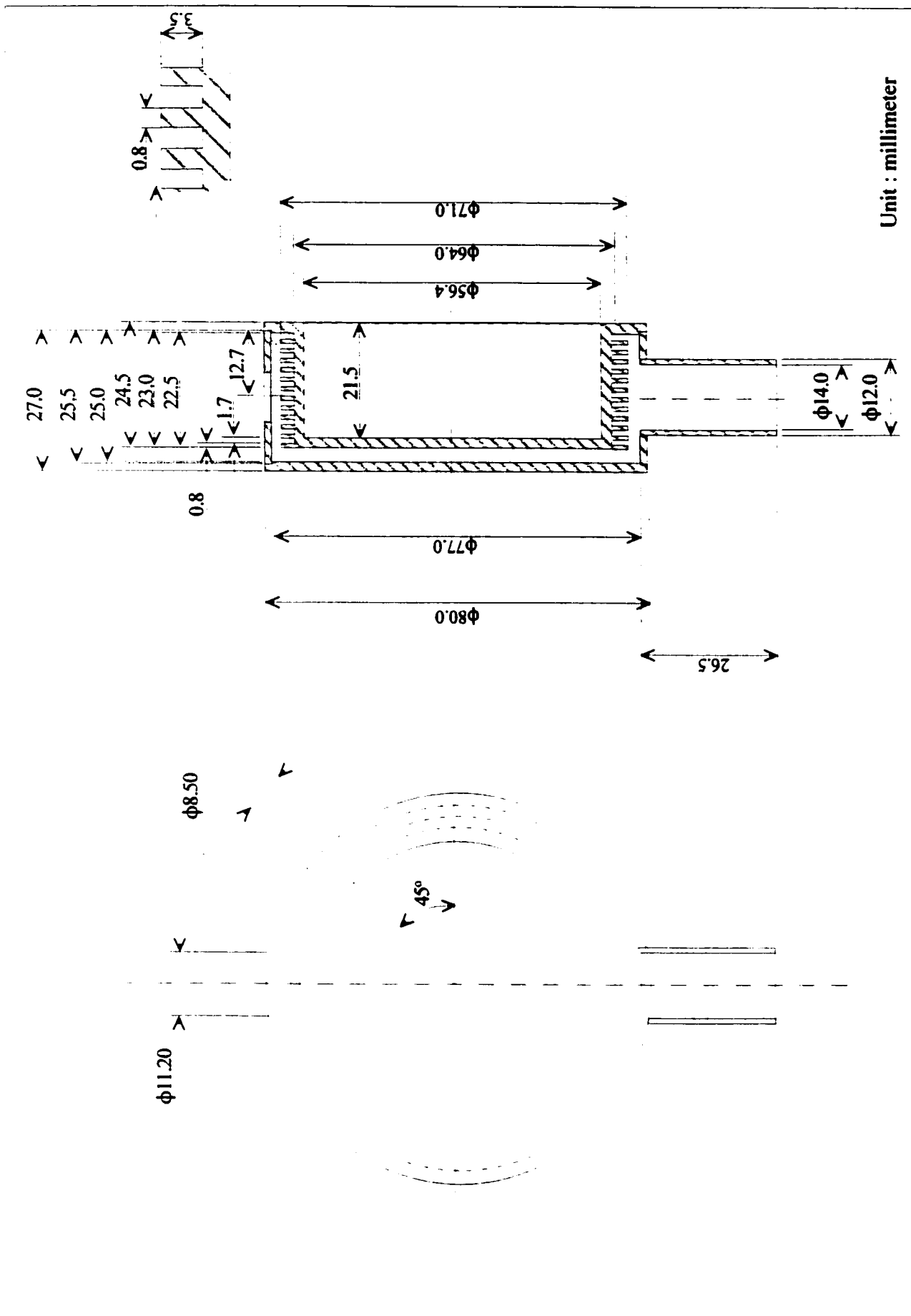


Figure 3.9. Condenser Section (MSL I)

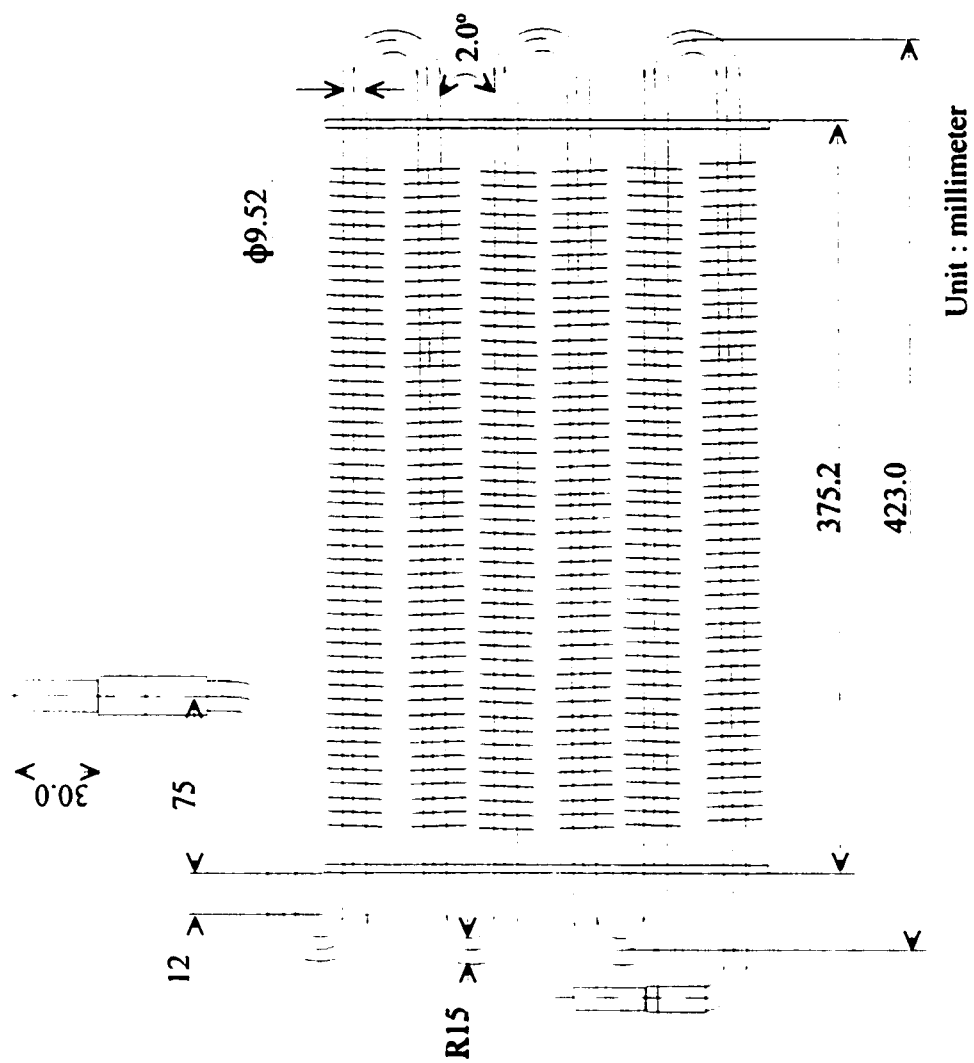


Figure 3.10. Evaporator Section (MSL I - Freezer side)

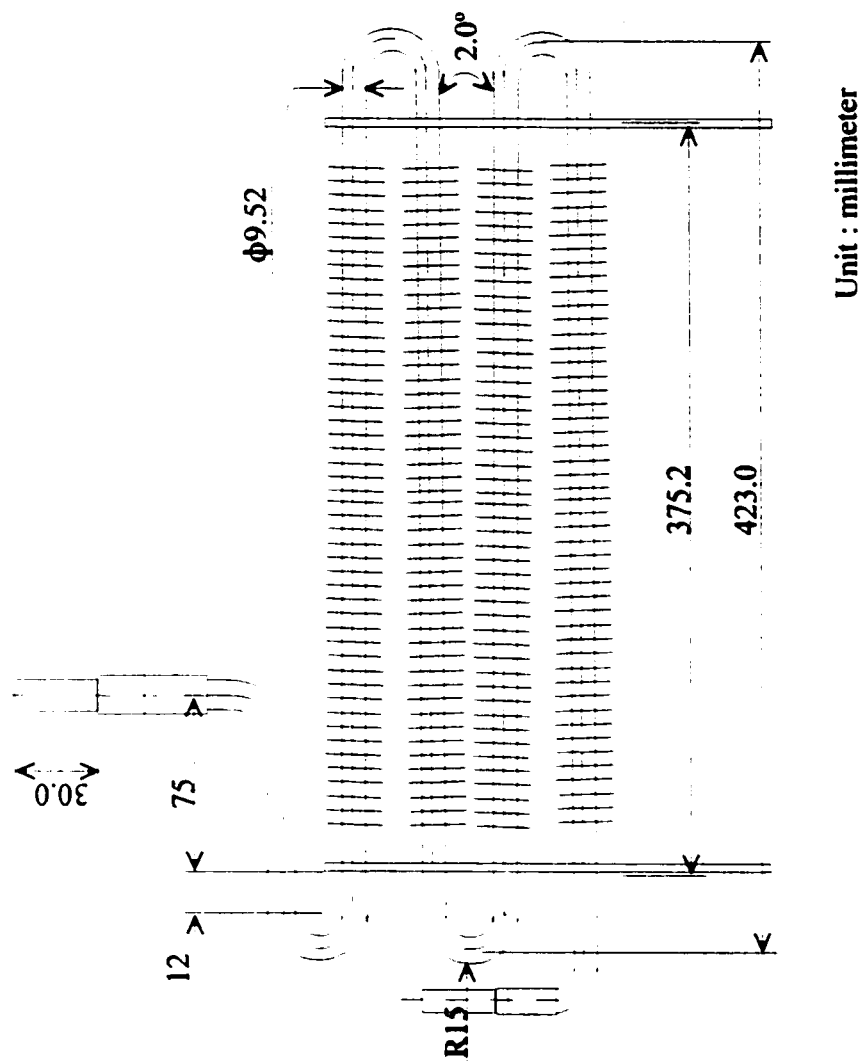


Figure 3.11. Evaporator Section (MSL I - Refrigerator side)



Figure 3.12 Test Assembly of Medium Scale Loop II (MSL II)

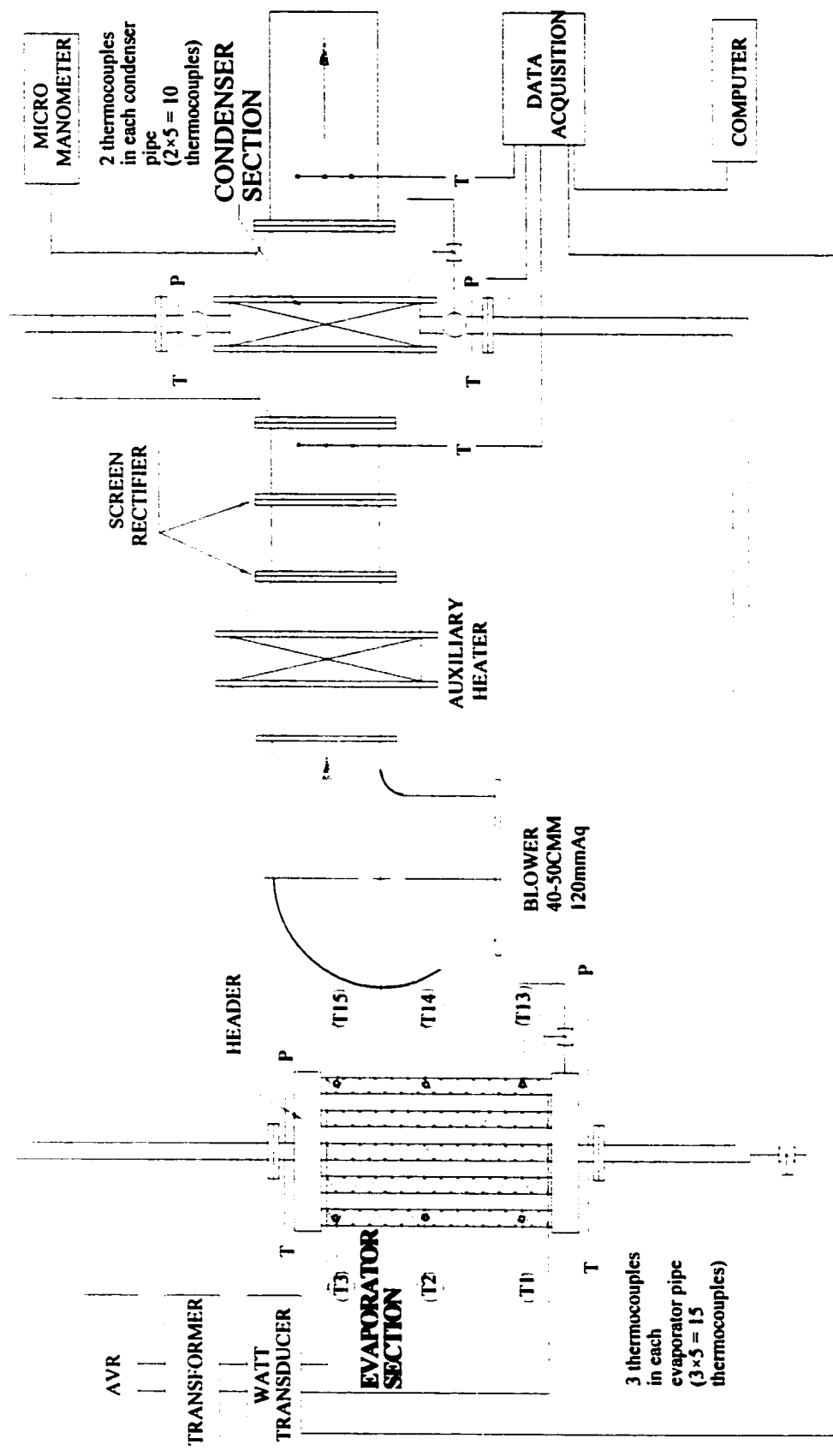


Figure 3.13 Experimental Setup of Large Scale Loop I (LSL I)

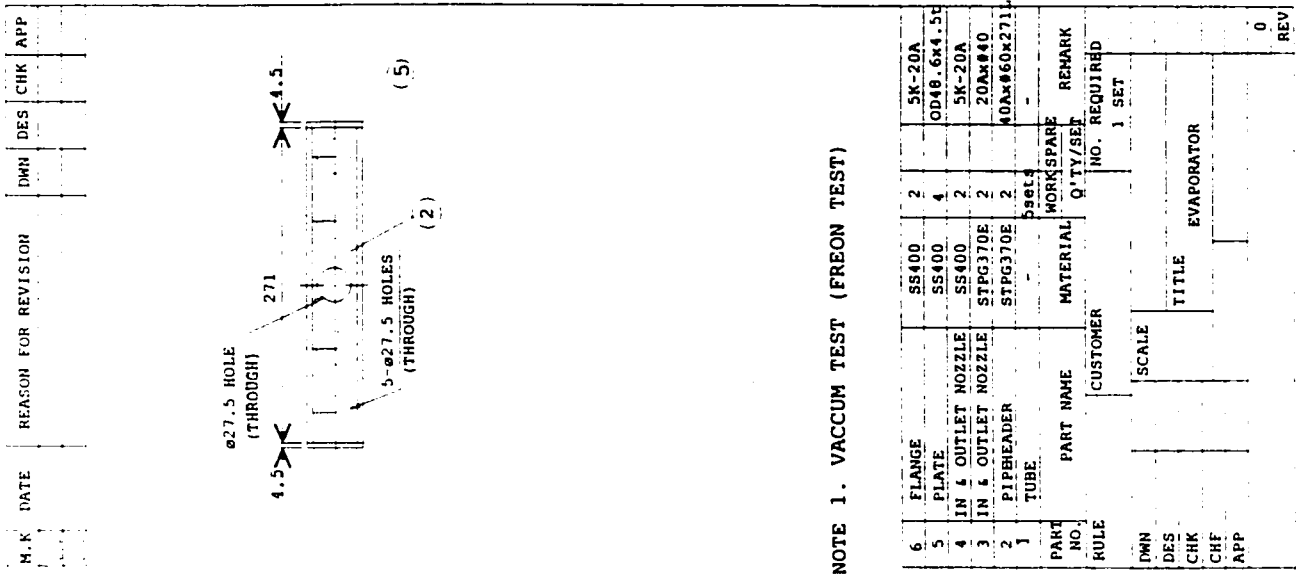


Figure 3.14. Evaporator Section (LSL I)

N.K.	DATE	REASON FOR REVISION	DWN DES	CHK	APP

NOTE 1. VACCUUM TEST (FREON TEST)

6	PLATE	SS400	4	OD48.6x4.5t
5	IN & OUTLET NOZZLE	SS400	2	5K-20A
4	IN & OUTLET NOZZLE	STPG370E	2	20A-#40
3	PIPE HEADER	STPG370E	2	40A-#60-271L
2	CONDENSER UNIT	SS400	1set	4.5t
1	FIN TUBE		5sets	
RULE		CUSTOMER		1 SET
DWN				
DES				
CHK				
APP				
		TITLE		CONDENSER
				0

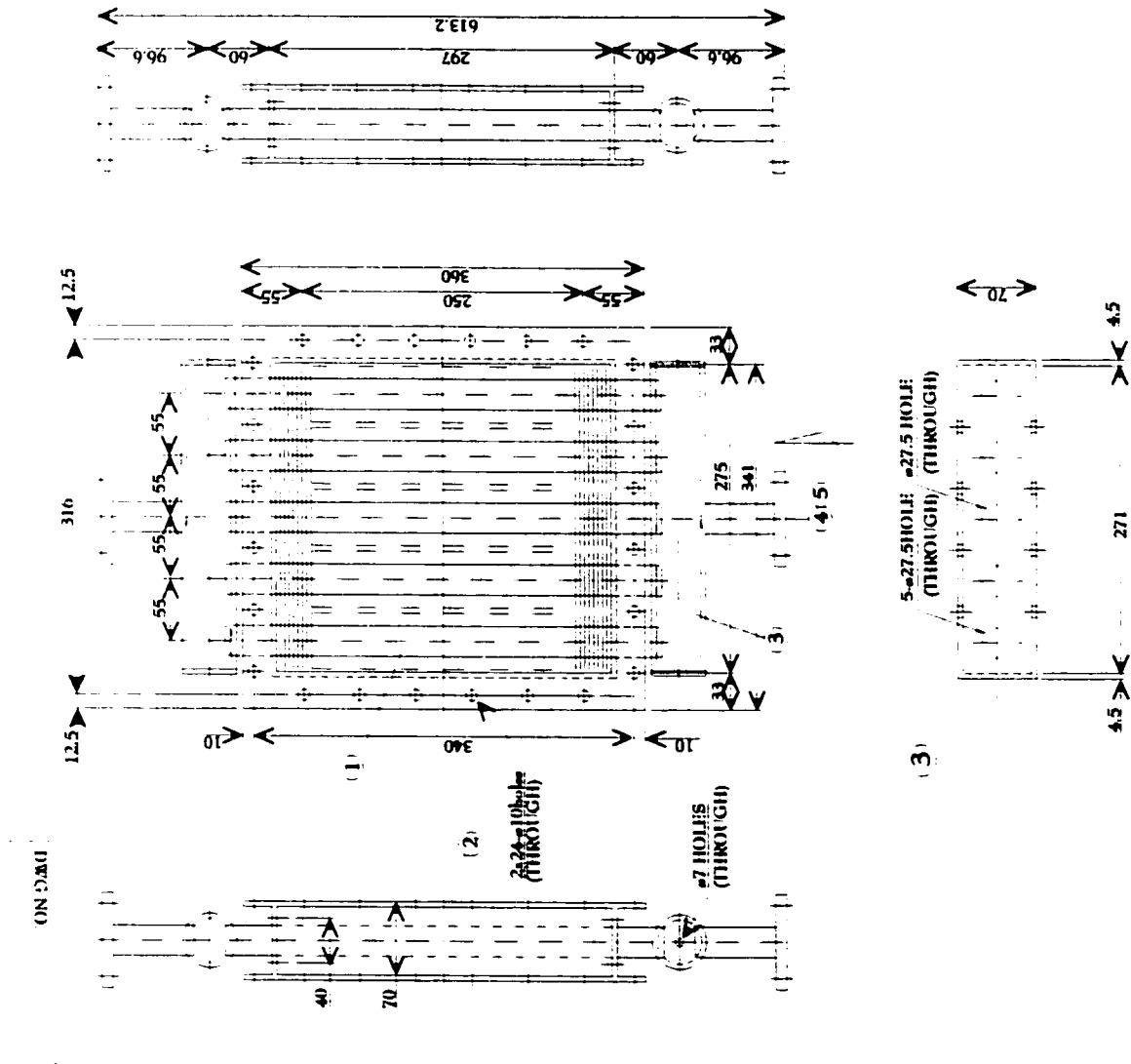


Figure 3.15. Condenser Section (LSL I)

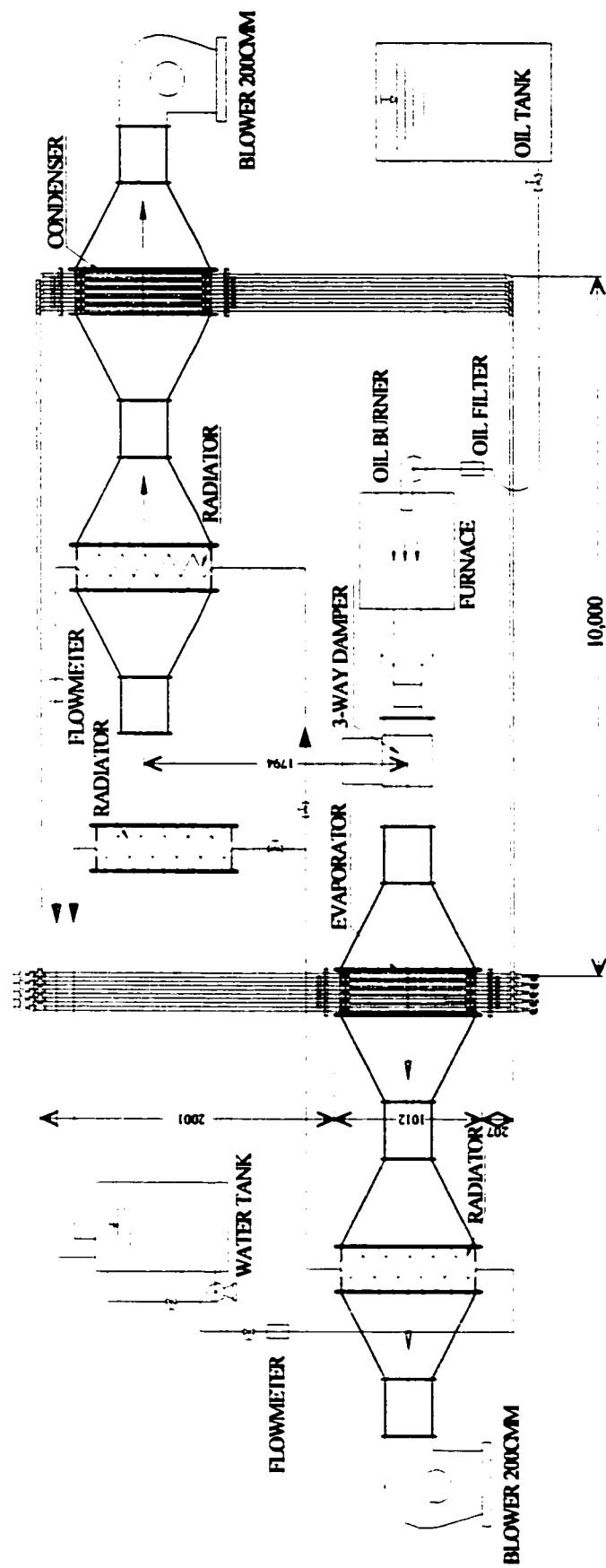


Figure 3.16. Experimental Setup of Large Scale Loop II (LSL II)

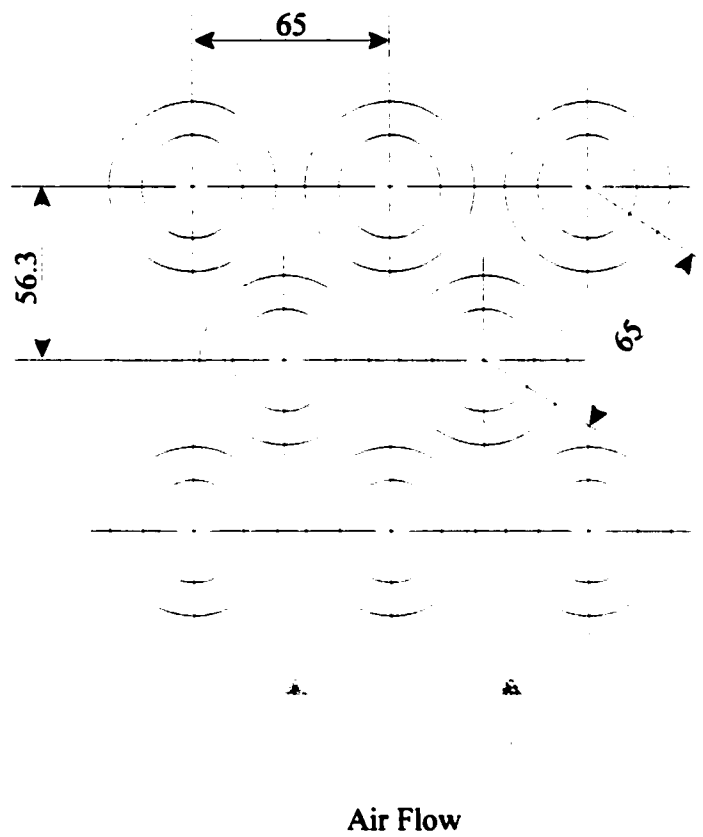


Figure 3.18 Pipe Configuration of Evaporator and Condenser (LSL II)

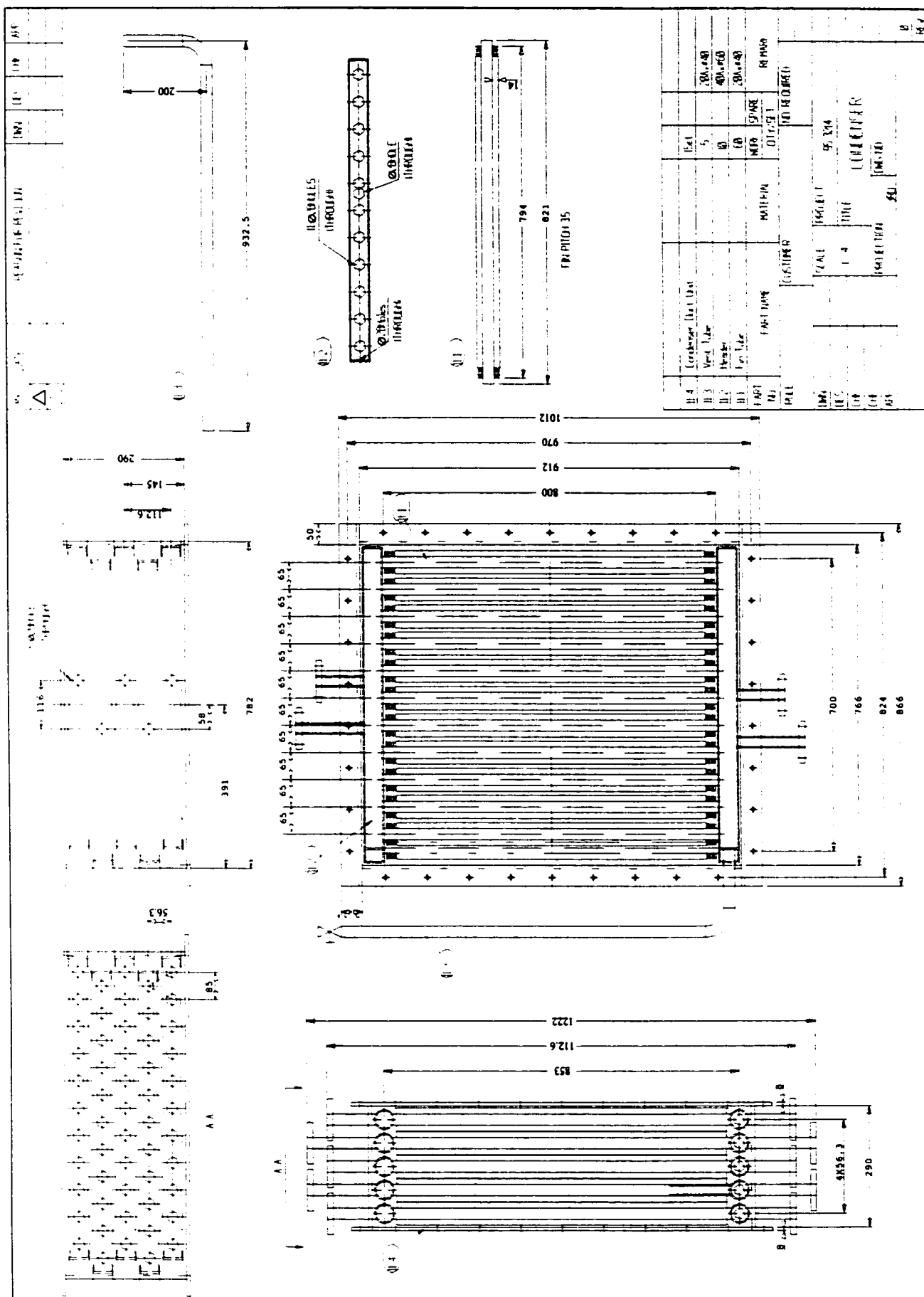
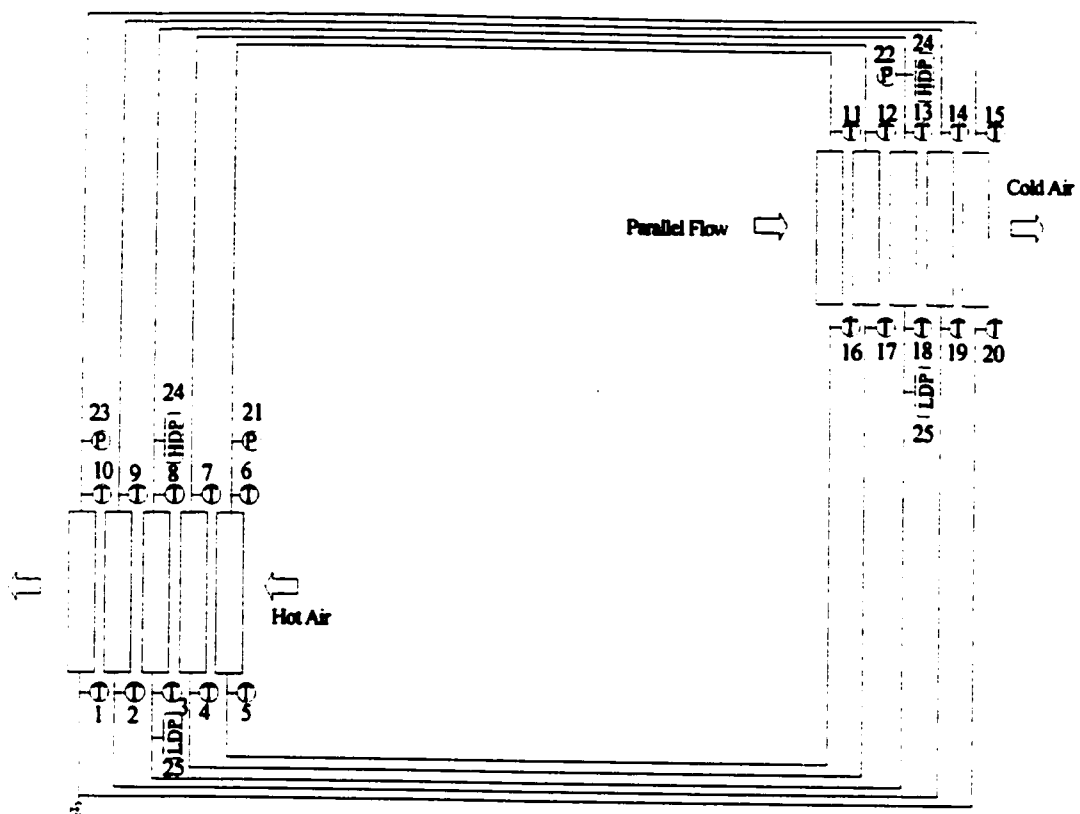
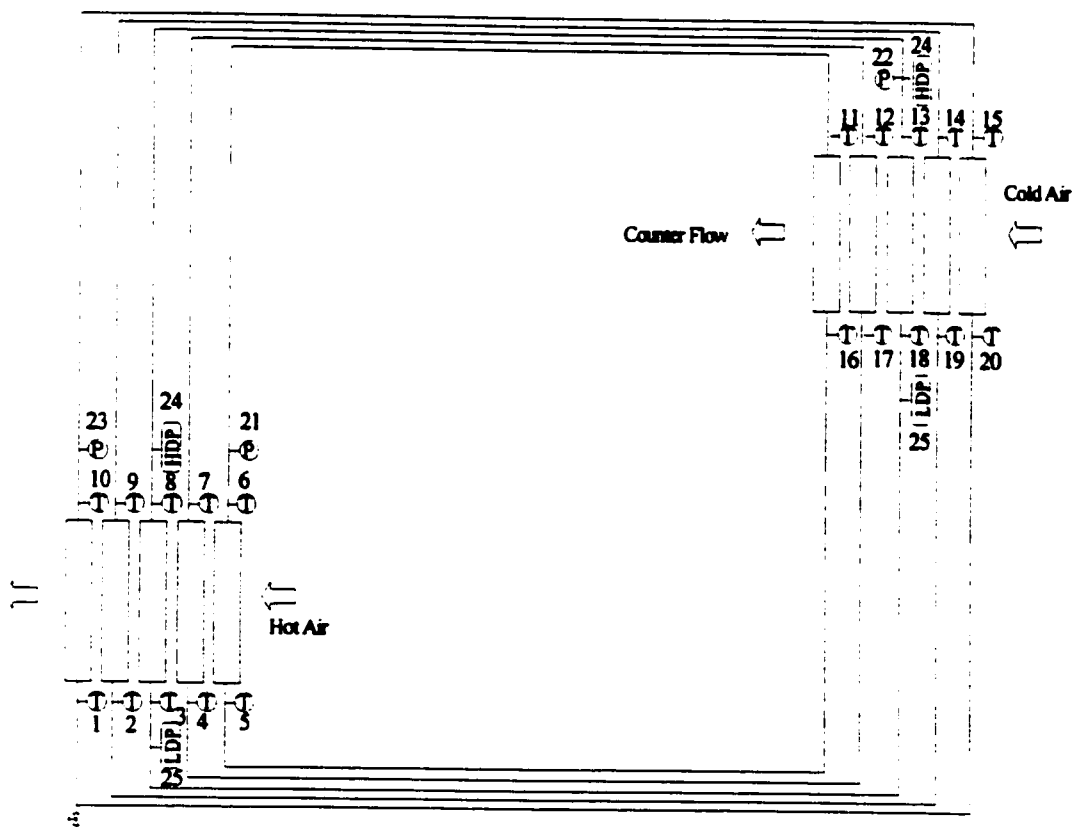


Figure 3.19 Condenser Section (LSL II)



(a) parallel flow of cold air in condenser section



(b) Counter flow of cold air in condenser section

Figure 3.20. Positions of Thermocouples and Pressure Tranducers (LSL II)

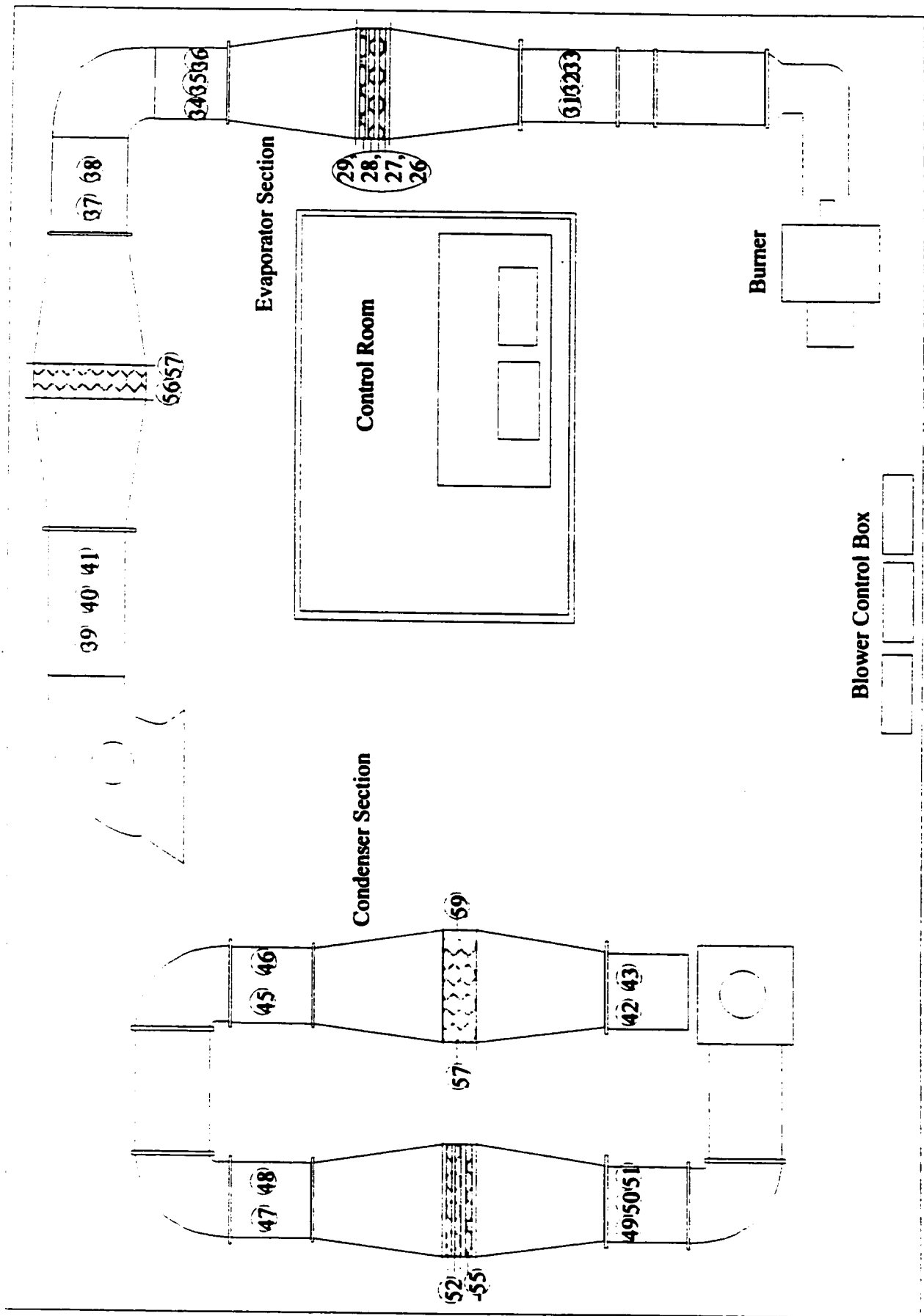


Figure 3.21. Positions of Thermocouples in LSL II Experimental Setup (LSL II)

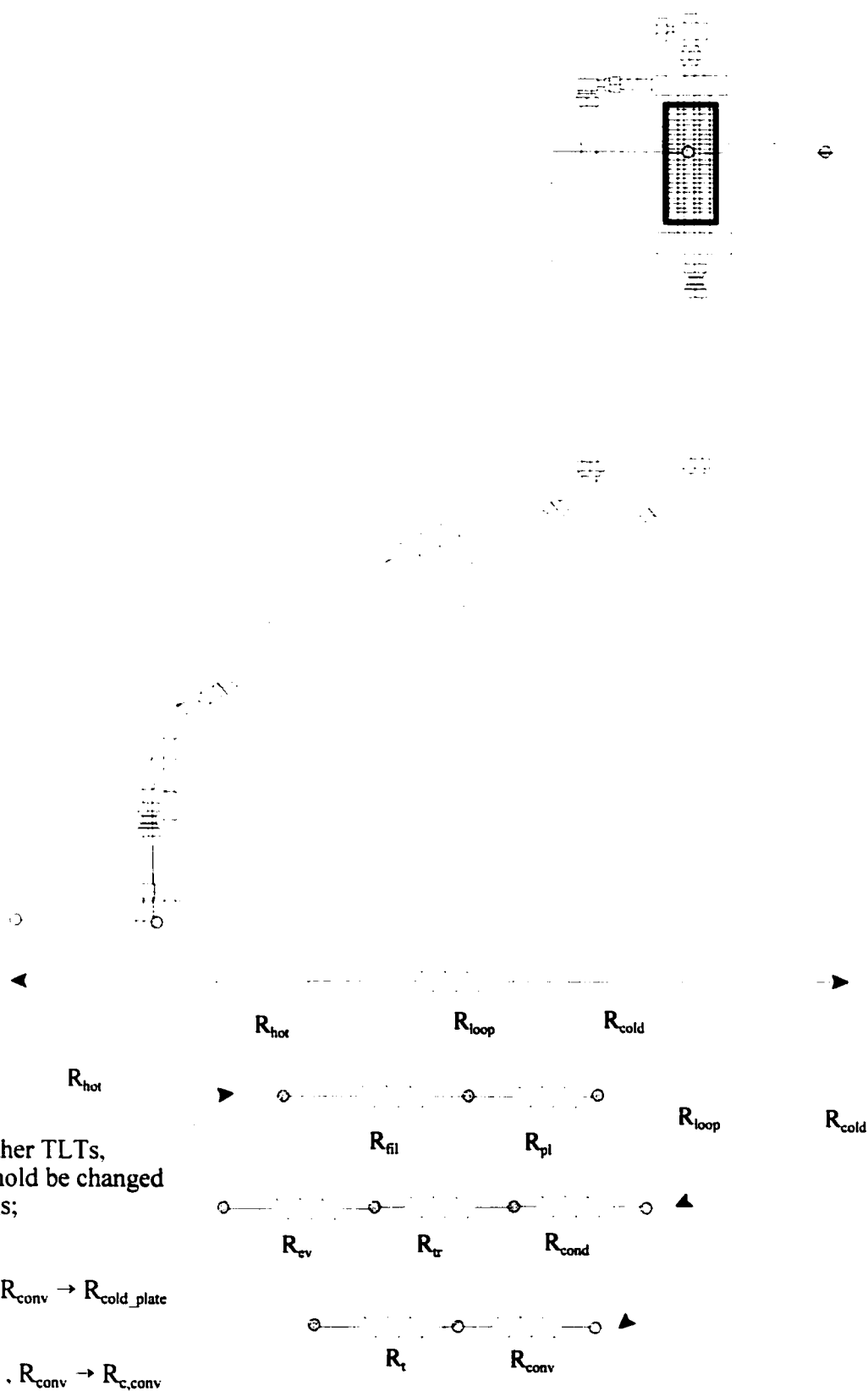


Figure 4.1. Thermal Resistance Network

Two-Phase Loop Thermosyphon with Single Pipe Evaporator <Lumped Method>

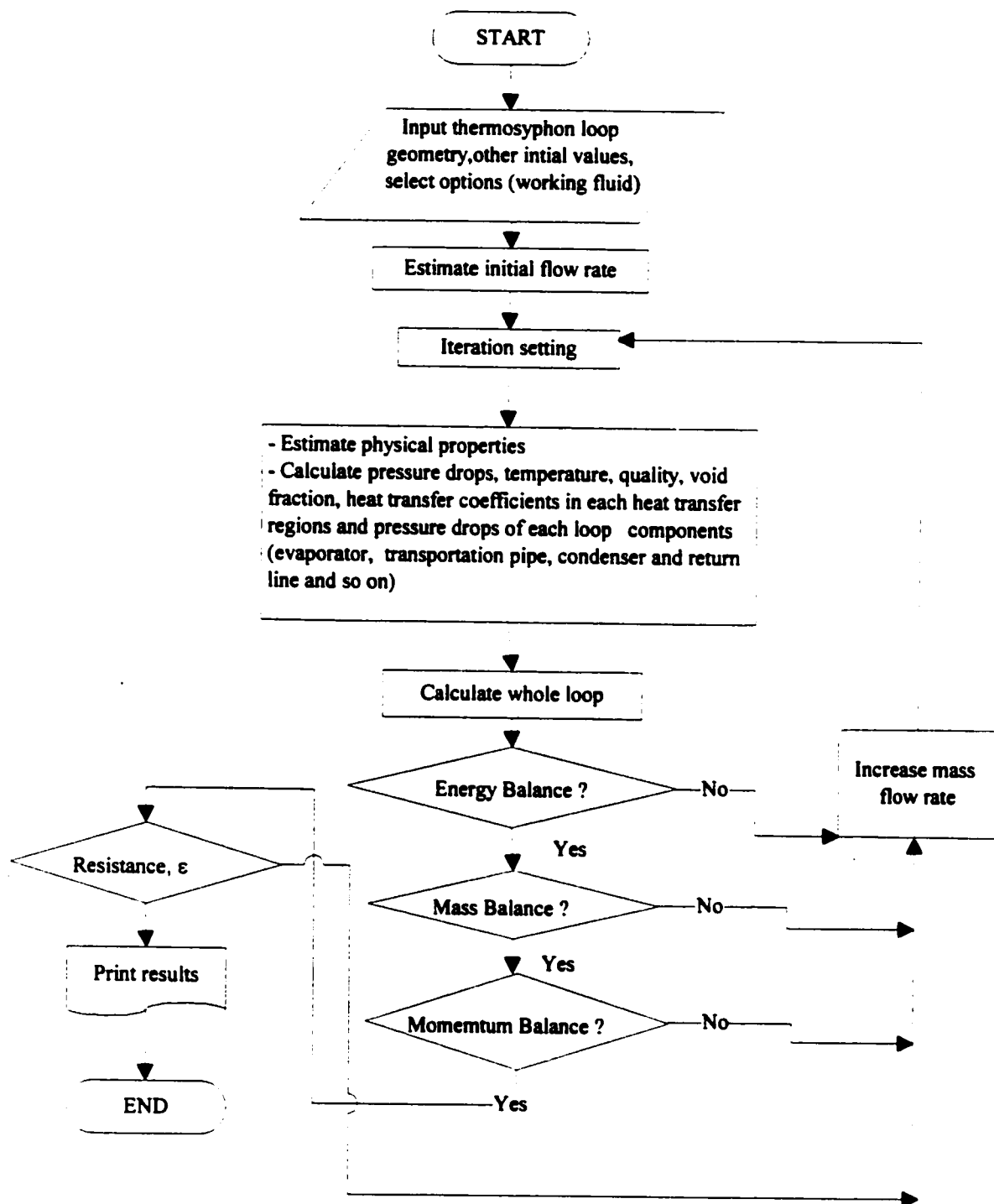


Figure 4.2 (a) Flow Chart for Single Loop Lumped Method

**Two-Phase Loop Thermosyphon with Multiple Pipe Evaporator
<Lumped Method>**

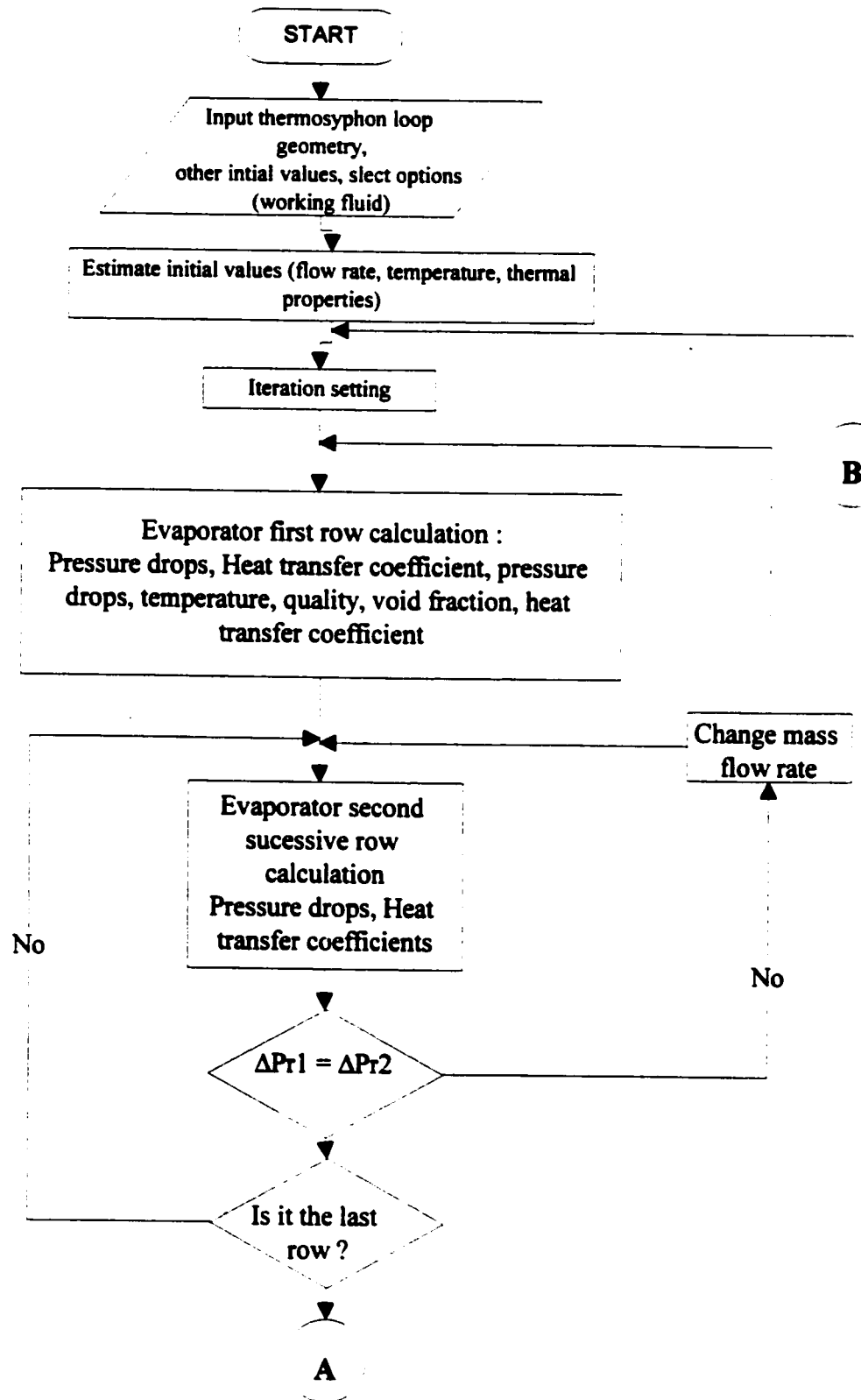


Figure 4.2 (b) Flow Chart for Multi Pipe Loop Lumped Method

Two-Phase Loop Thermosyphon with Multiple Pipe Evaporator
<Lumped Method>

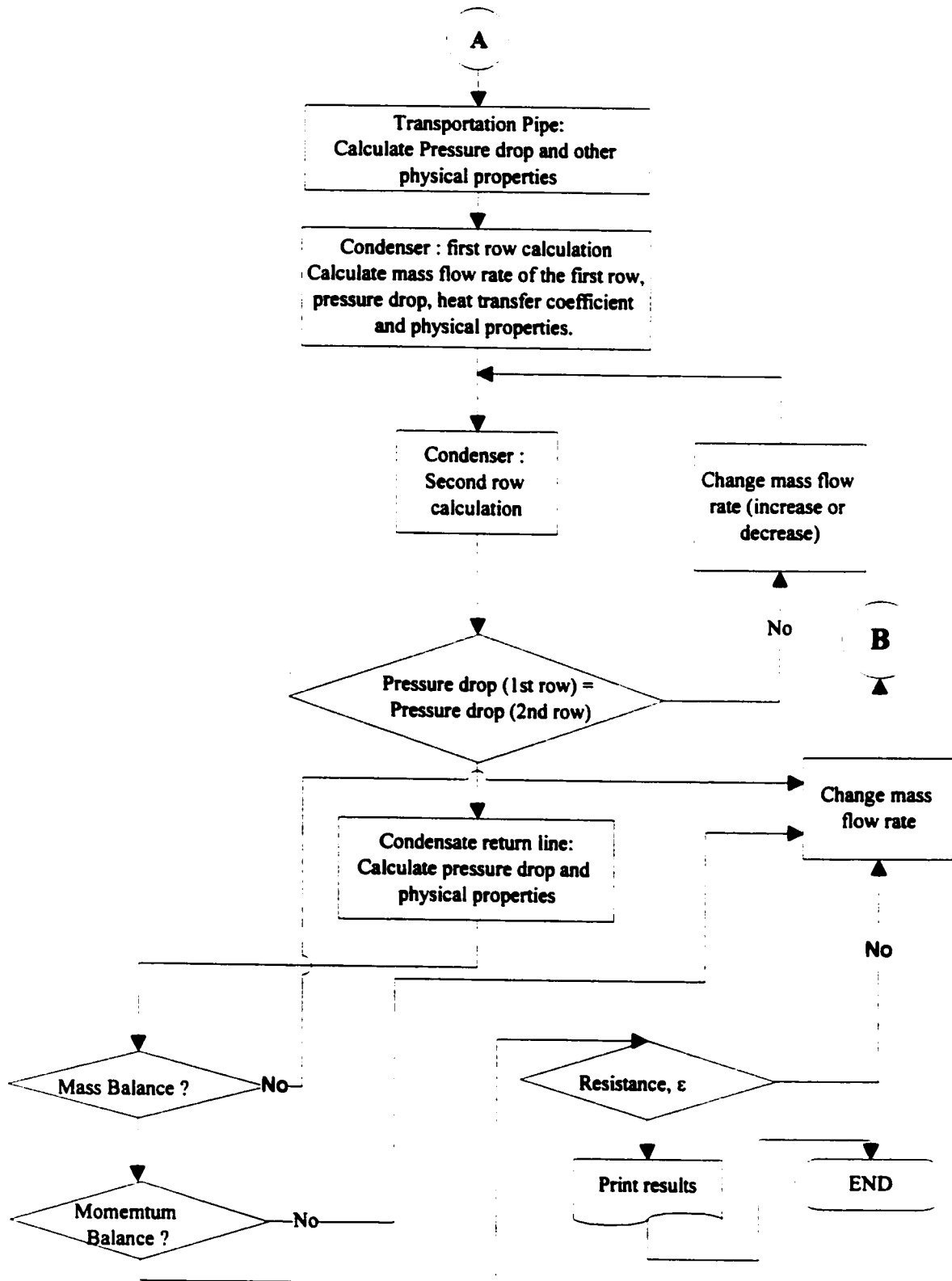


Figure 4.2 (b) Flow Chart for Multi Pipe Loop Lumped Method (Continued)

Two-Phase Loop Thermosyphon with Single Pipe Evaporator <Sectorial Method>

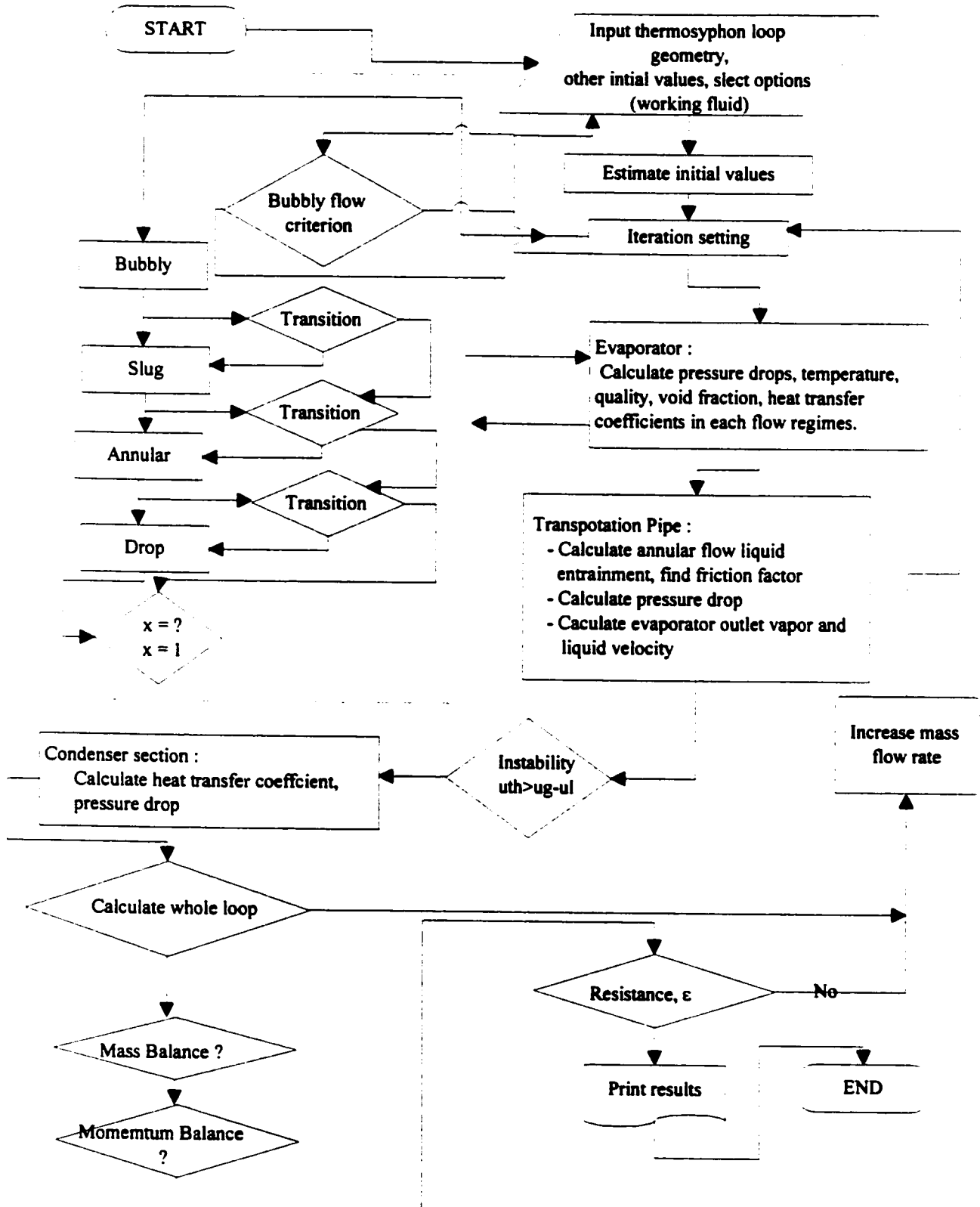


Figure 4.2 (c) Flow Chart for Single Loop Sectorial Method

Two-Phase Loop Thermosyphon with Multiple Pipe Evaporator <Sectorial Method>

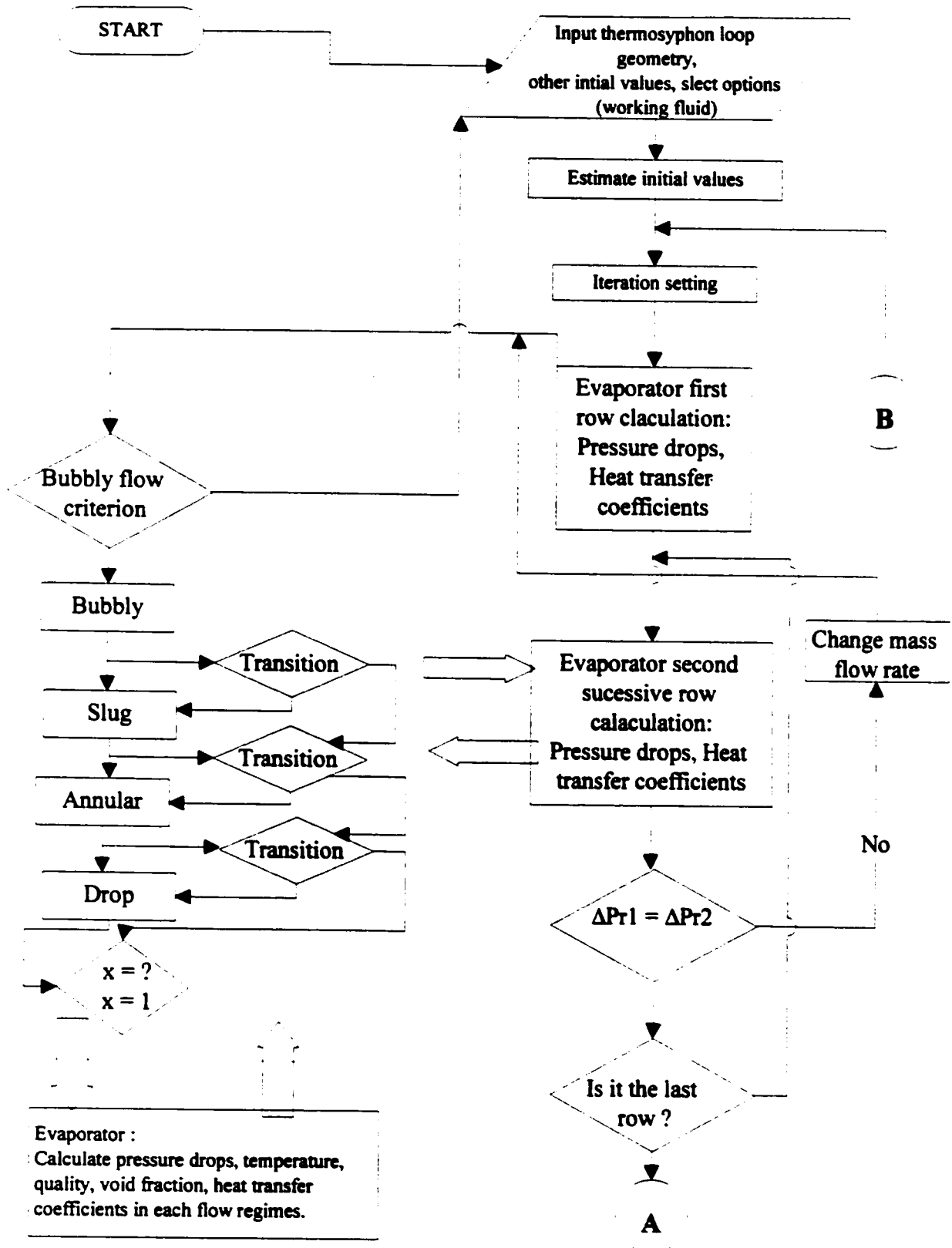


Figure 4.2 (d) Flow Chart for Multi Pipe Loop Sectorial Method

Two-Phase Loop Thermosyphon with Multiple Pipe Evaporator
<Sectorial Method>

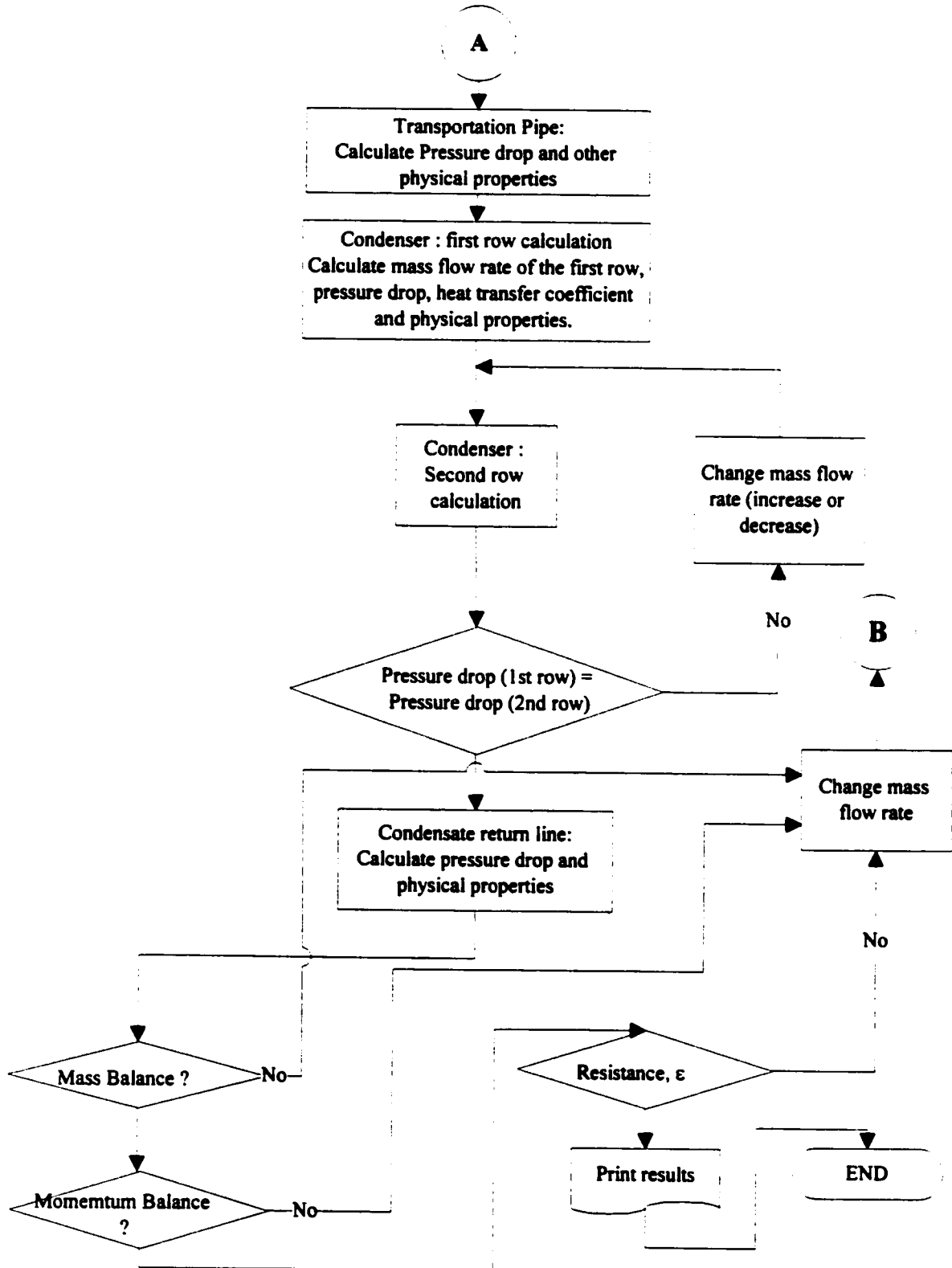
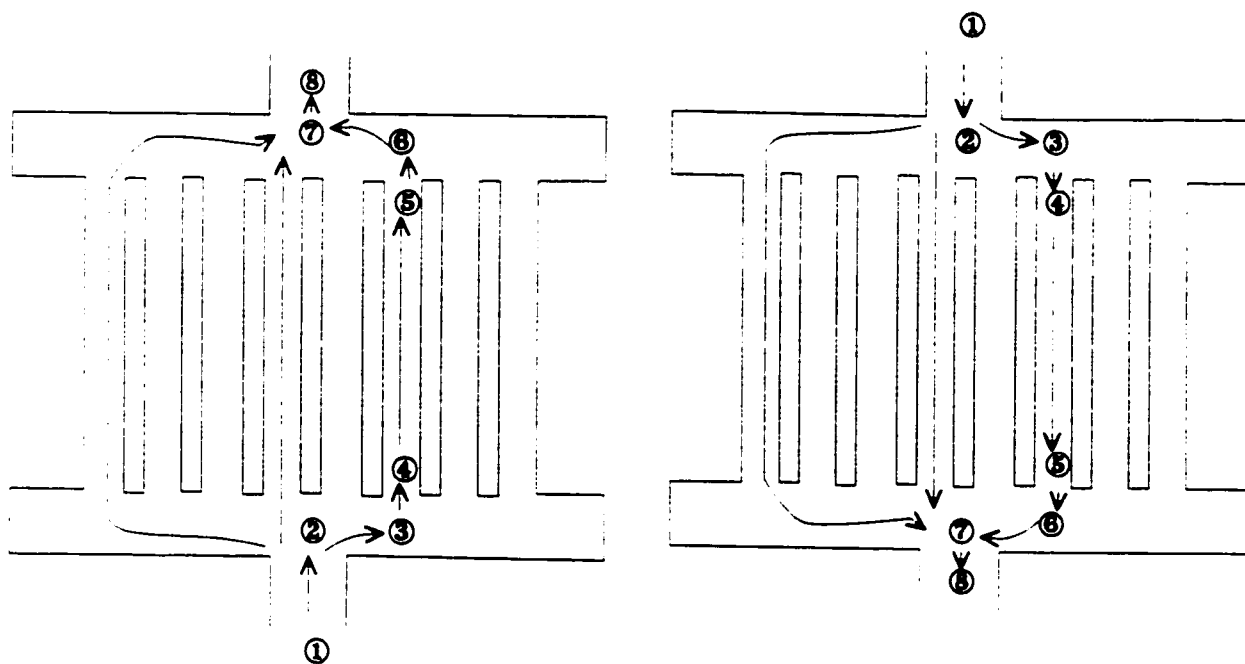


Figure 4.2 (d) Flow Chart for Multi Pipe Loop Sectorial Method (Continued)



(a) Evaporator Section

(b) Condenser Section

- 1-2 : Sudden expansion
- 2-3 : Flow turning
- 3-4 : Sudden contraction
- 4-5 : Two-phase flow Pressure drop
- 5-6 : Sudden expansion
- 6-7 : Flow turning
- 7-8 : Sudden contraction

Figure 4.3 Pressure Drop Consideration for Evaporator and Condenser Element (LSL I and LSL II)

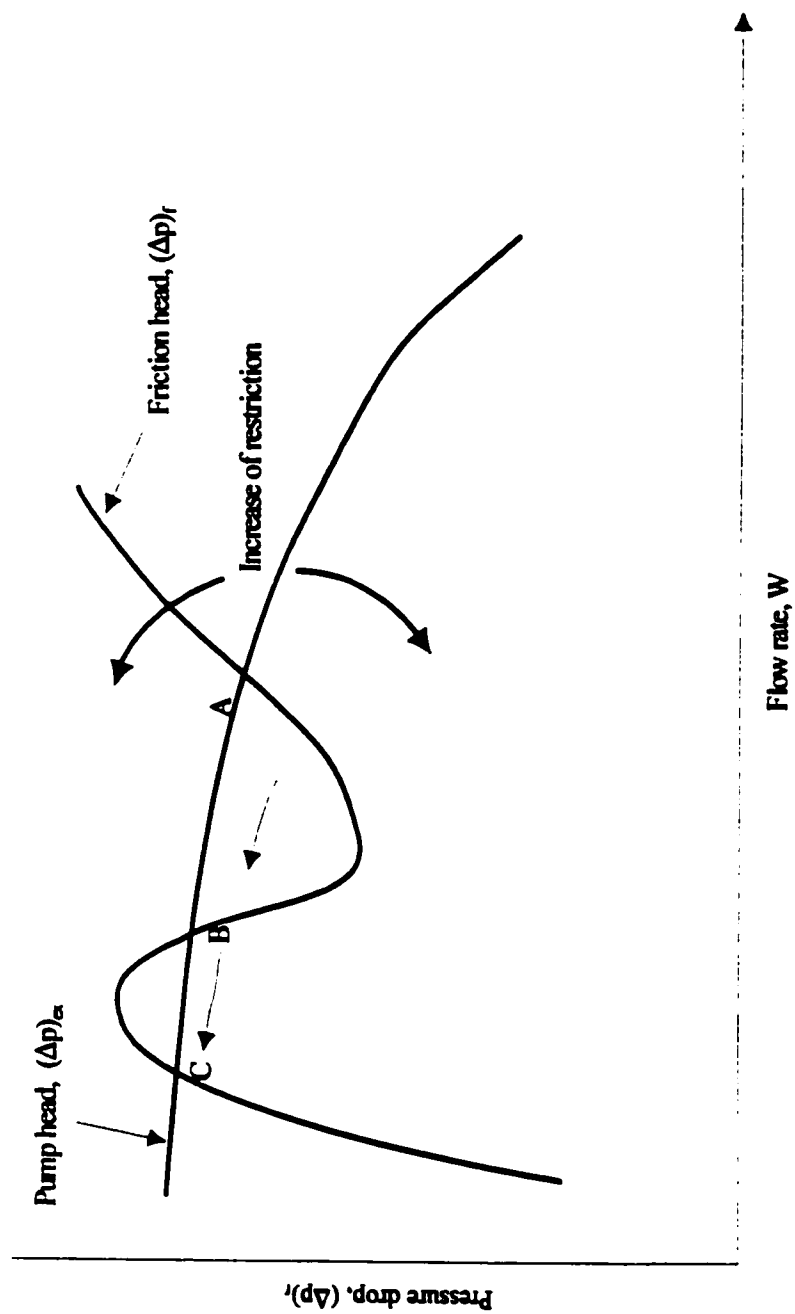


Figure 4.4 Supply and Demand of Pressure Heads [Hsu, 1976]

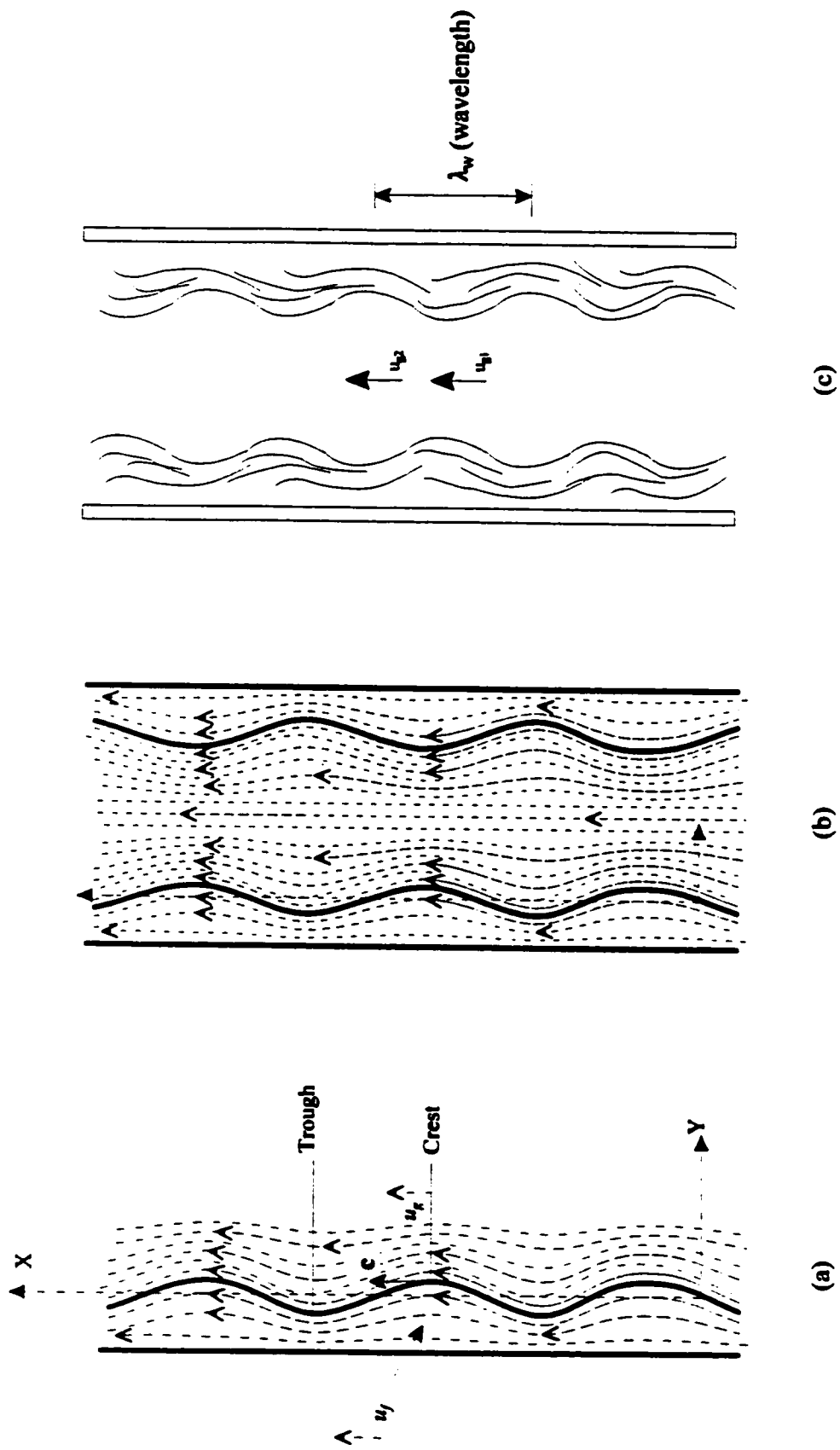


Figure 4.5 Streamlines in two-phase flow with a wavy interface. [Hewitt, 1970]

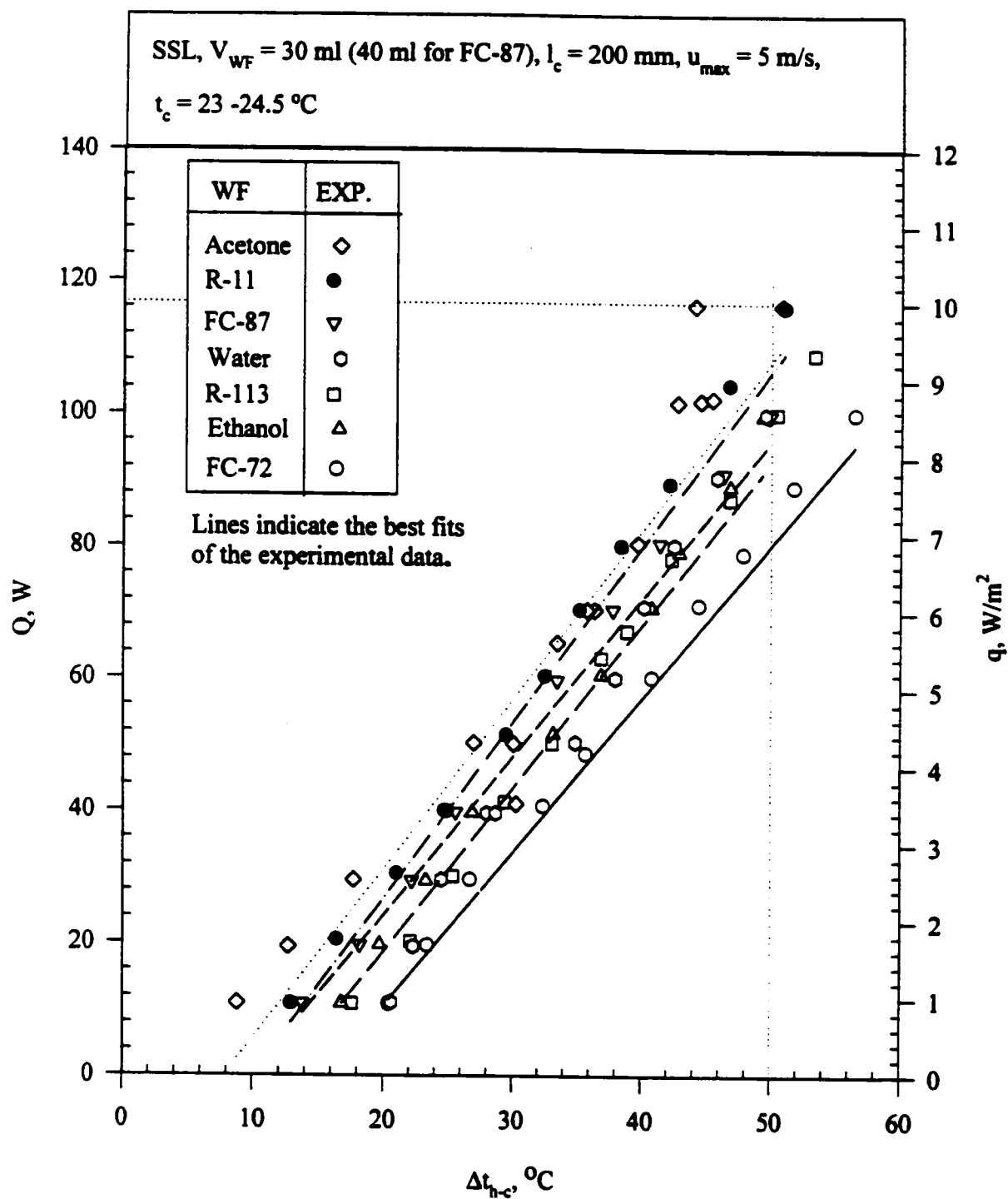


Figure 5.1 (a) Effect of Δt_{h-c} on Q and q ; Forced Convection Mode with One Condenser, SSL

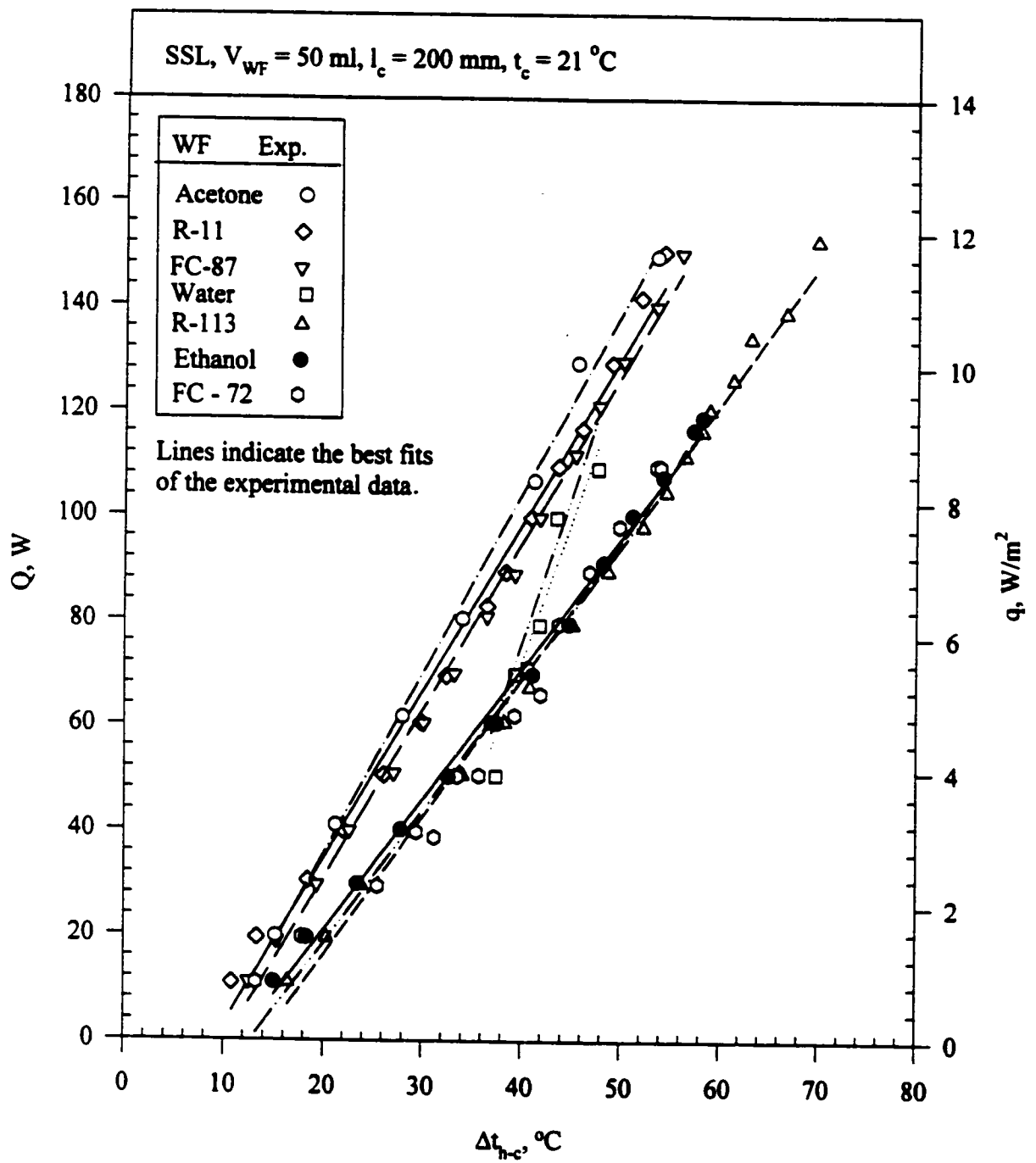


Figure 5.1 (b) Effect of Δt_{h-c} on Q and q
with Two Condensers, SSL

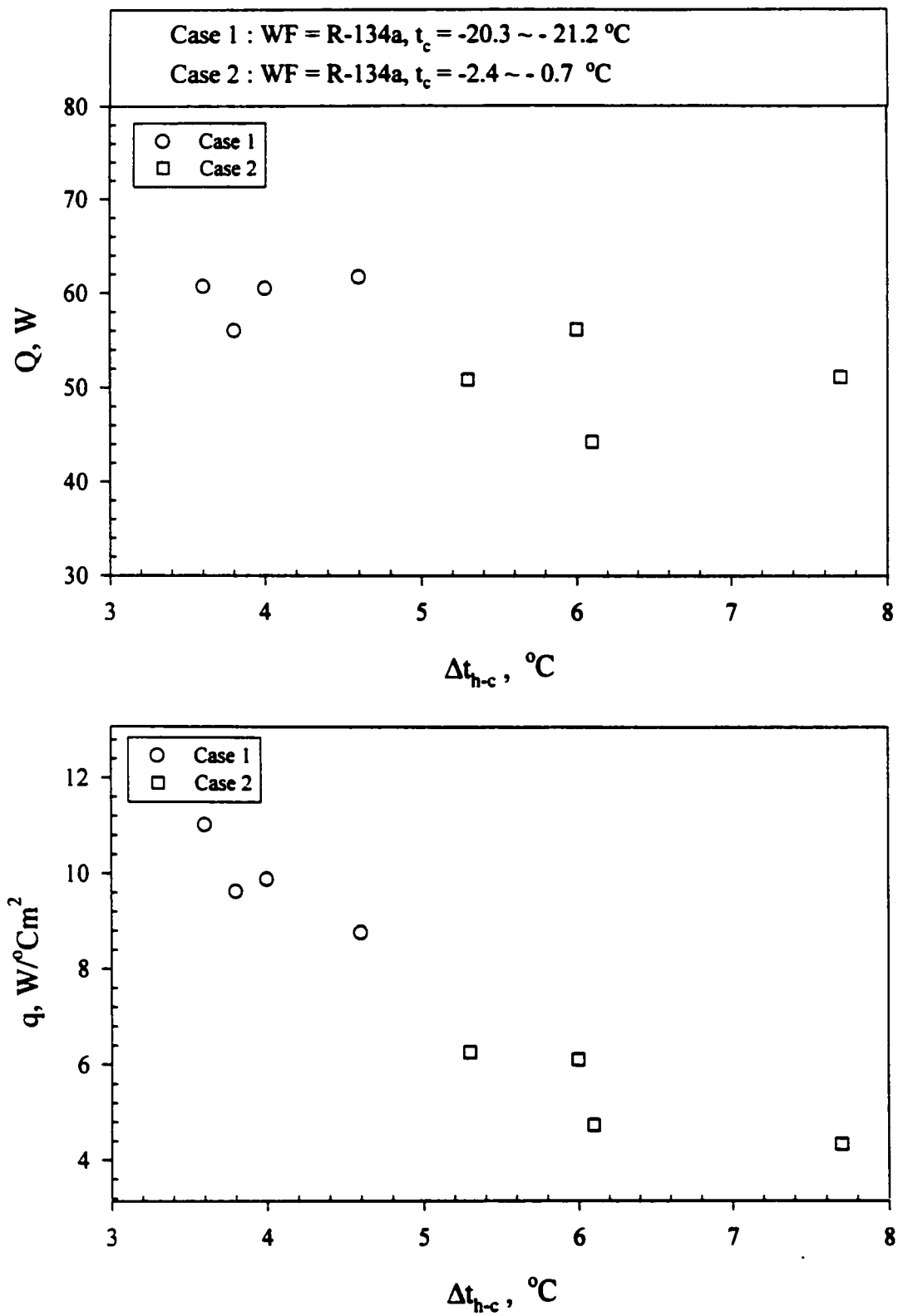


Figure 5.1 (c) Effect of Δt_{h-c} on Q and q ; MSL I

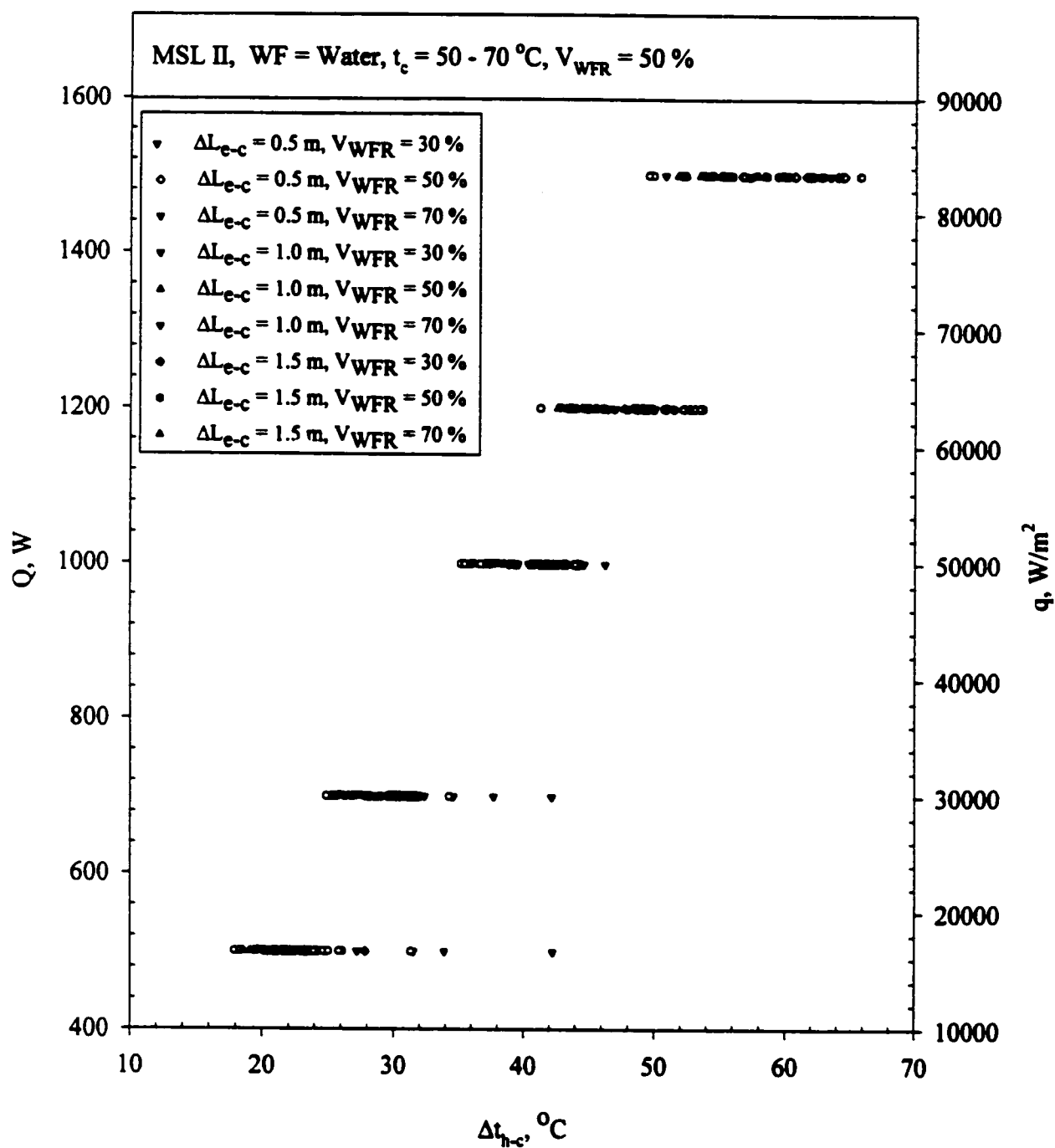


Figure 5.1 (d). Effect of Δt_{h-c} on Q and q ; MSL II

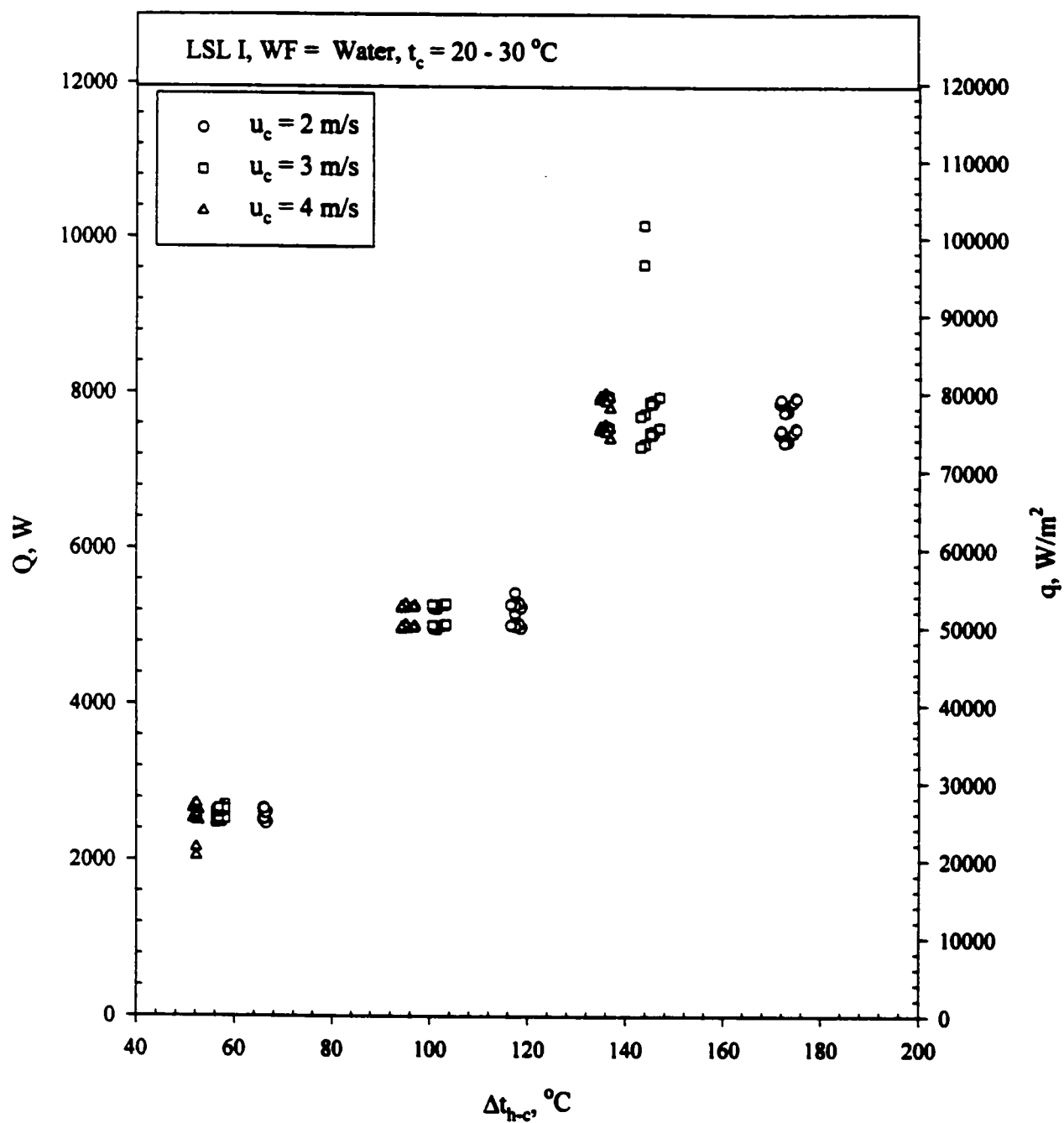


Figure 5.1 (e) Effect of Δt_{h-c} on Q and q ; LSL I

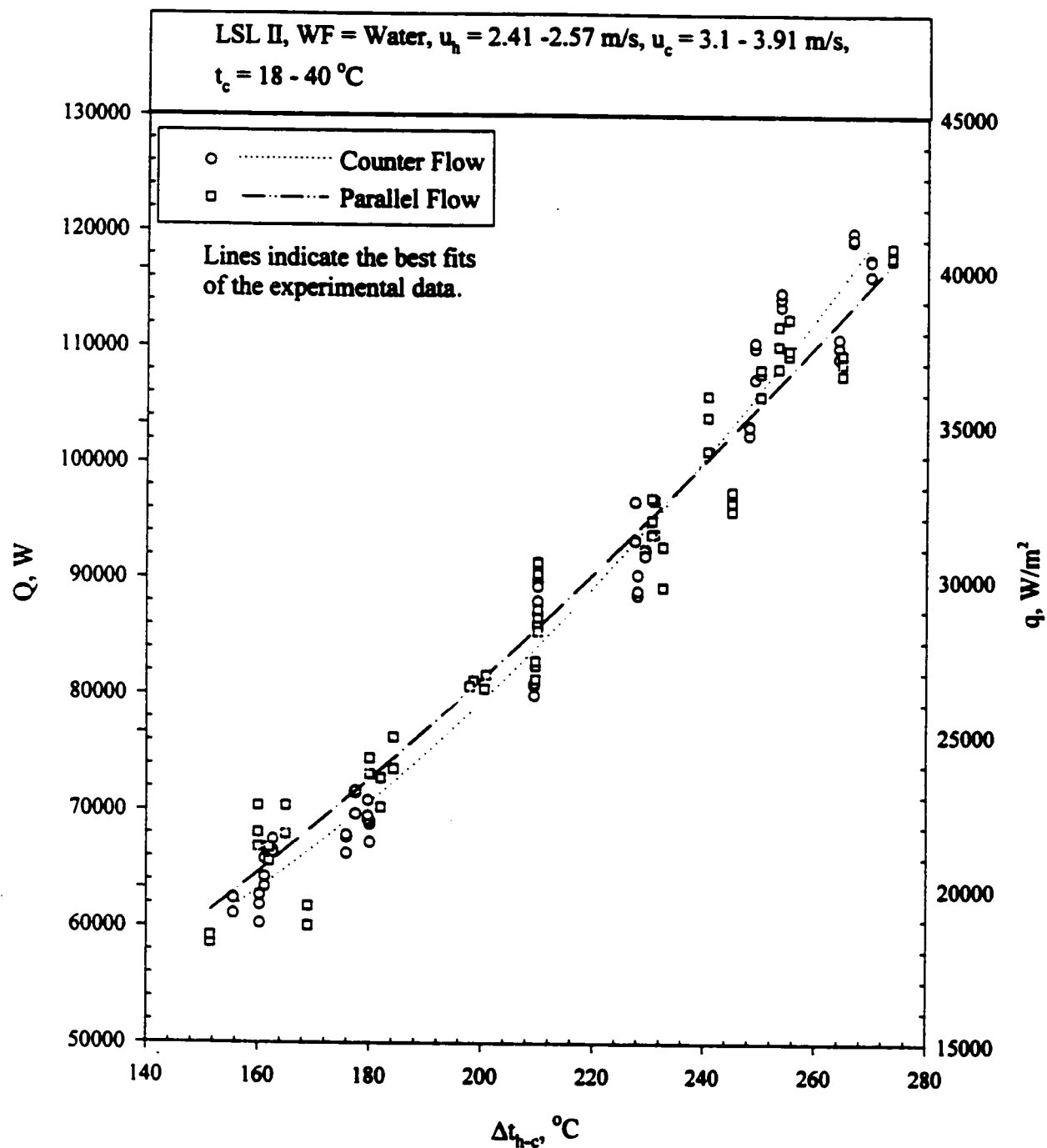


Figure 5.1 (f) Effect of Δt_{h-c} on and q ; LSL II

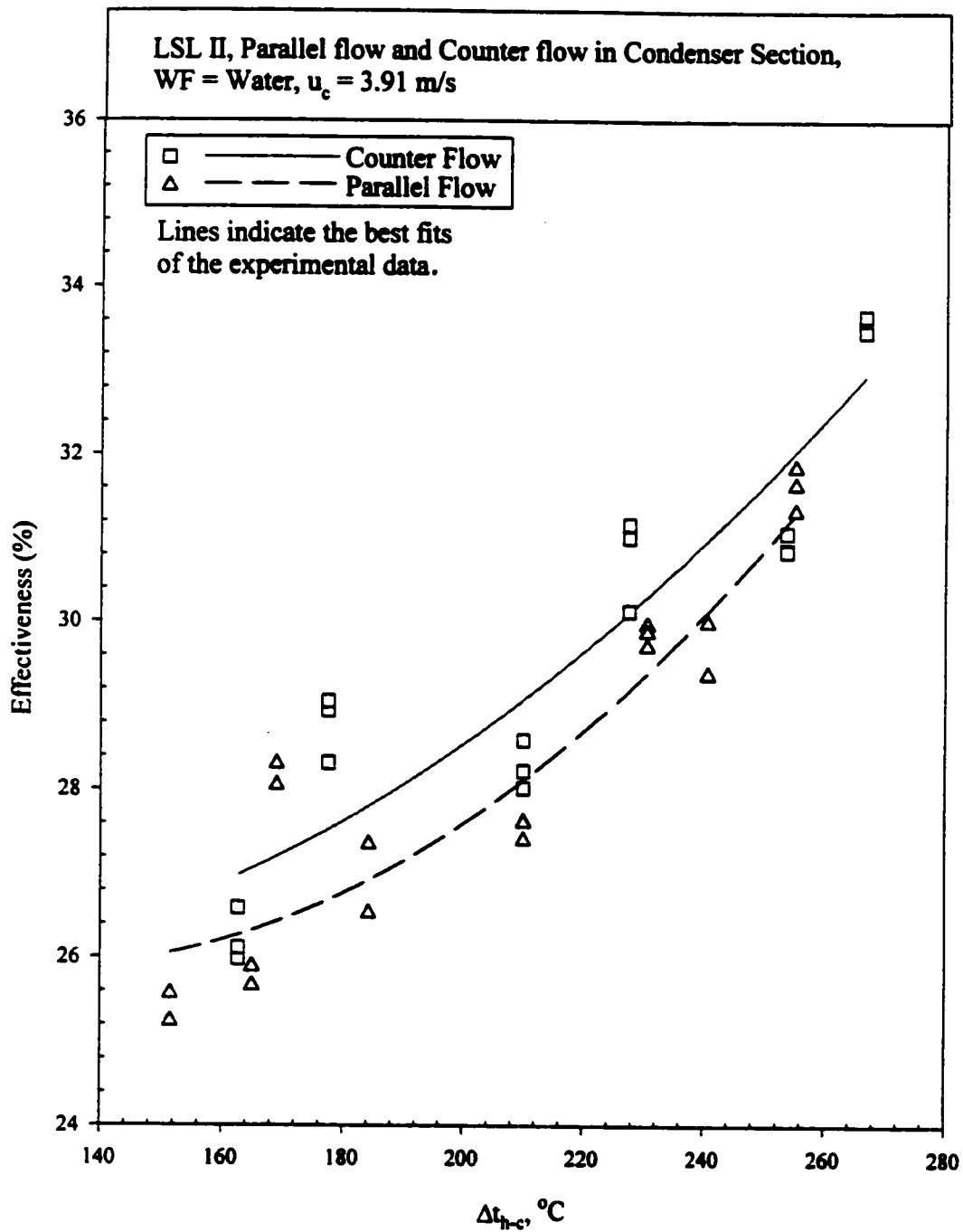


Figure 5.2 Effect of Δt_{h-c} on System Effectiveness; LSL II

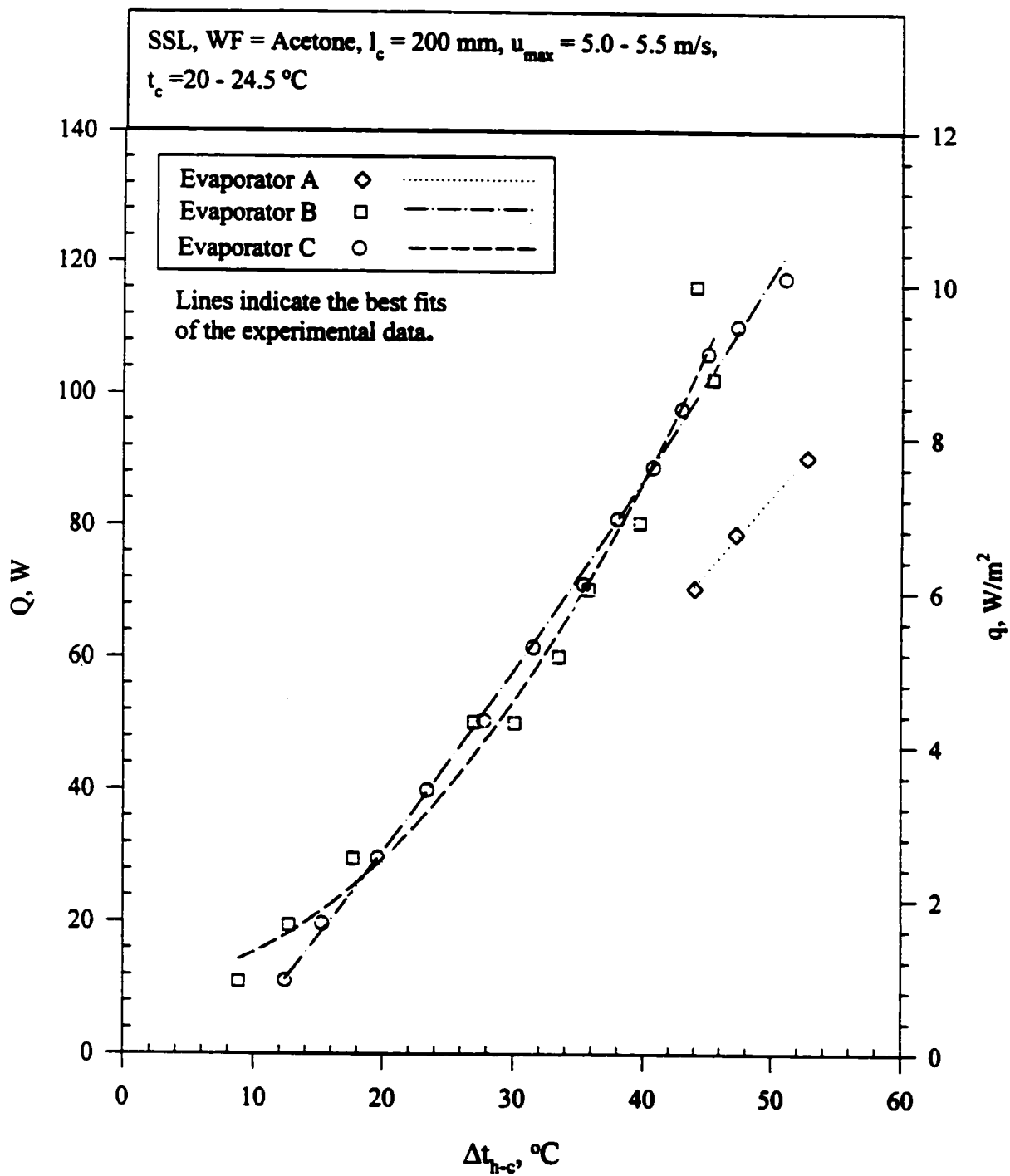


Figure 5.3 Effect of Δt_{h-c} on Q and q with Different Evaporator Configurations; SSL

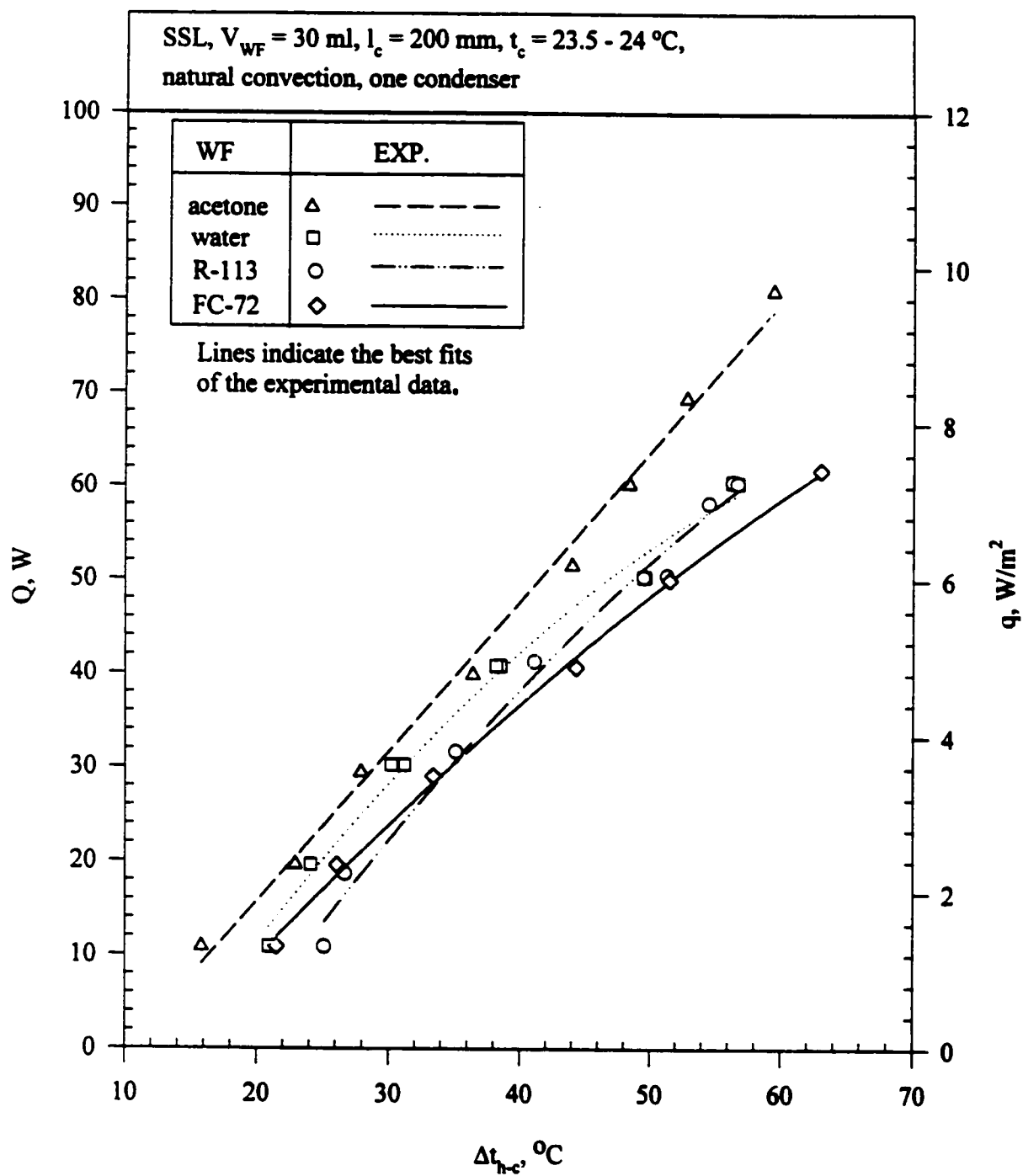


Figure 5.4 (a). Effect of Δt_{h-c} on Q and q ; natural convection mode with one condenser, SSL

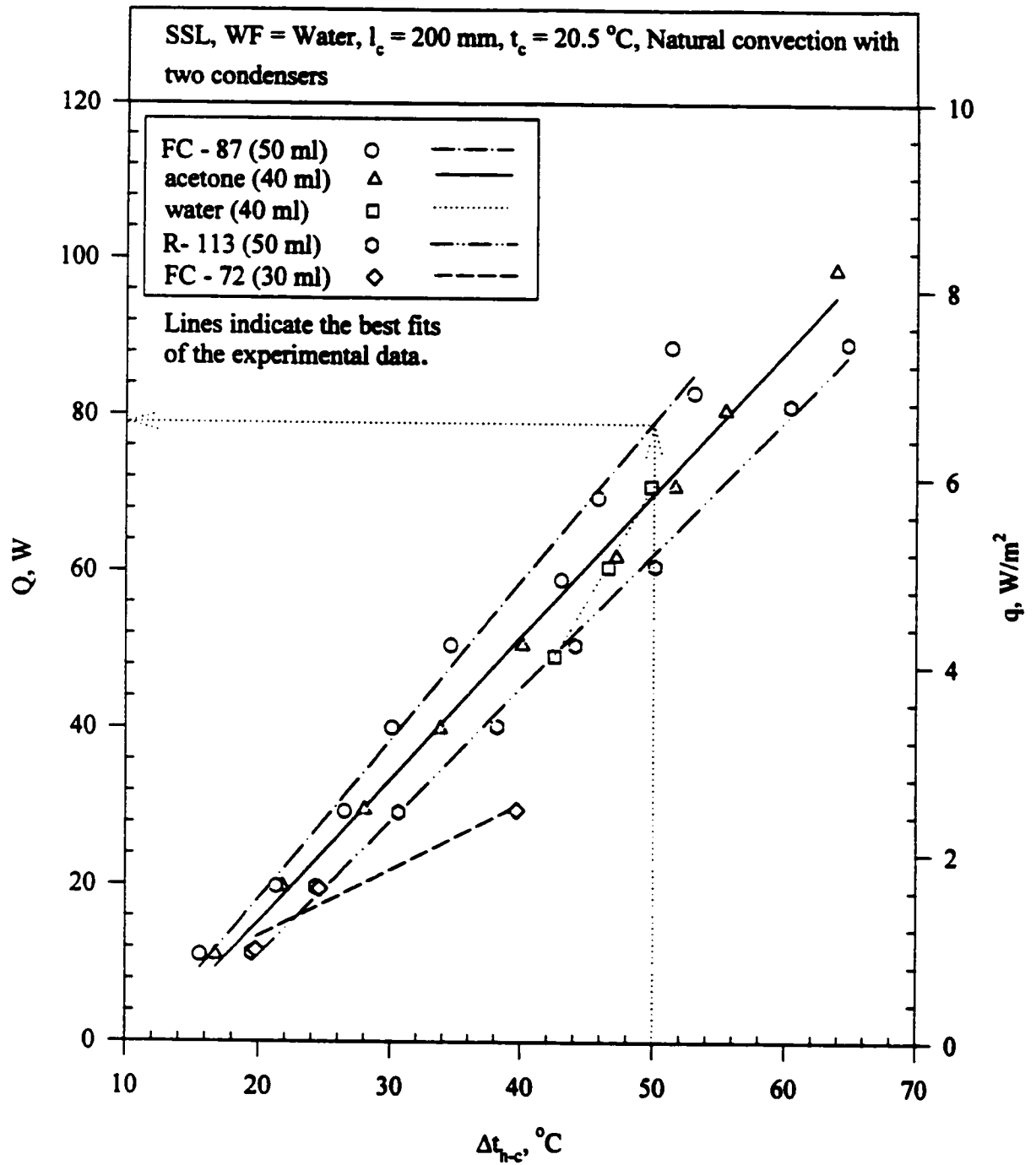


Figure 5.4 (b). Effect of Δt_{h-c} on Q and q ; natural convection mode with two condensers, SSL

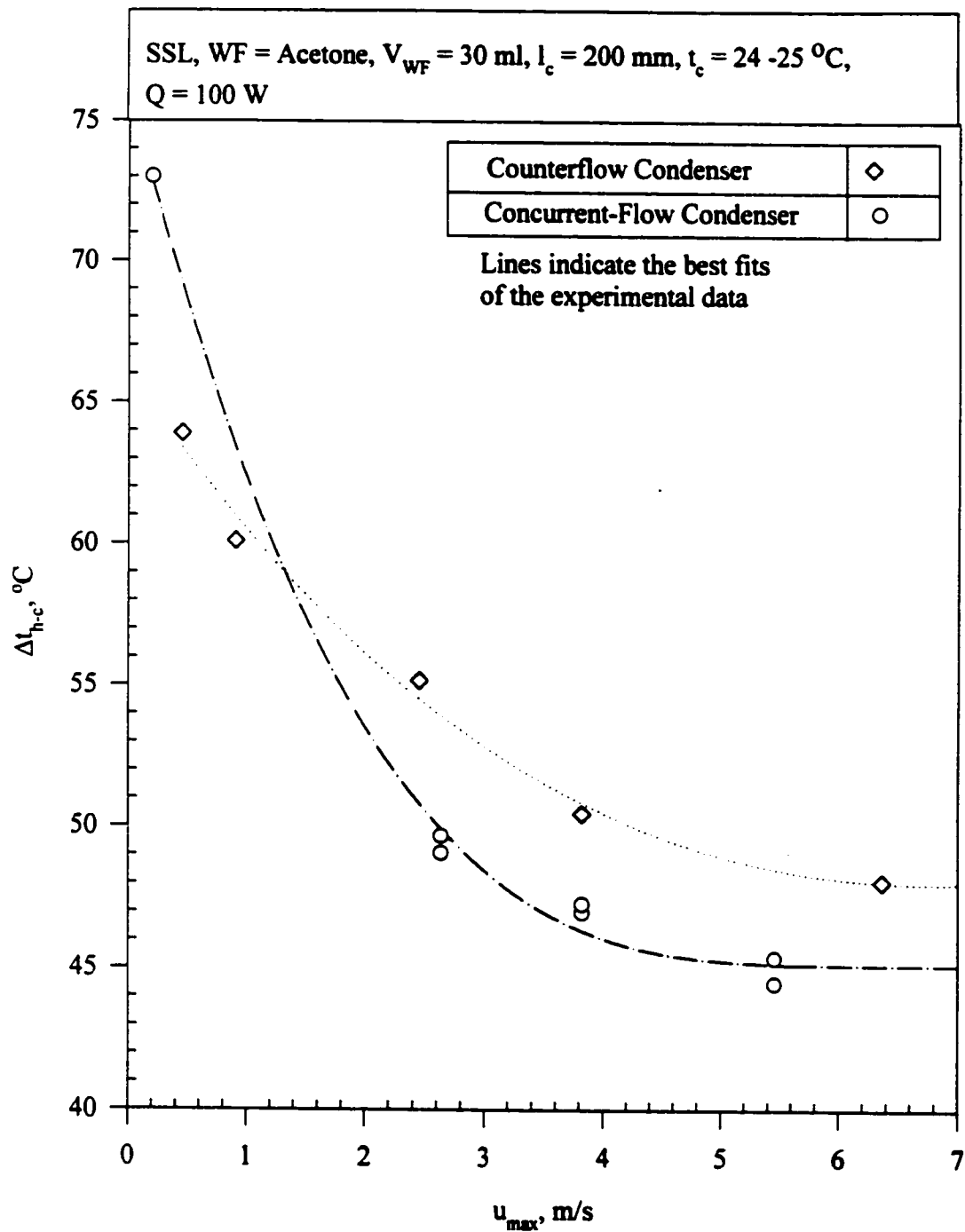


Figure 5.5 (a) Effect of u_{max} on Δt_{h-c} with Different Condenser Configuration; SSL

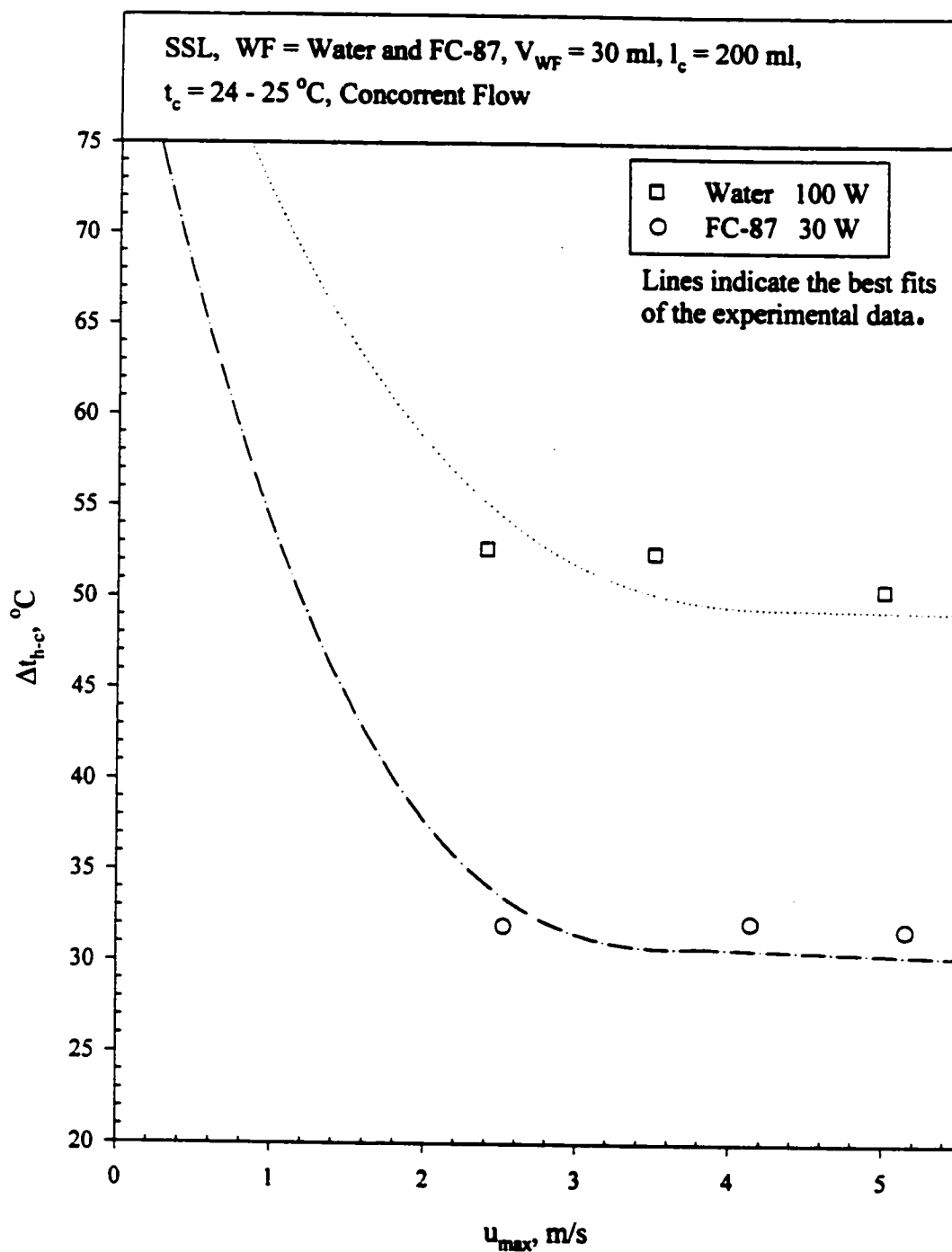


Figure 5.5 (b). Effect of u_{max} on Δt_{h-c} ; WF = Water, FC-87, Concurrent flow, SSL

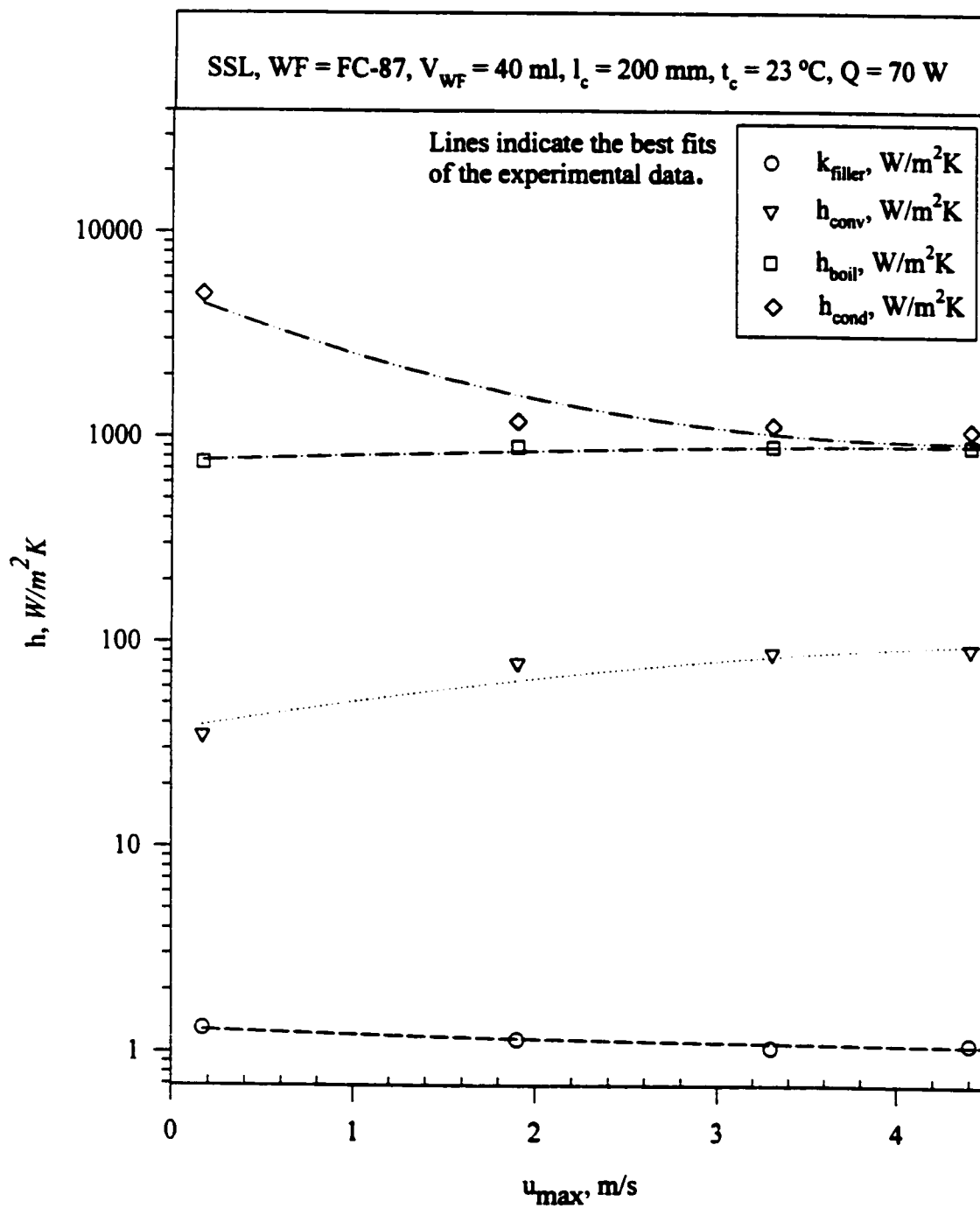


Figure 5.5 (c). Effect of Air Velocity on Heat Transfer Coefficients and Thermal Conductivity; SSL

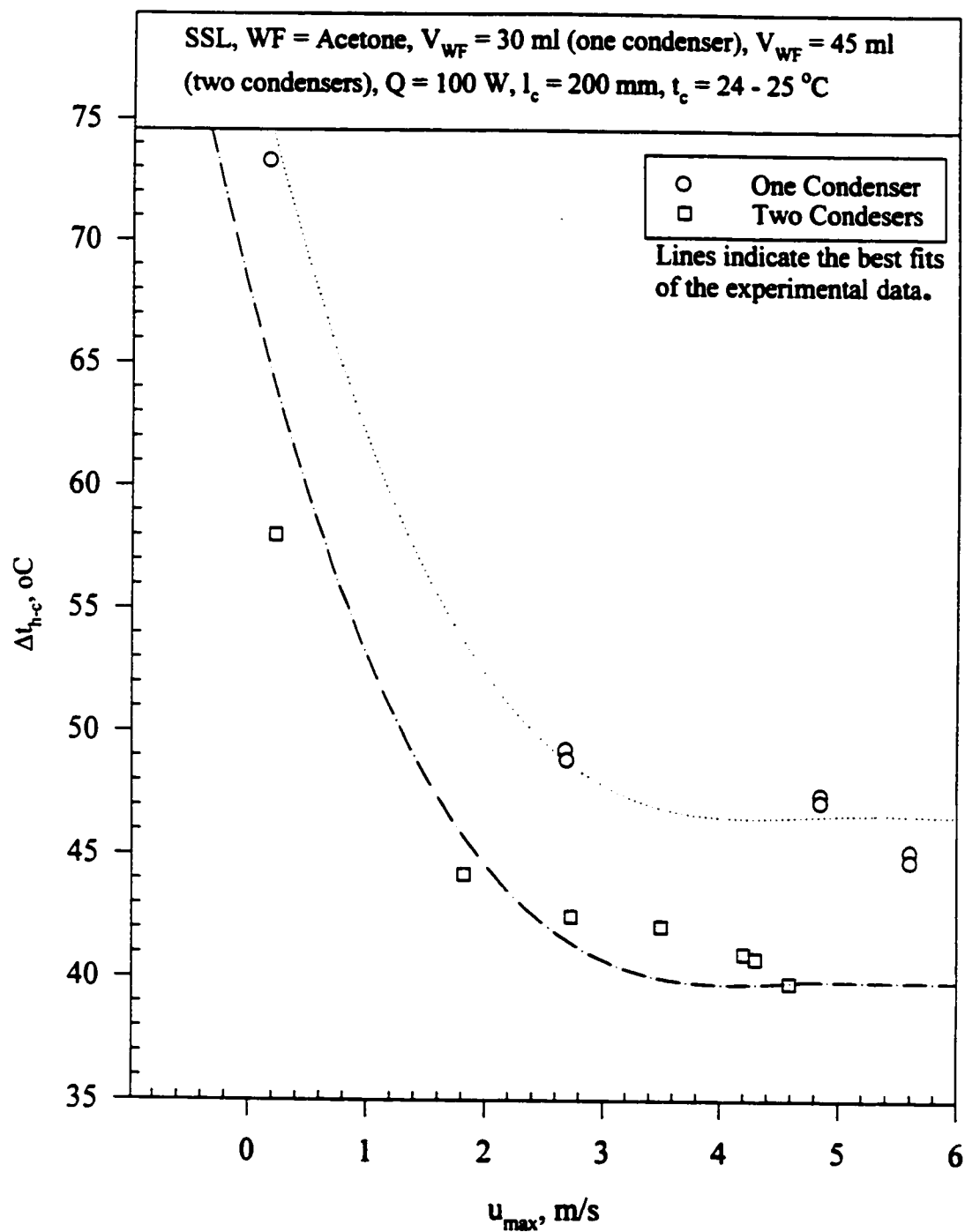


Figure 5.5 (d). Effect of u_{max} on Δt_{h-c} ;
 Comparison between One condenser and two condensers, SSL

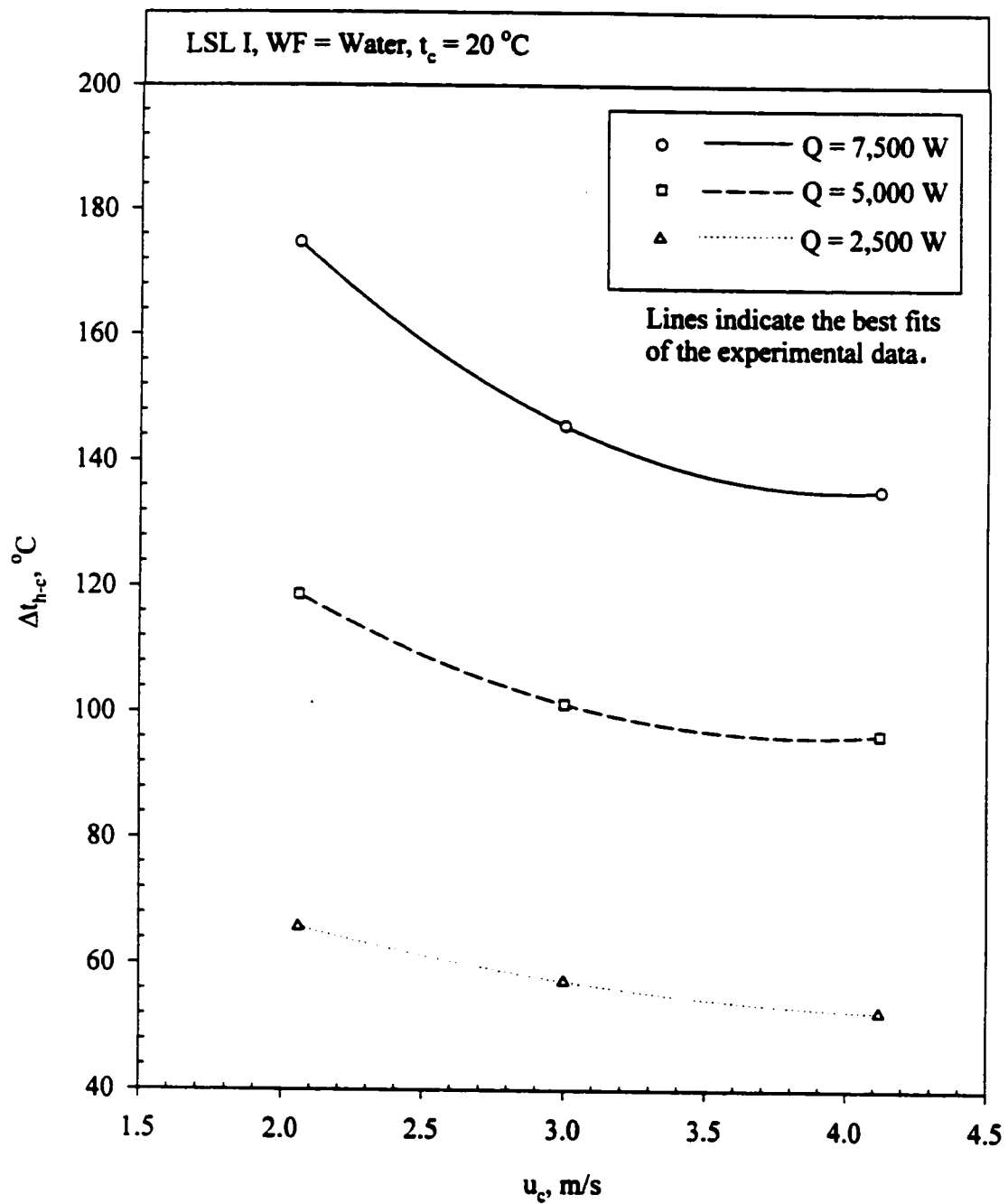


Figure 5.5 (e). Effect of u_c on Δt_{h-c} ; LSL I

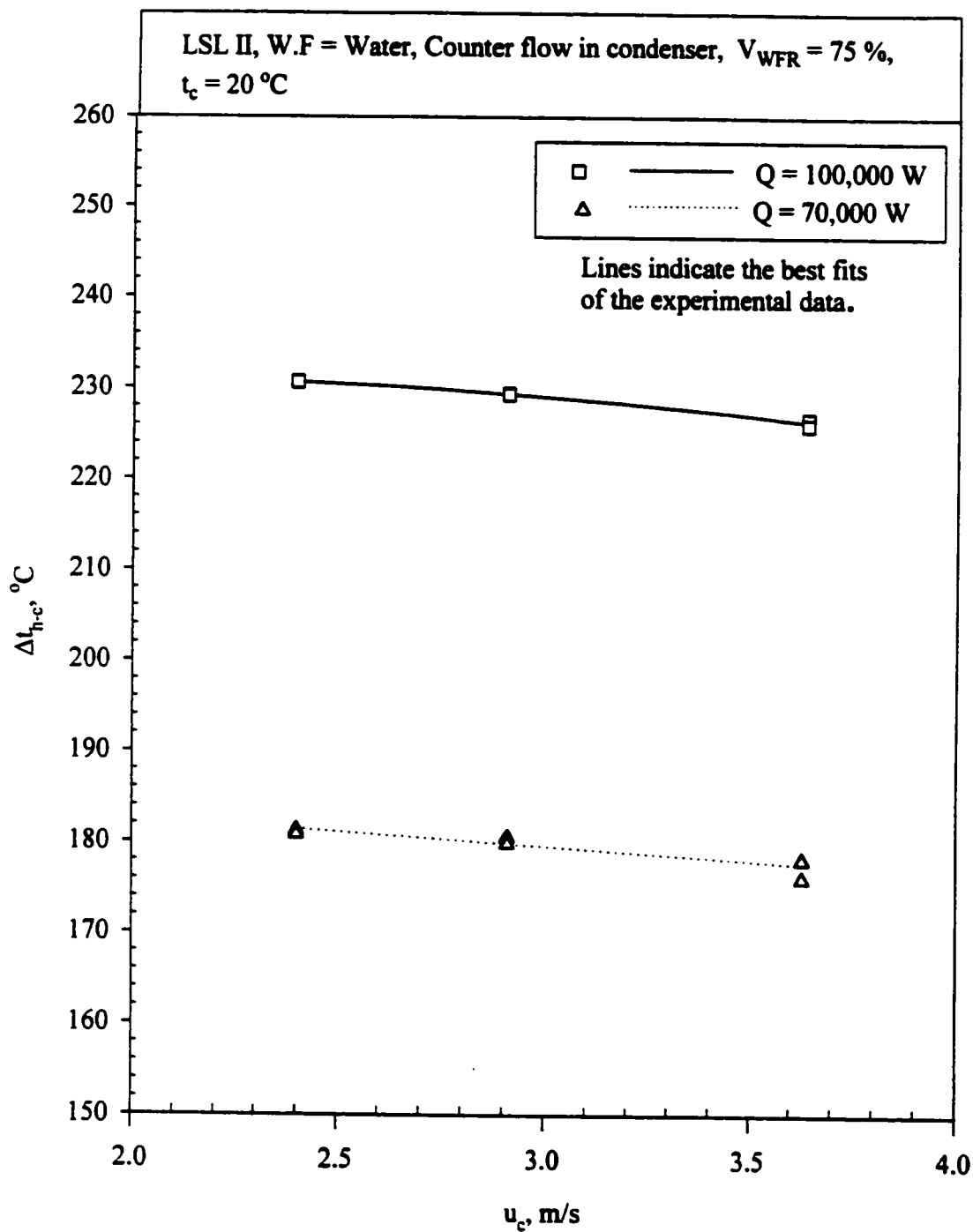


Figure 5.5 (f). Effect of u_c on Δt_{h-c} ; LSL II

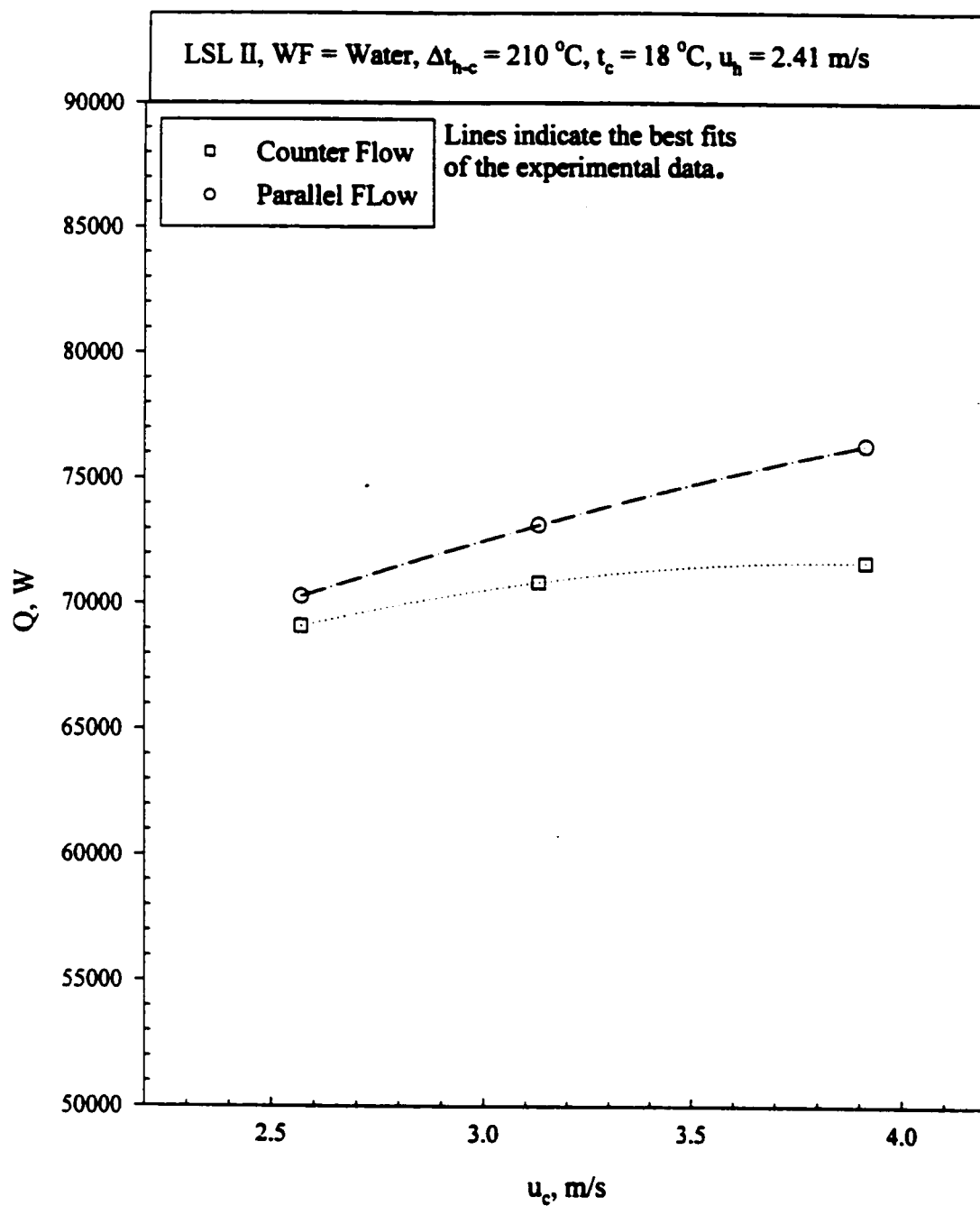


Figure 5.5 (g). Effect of u_c on Q ; LSL II

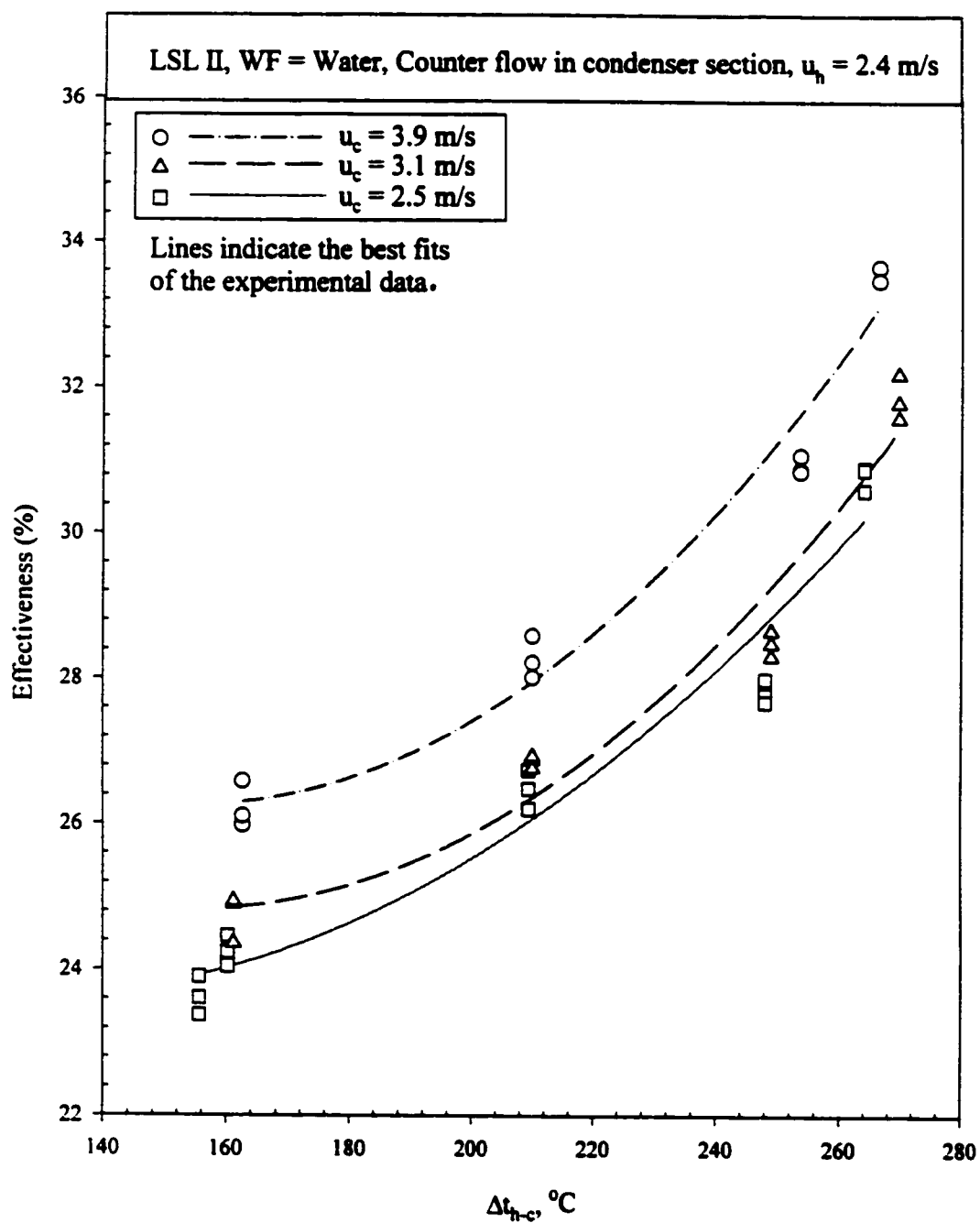


Figure 5.6 (a) Effect of Δt_{h-c} on Effectiveness with different u_c ; LSL II with Counter flow

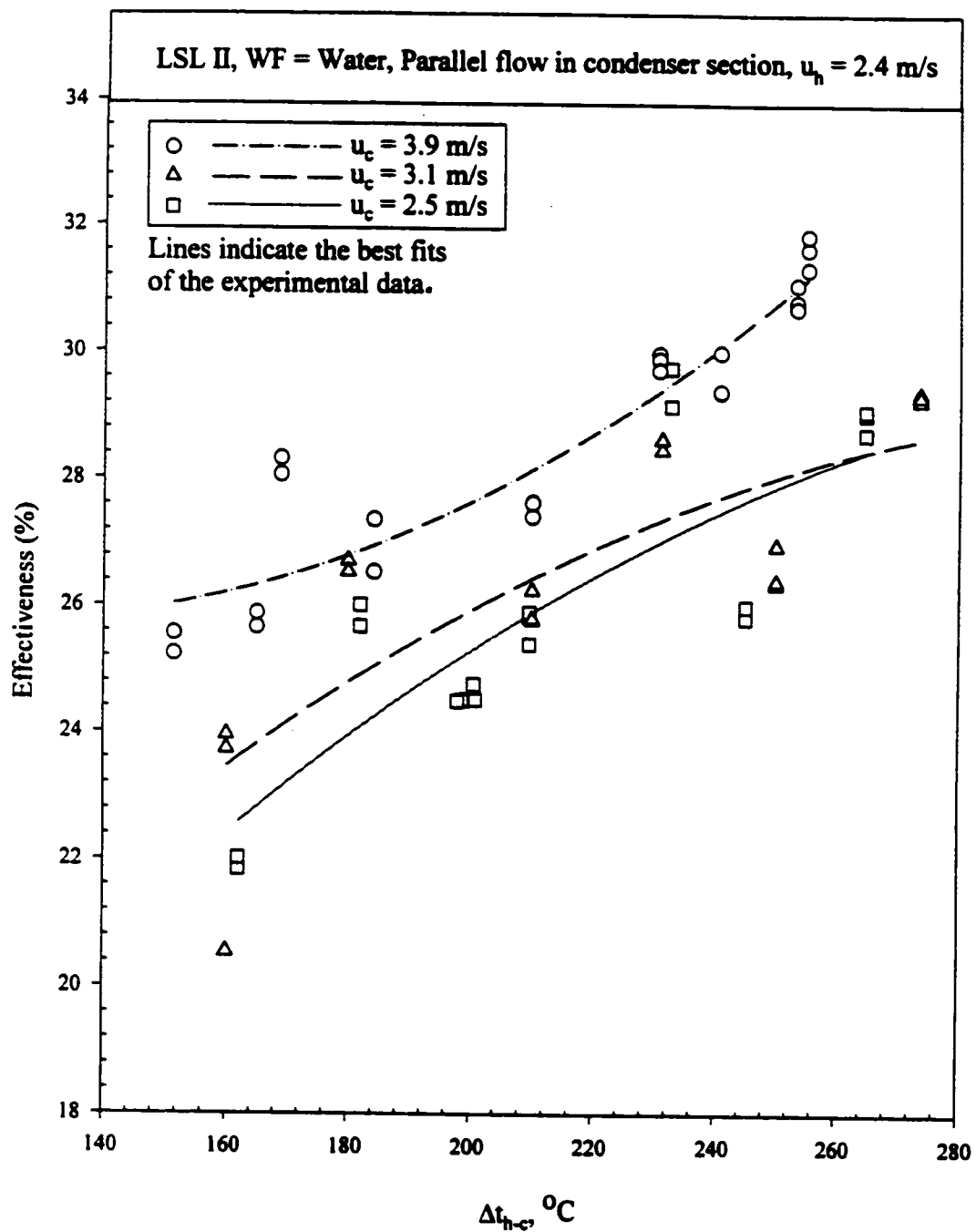


Figure 5.6 (b). Effect of Δt_{h-c} on Effectiveness with different u_c ; LSL II with Parallel flow

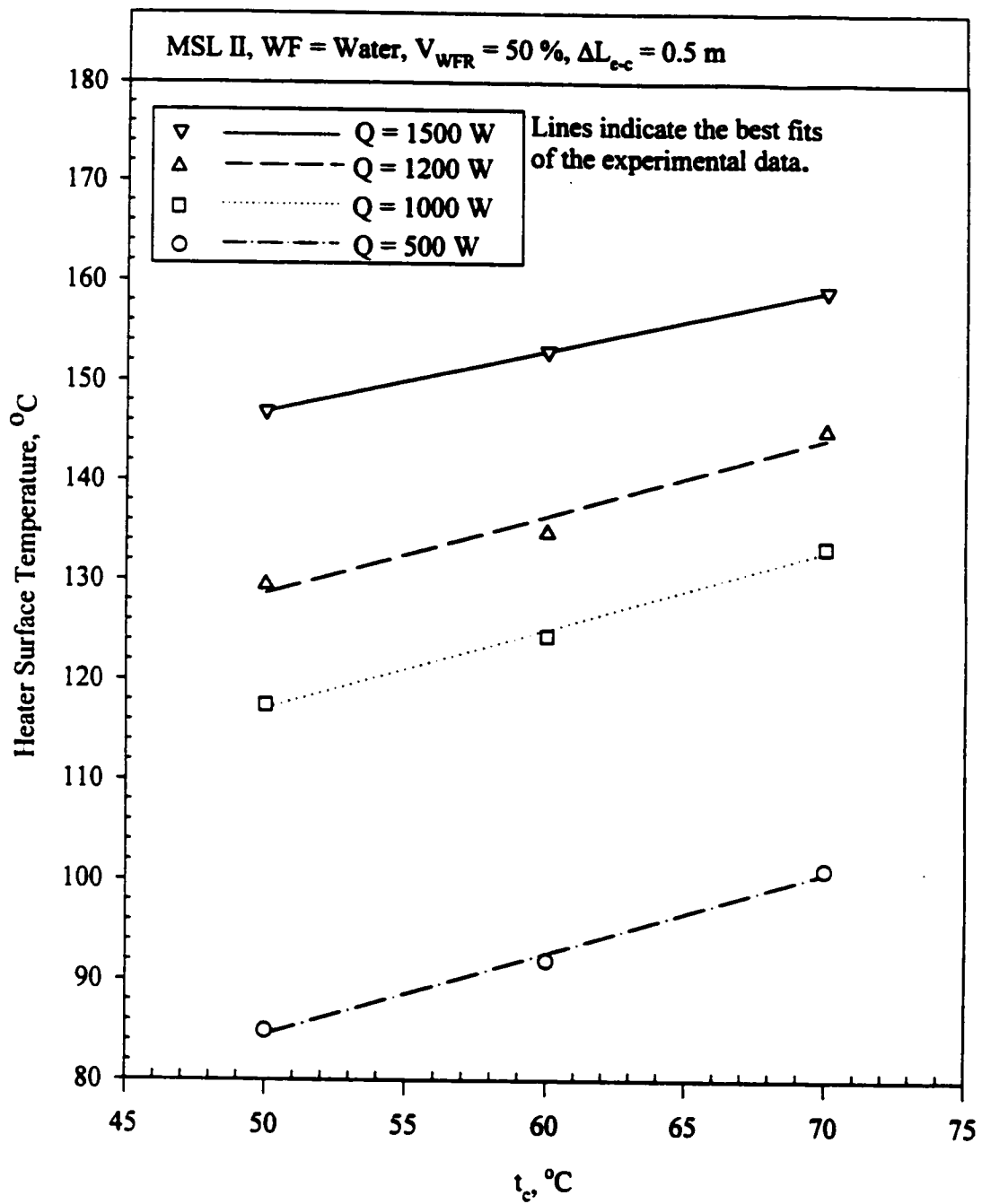


Figure 5.7 (a). Effect of t_c on Evaporator Surface Temperature ; MSL II
Coolant : Water

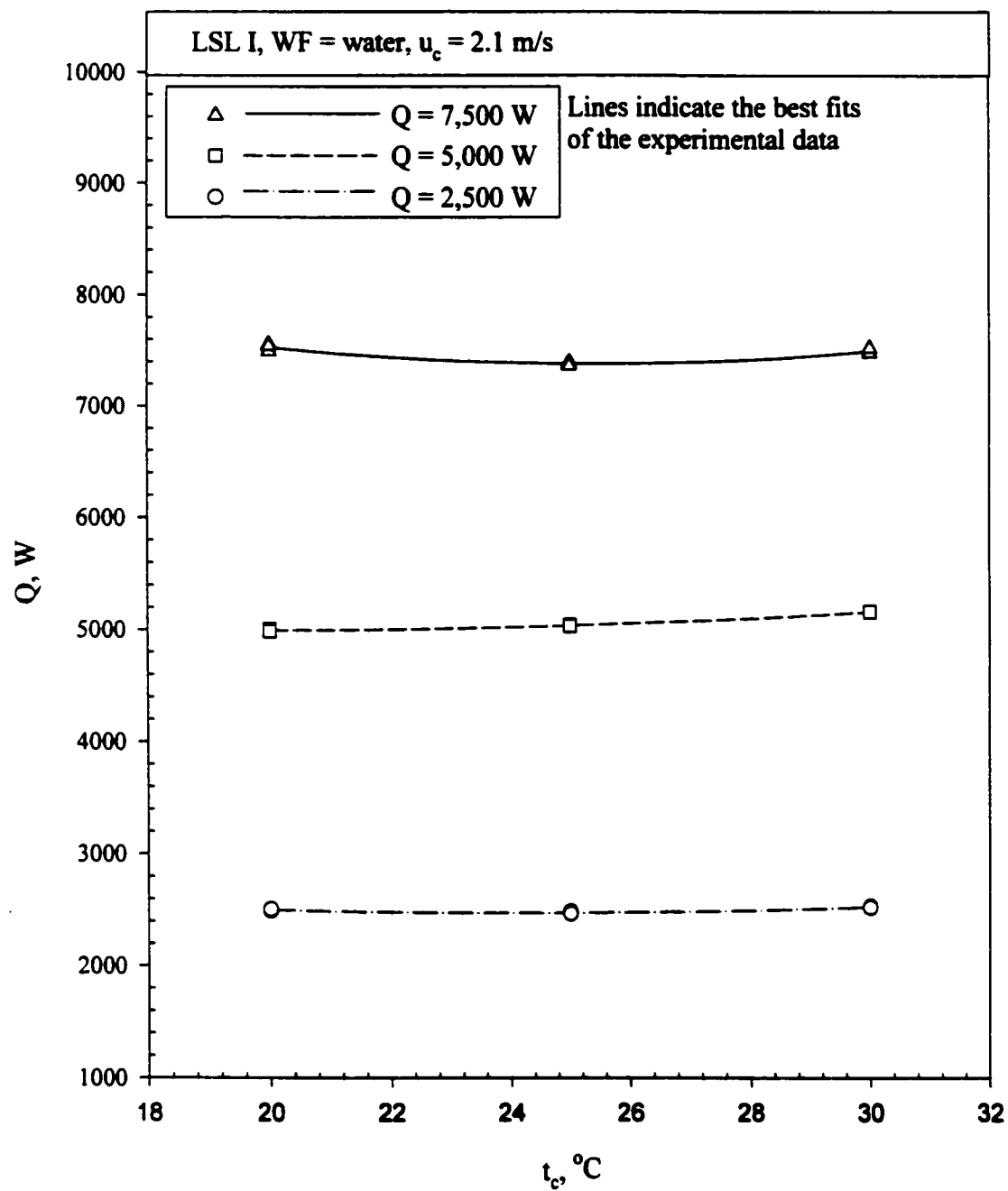


Figure 5.7 (b). Effect of t_c on Q ; LSL I
 Cooling : Air

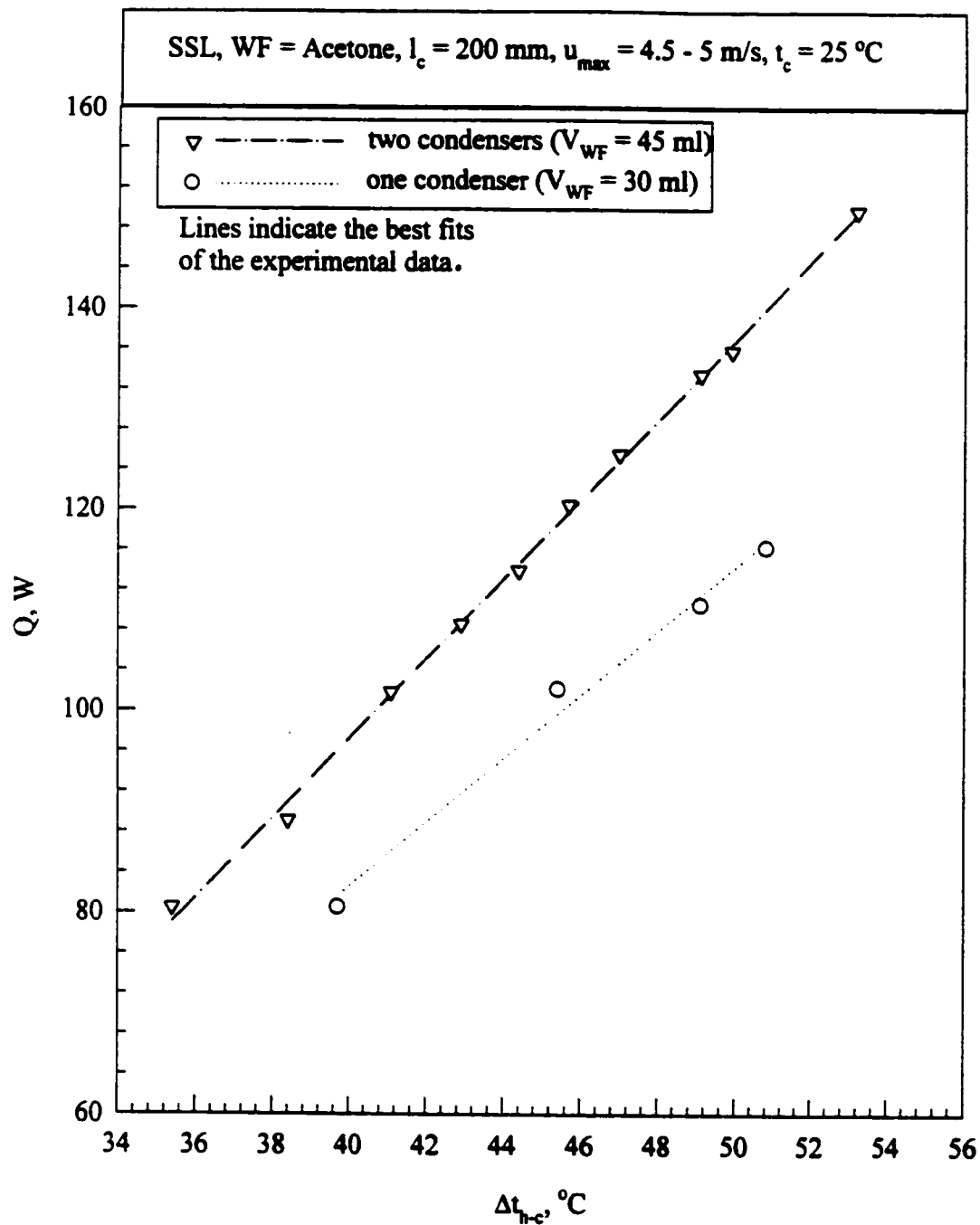


Figure 5.8. Effect of Number of Condensers on Q vs Δt_{h-c} ; SSL

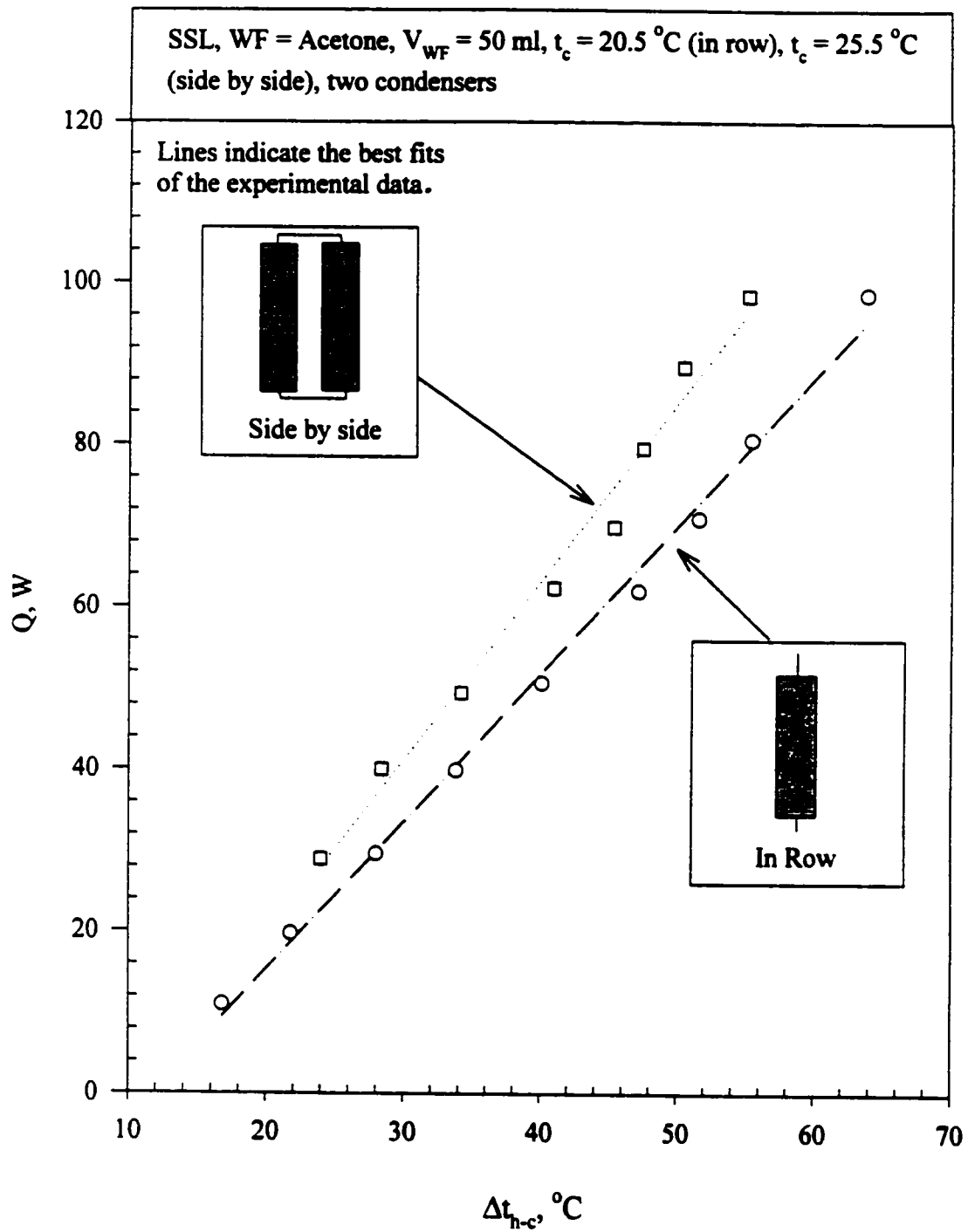


Figure 5.9. Effect of Q vs. Δt_{h-c} with Different Condenser Arrangement ; SSL

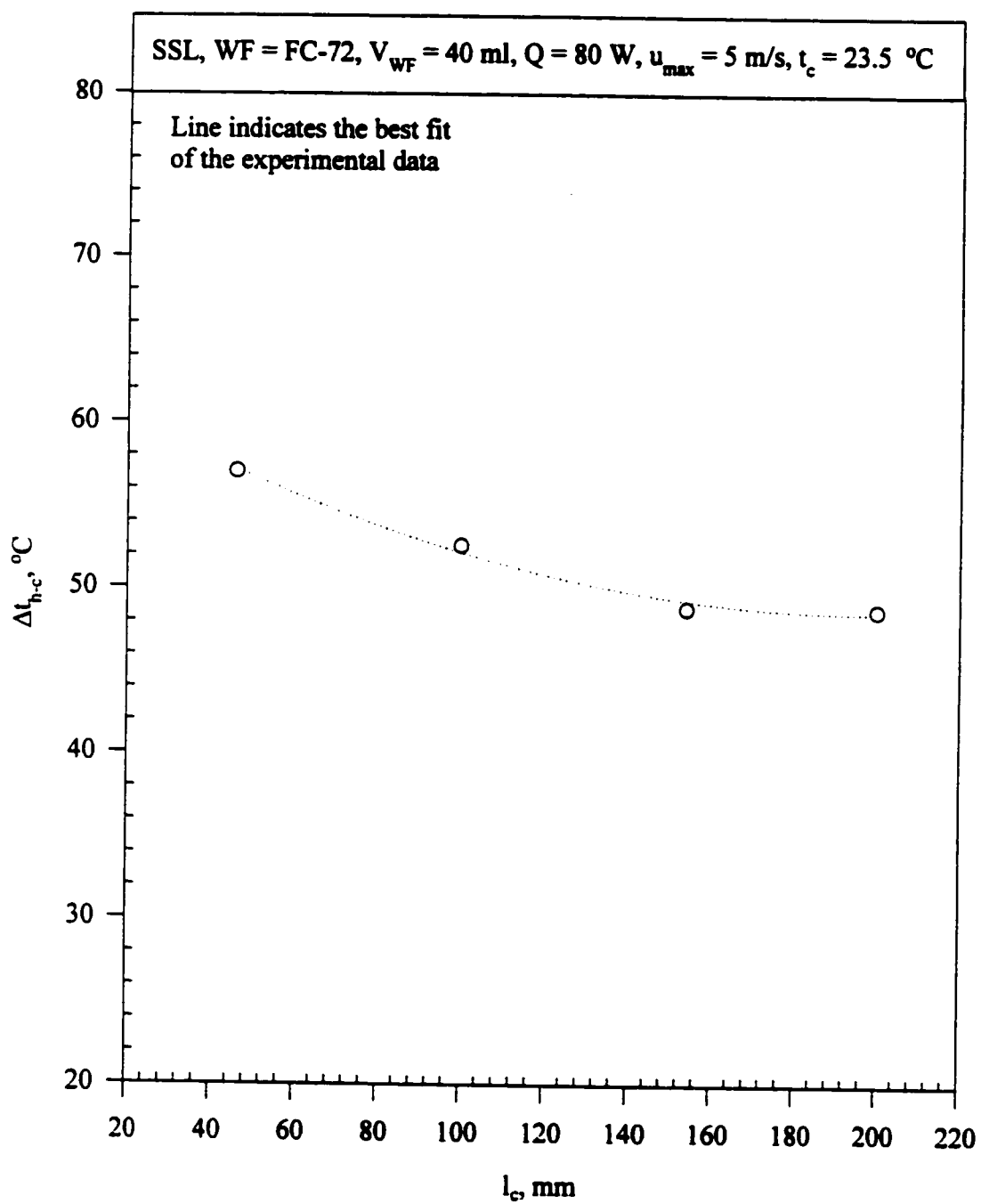


Figure 5.10. Effect of l_c on Δt_{h-c} ; SSL

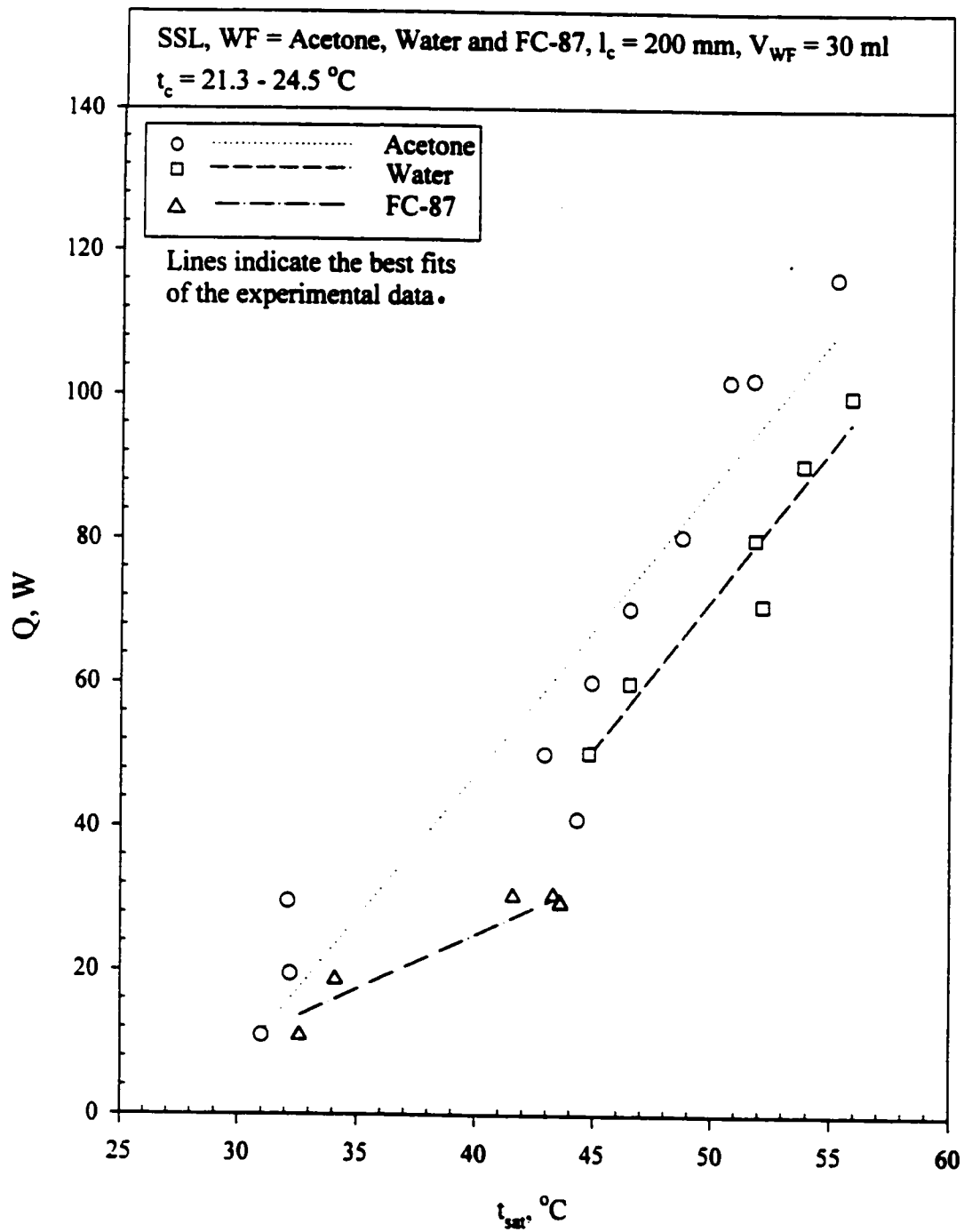


Figure 5.11 (a). Effect of t_{sat} on Q ; SSL

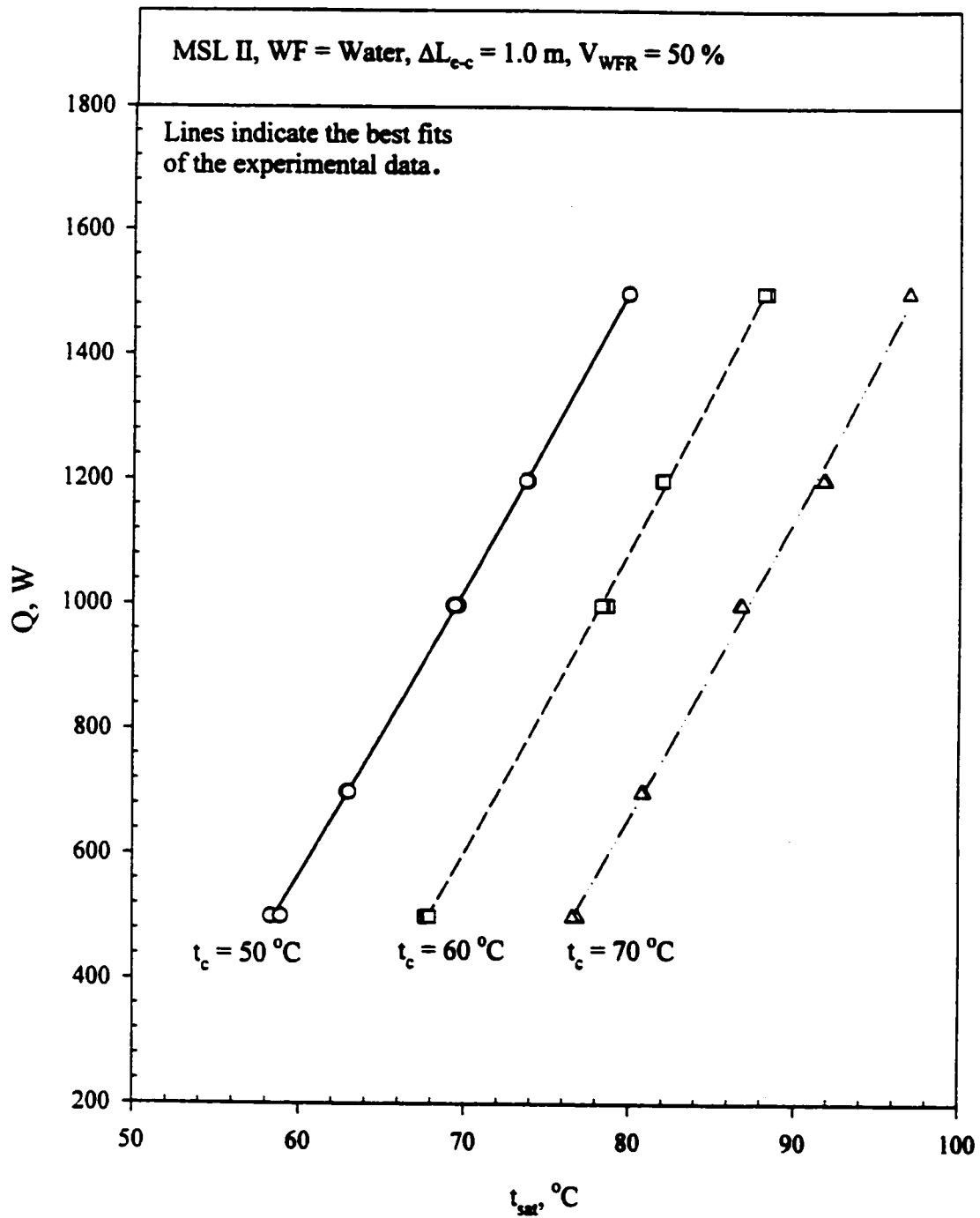


Figure 5.11 (b). Effect of t_{sst} on Q ; MSL II

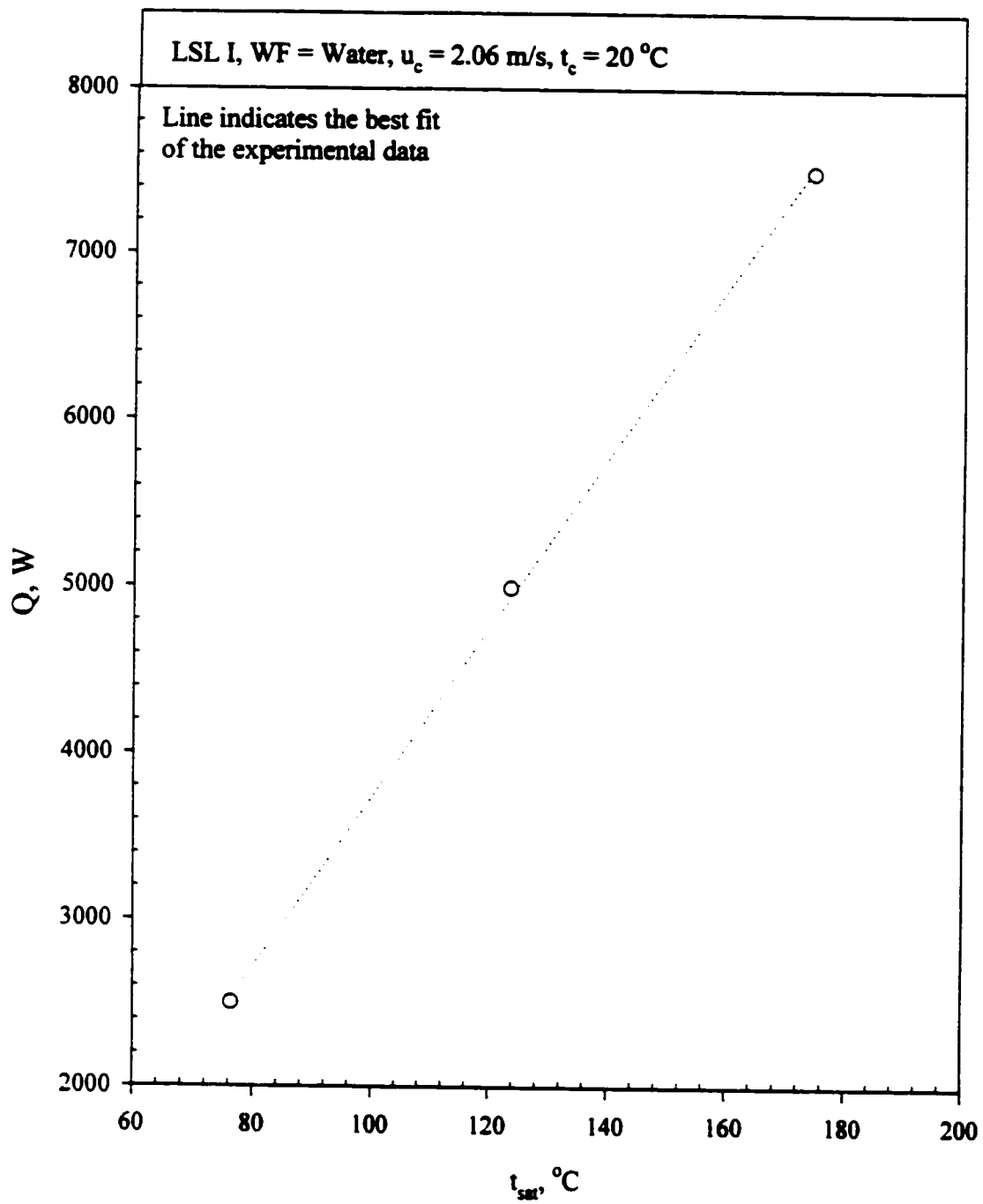


Figure 5.11 (c). Effect of t_{sm} on Q ; LSL I

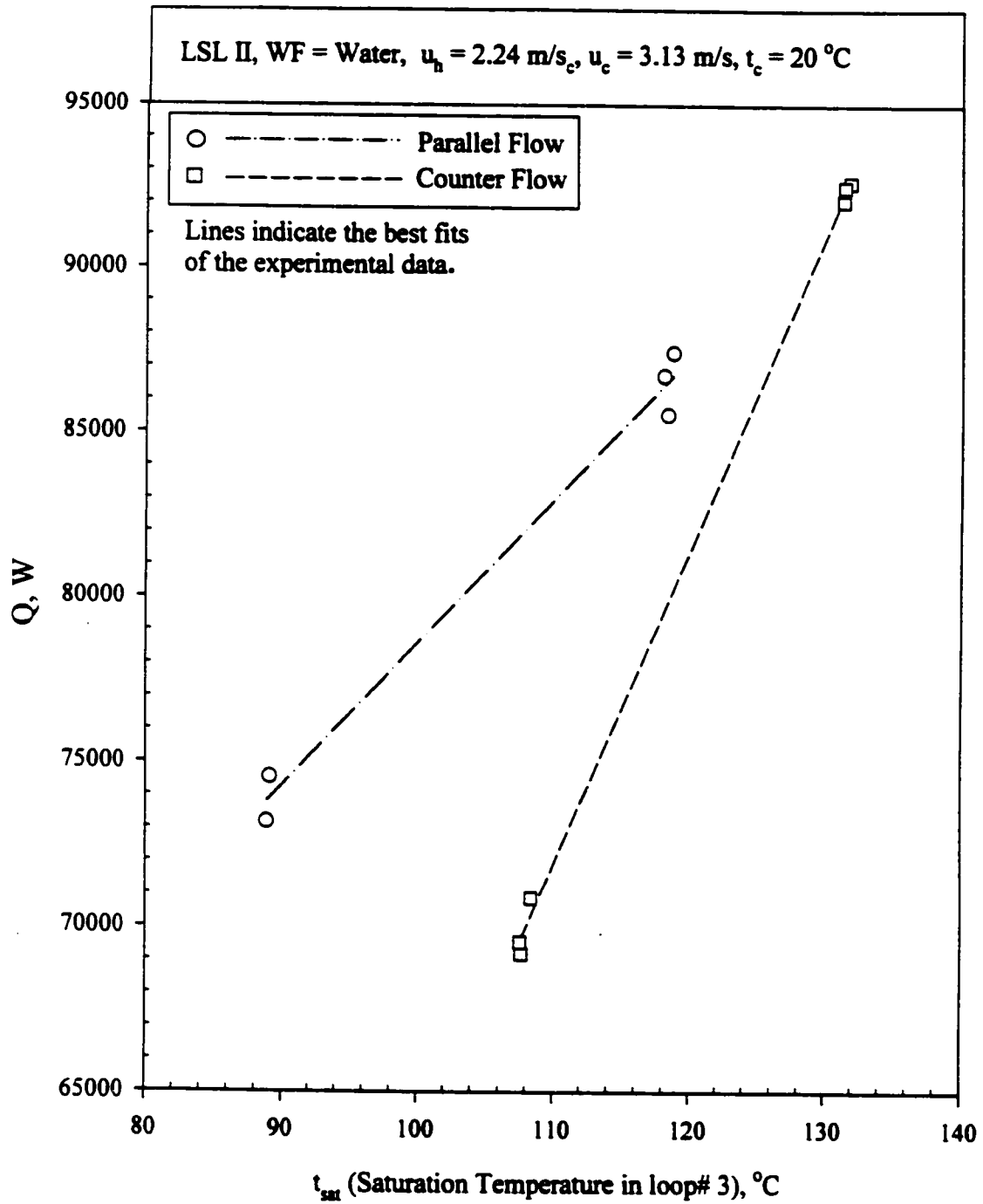


Figure 5.11 (d). Effect of t_{sat} on Q ; LSL II

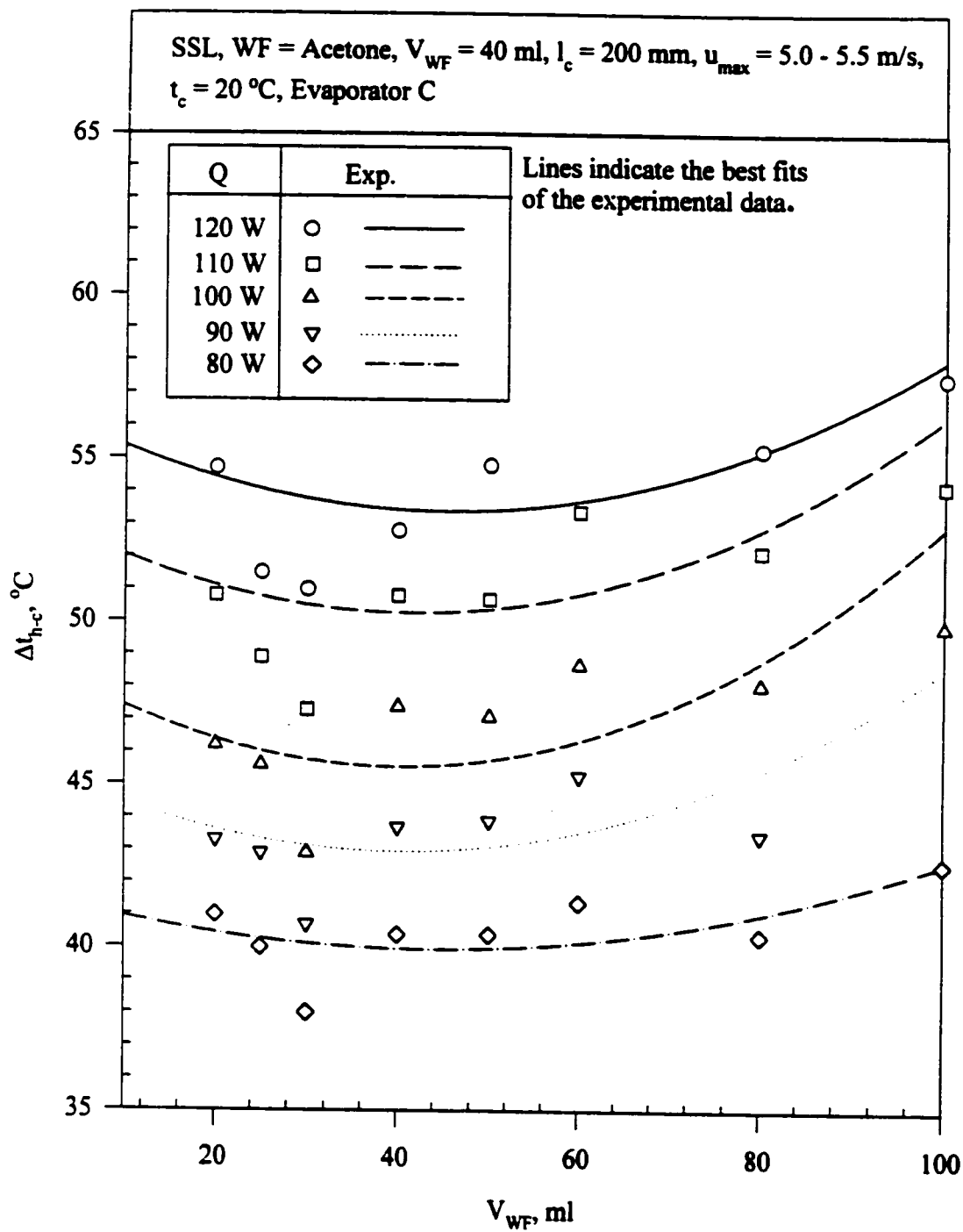


Figure 5.12 (a). Effect of V_{WF} on Δt_{h-c} ; Forced Convection Mode ; SSL

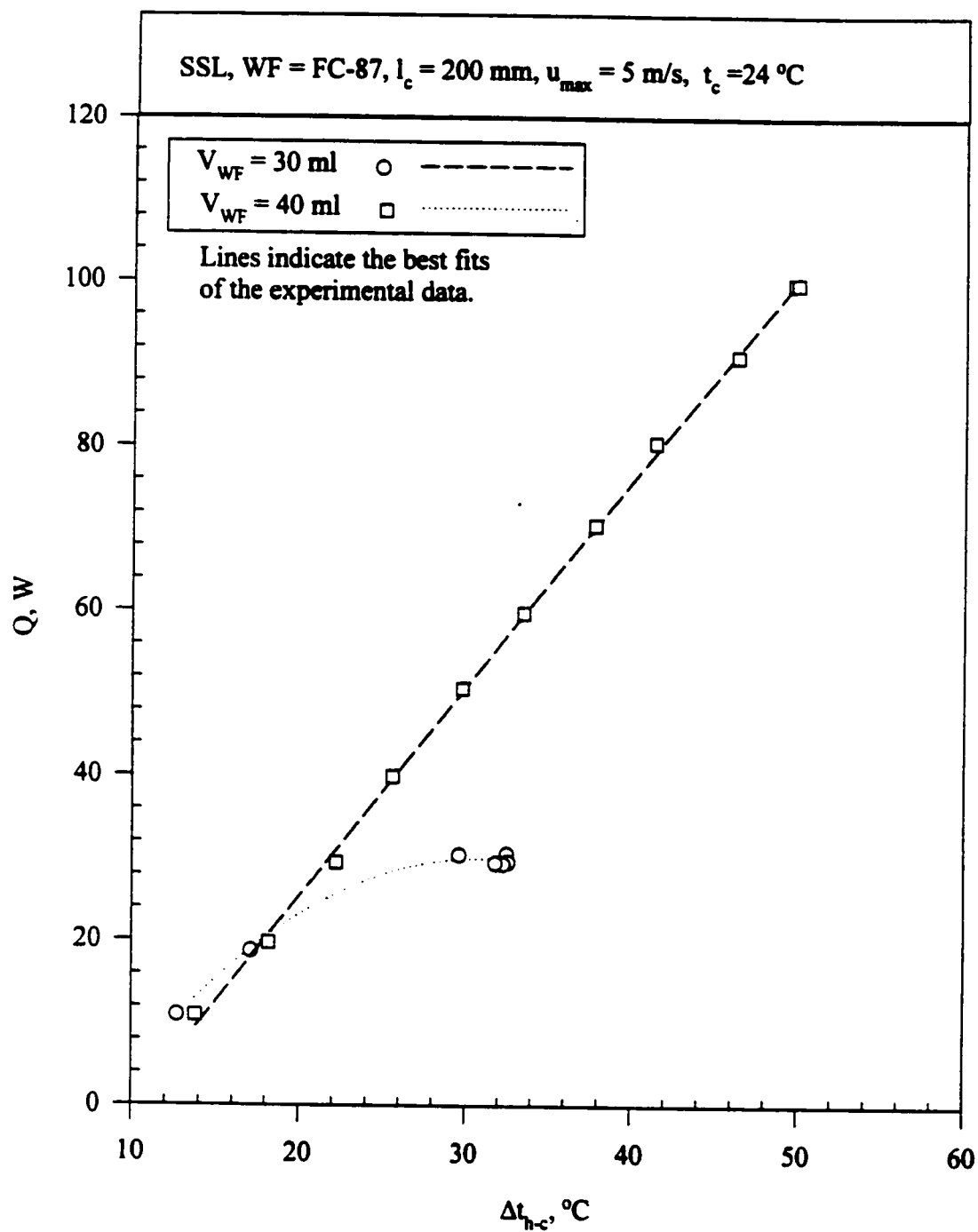


Figure 5.12 (b). Effect of V_{WF} on Δt_{h-c} vs. Q ; SSL

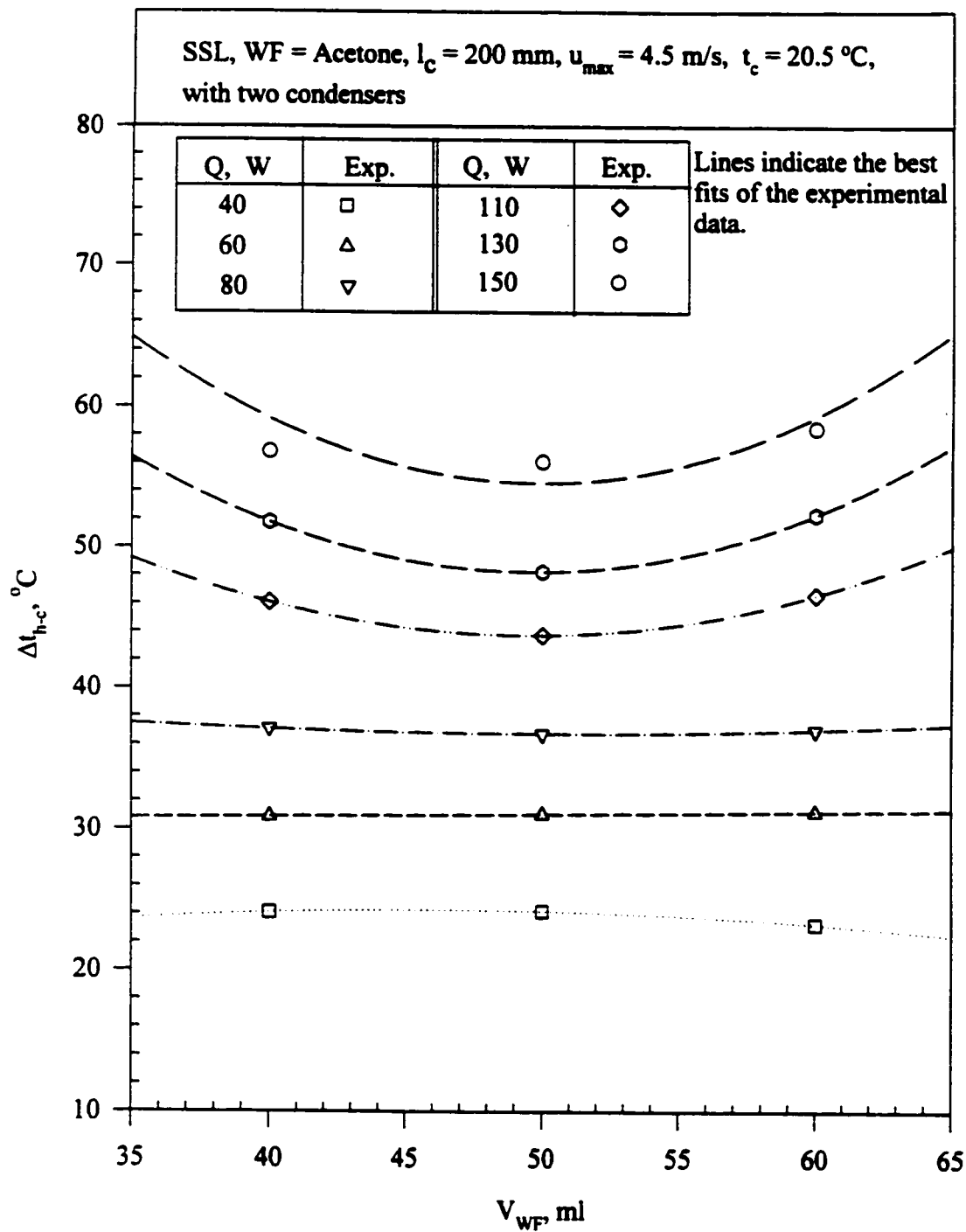


Figure 5.12 (c) Effect of V_{WF} on Δt_{h-c} ; SSL

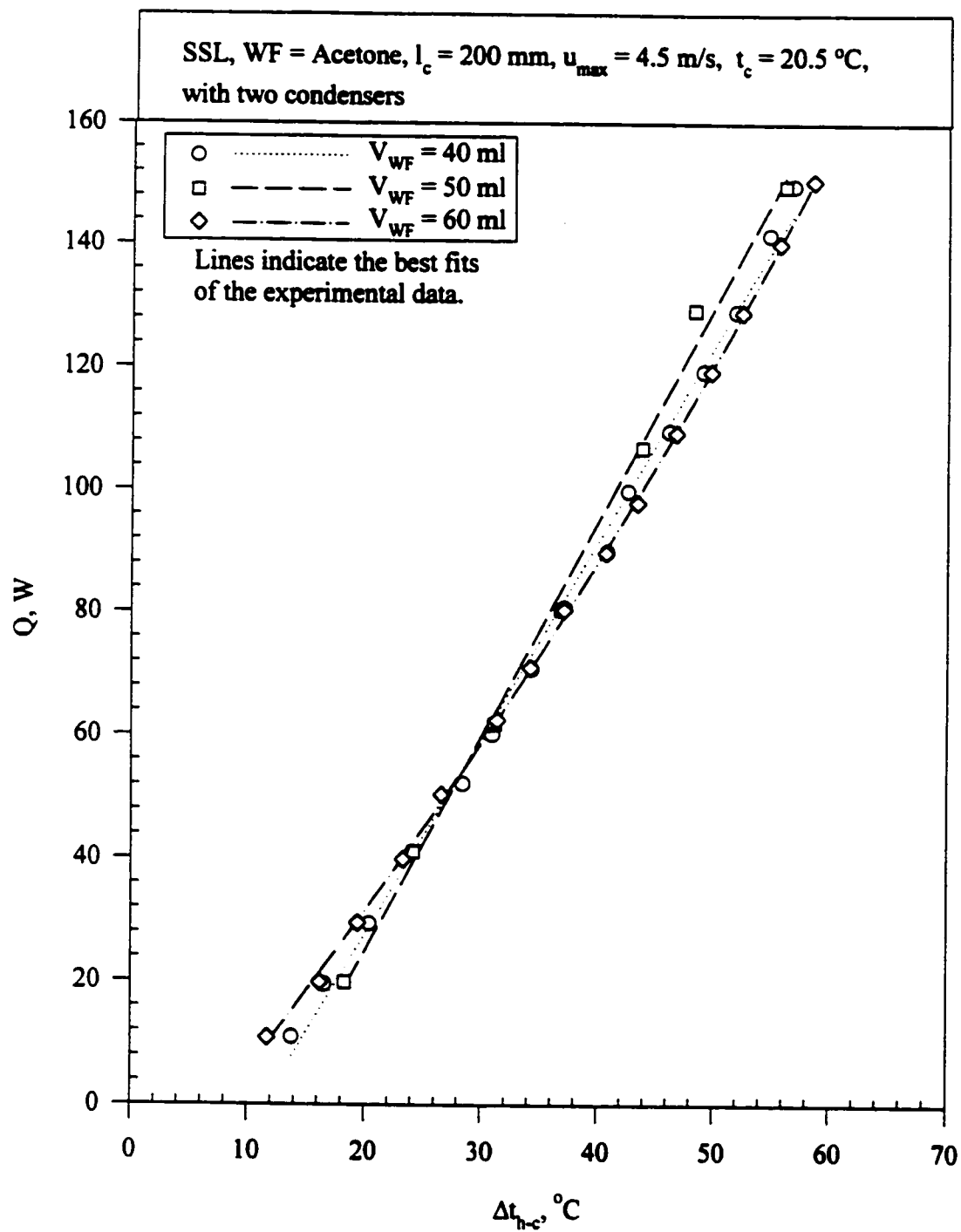


Figure 5.12 (d). Effect of V_{WF} on Δt_{h-c} vs. Q ; SSL

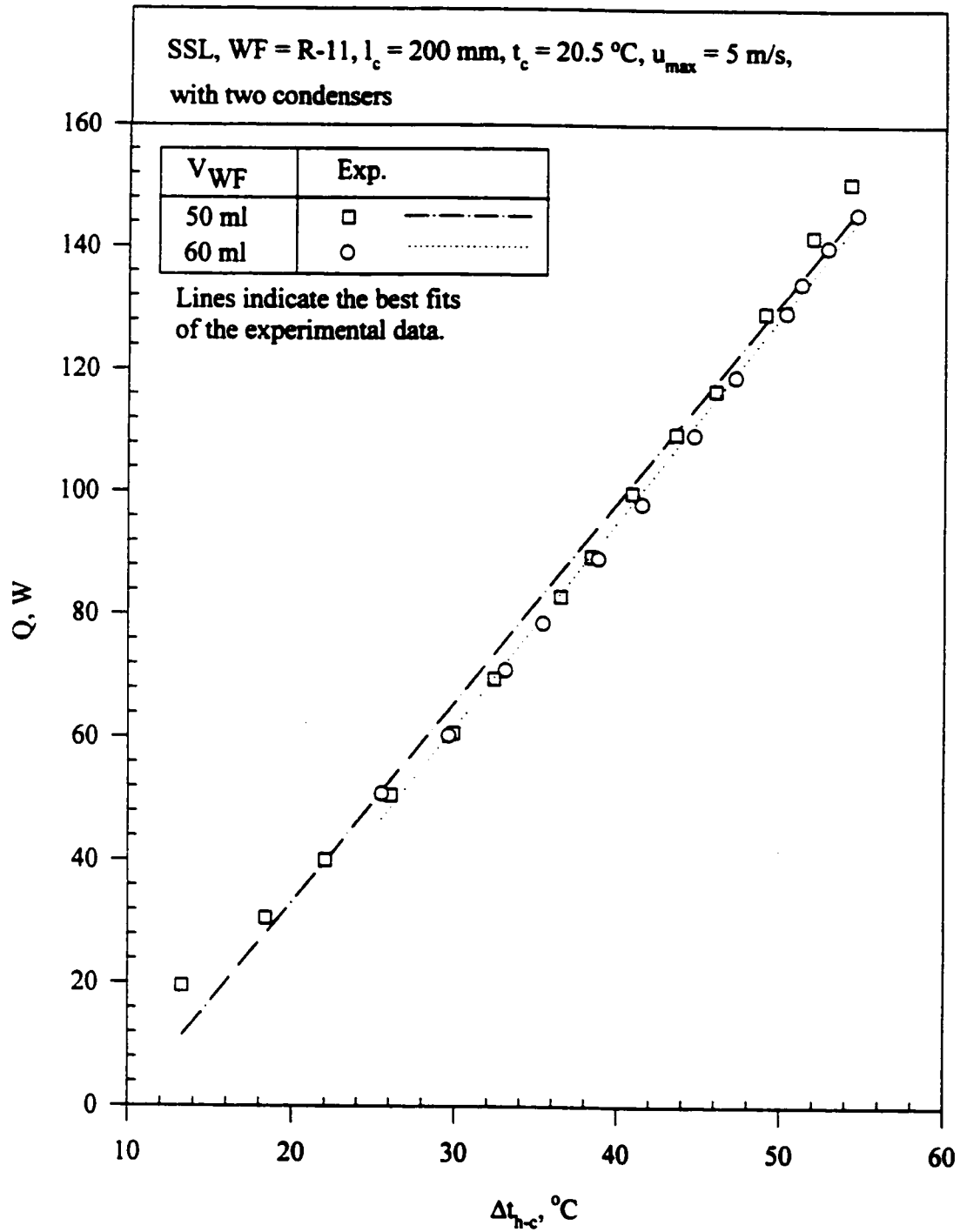


Figure 5.12 (e). Effect of V_{WF} on Δt_{h-c} vs. Q ; SSL

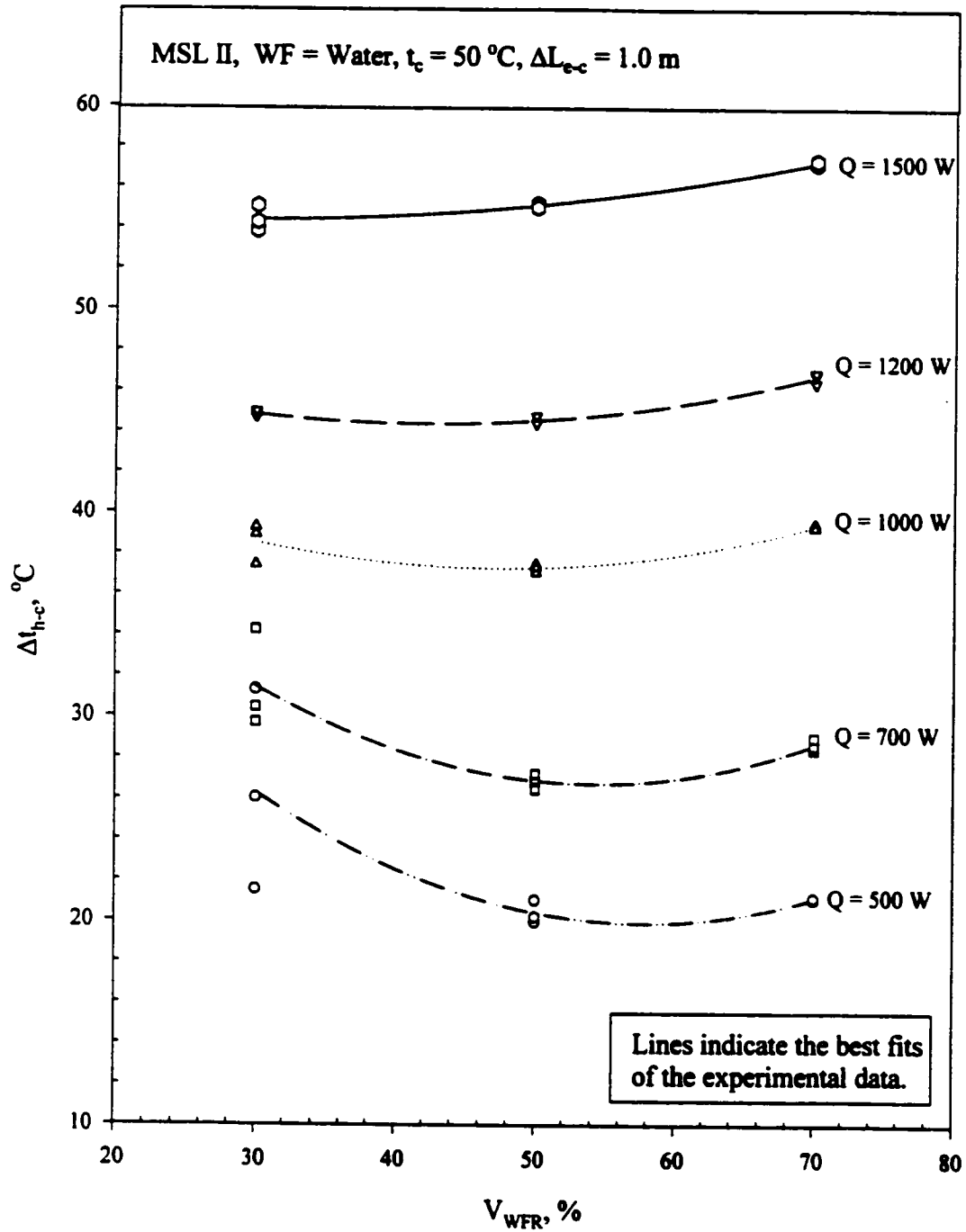


Figure 5.12 (f). Effect of V_{WFR} on Δt_{h-c} ; MSL II

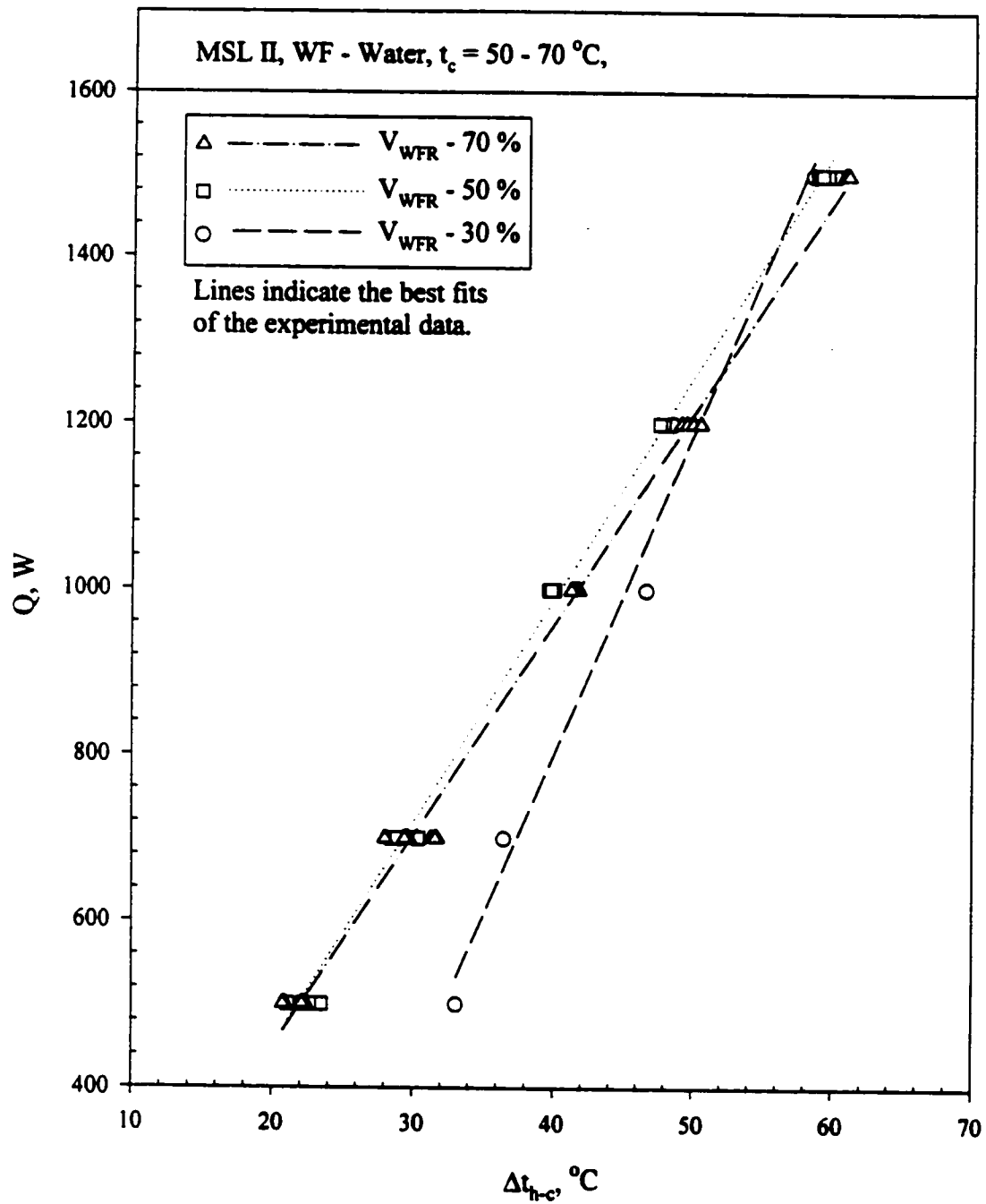


Figure 5.12 (g). Effect of V_{WFR} on Δt_{h-c} vs. Q ; MSL II

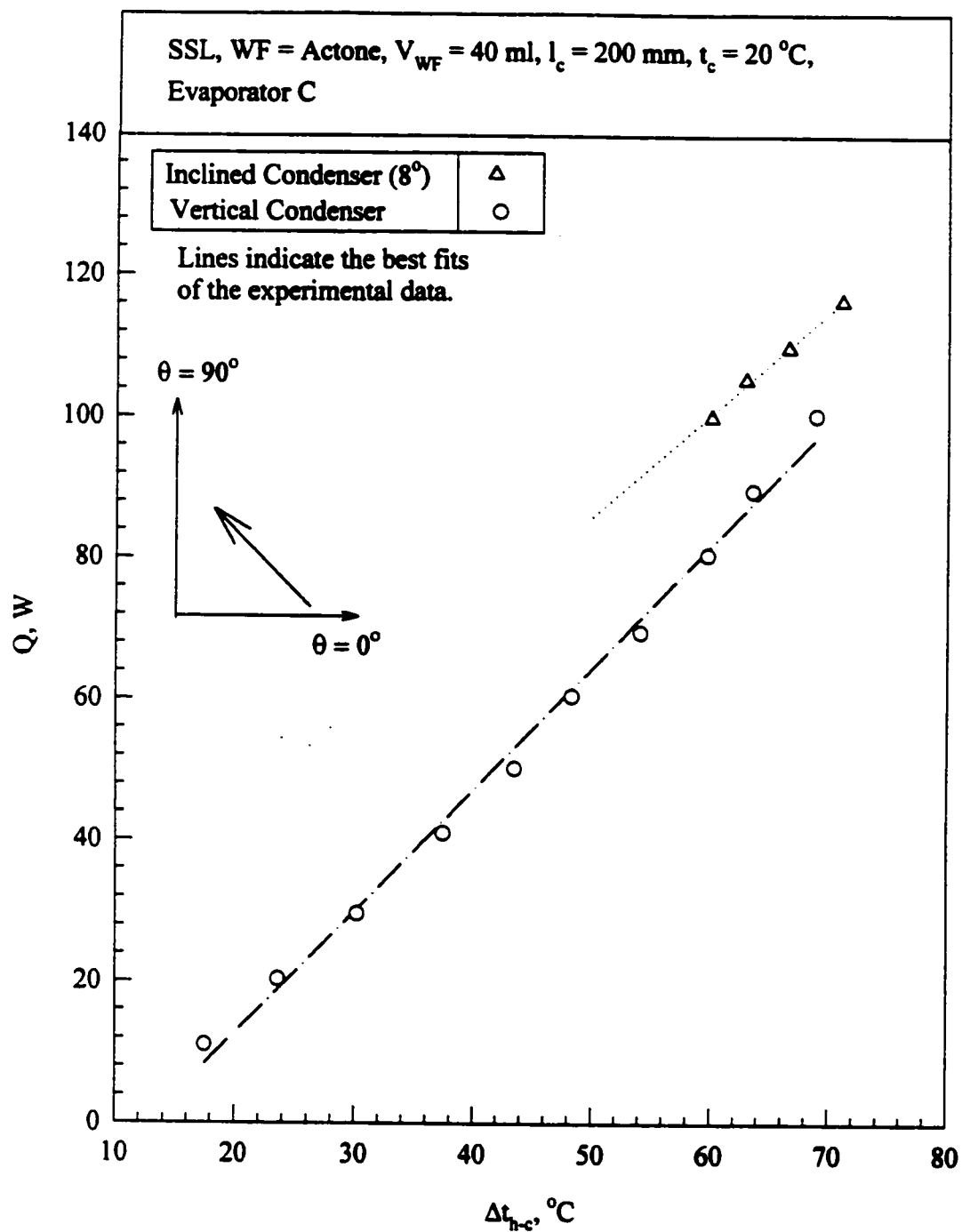


Figure 5.13 (a). Effect of Condenser Orientation; Natural Convection Mode, SSL

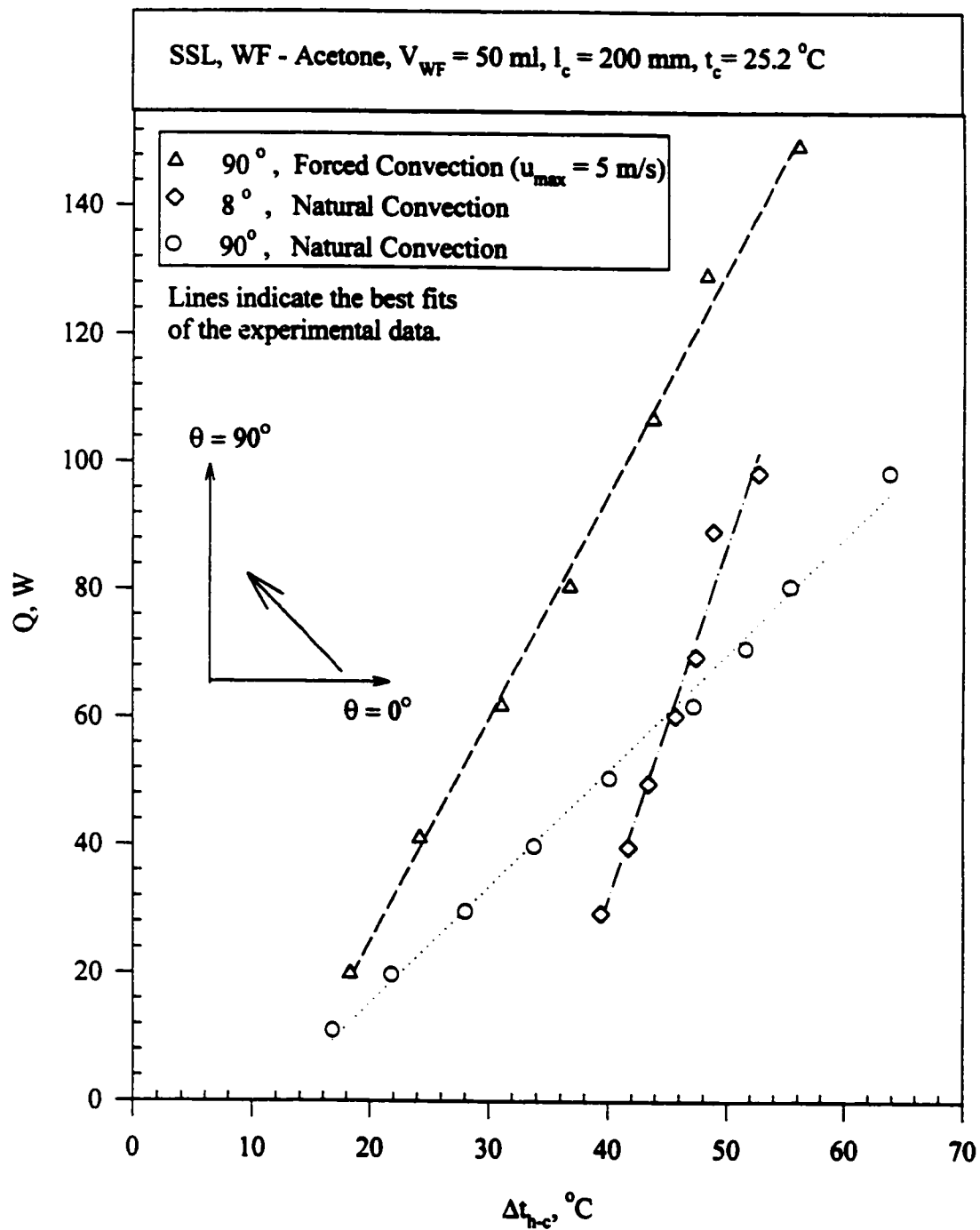


Figure 5.13 (b). Effect of Inclination of Condensers ; SSL

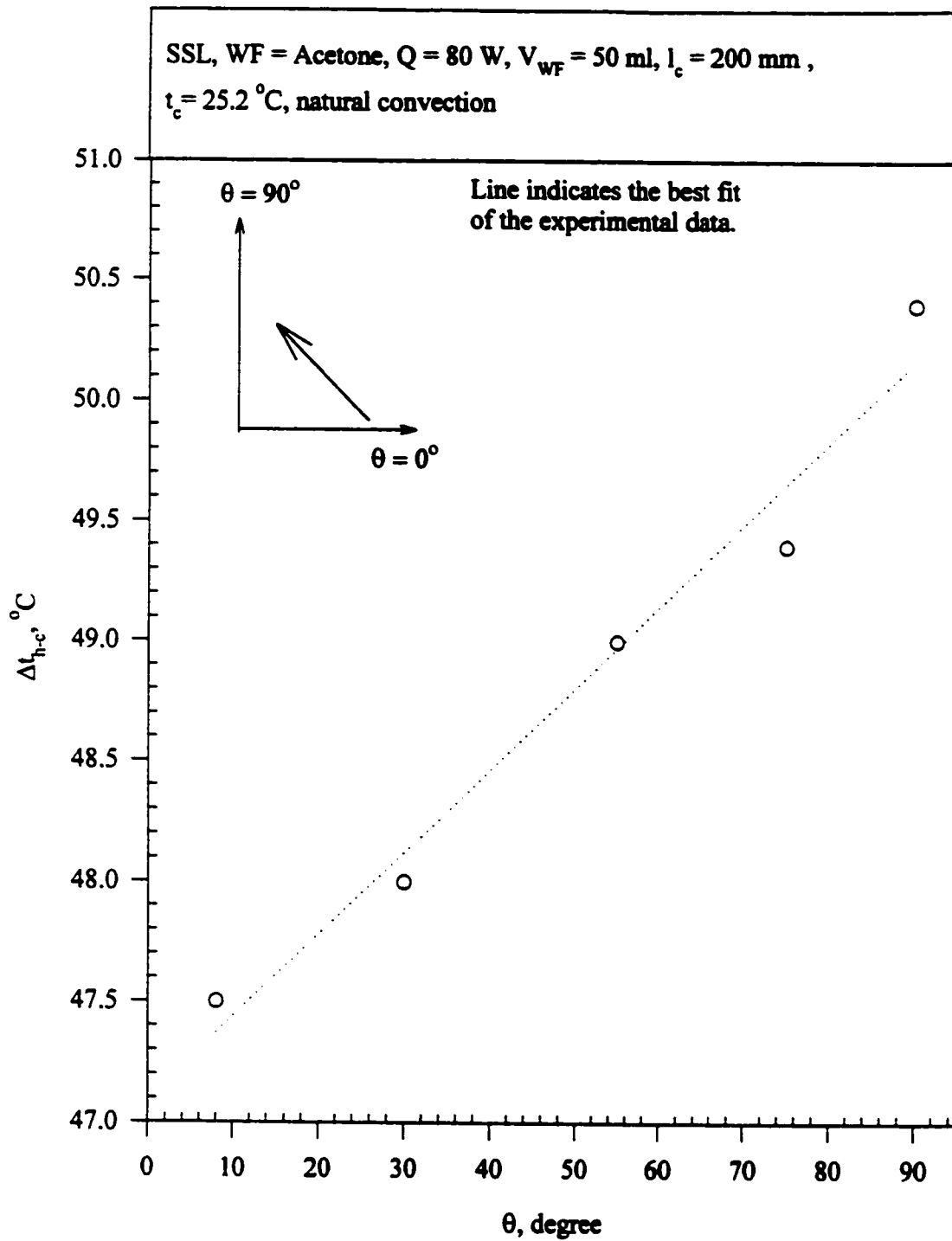


Figure 5.13 (c). Effect of Inclination of Condensers; SSL

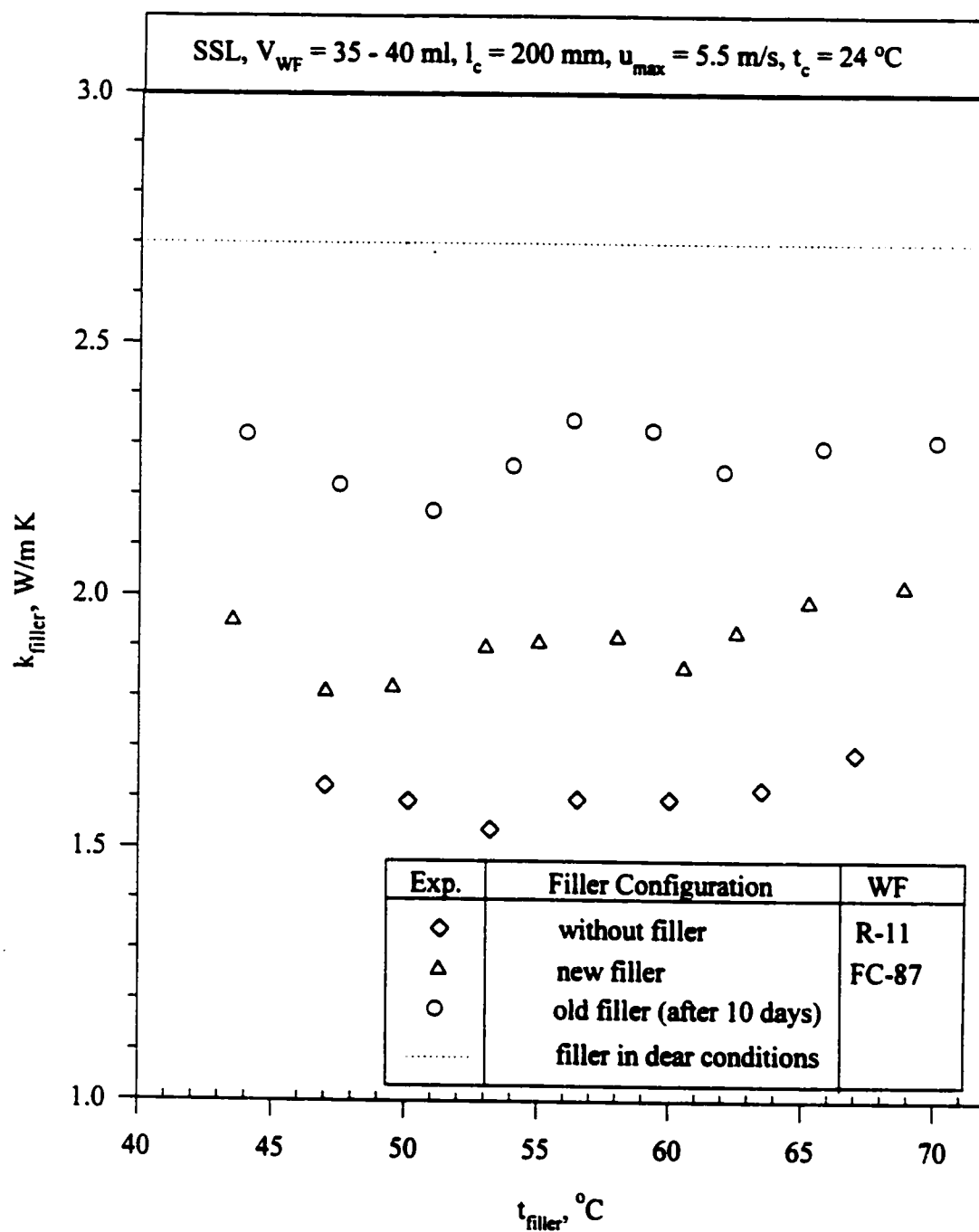


Figure 5.14. Contact Resistance between Evaporator and Heater with the same Applied Pressure ; SSL

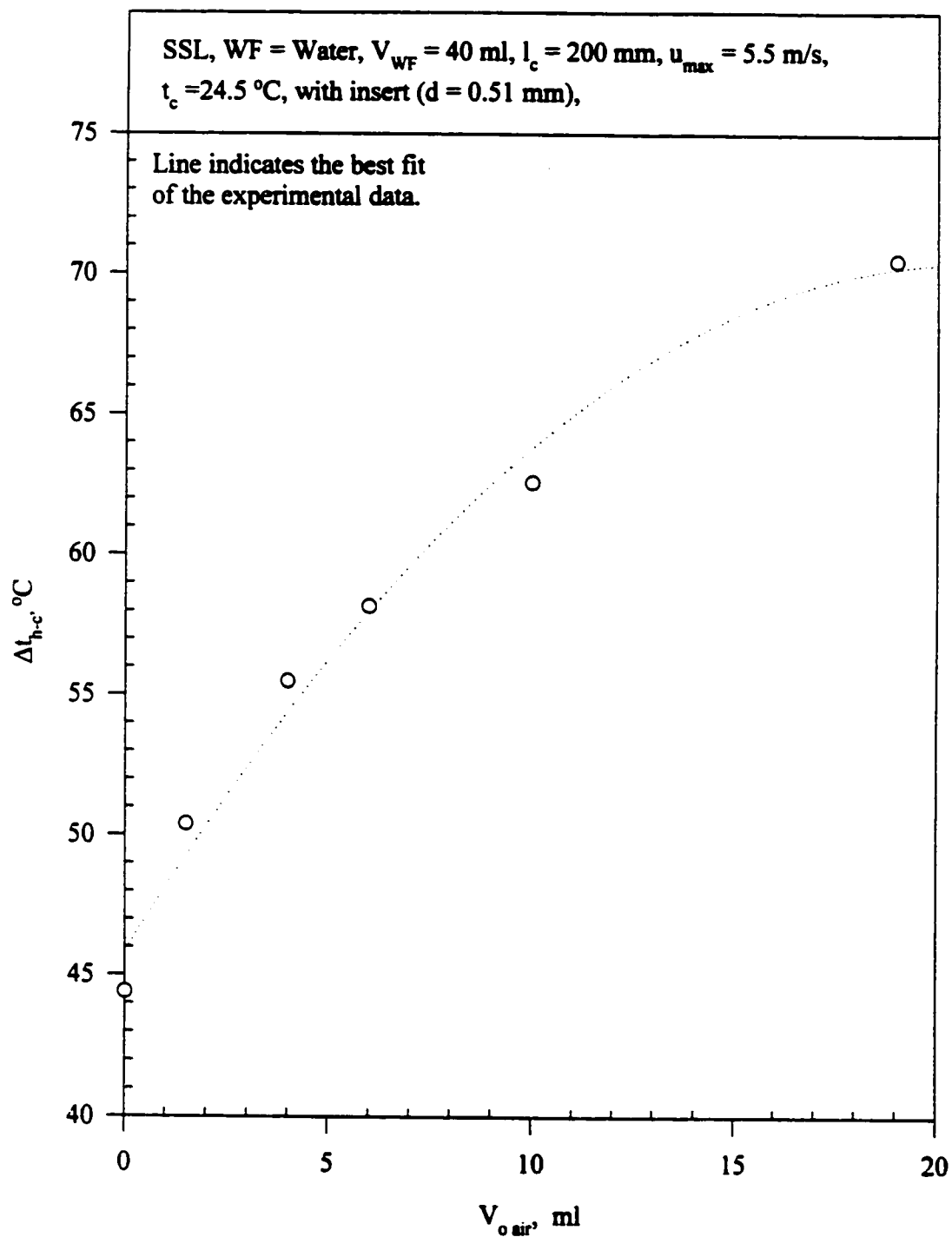
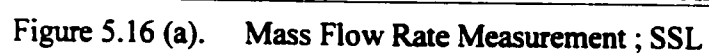


Figure 5.15. Effect of Non-condensable Gas in the Loop on Performance; SSL



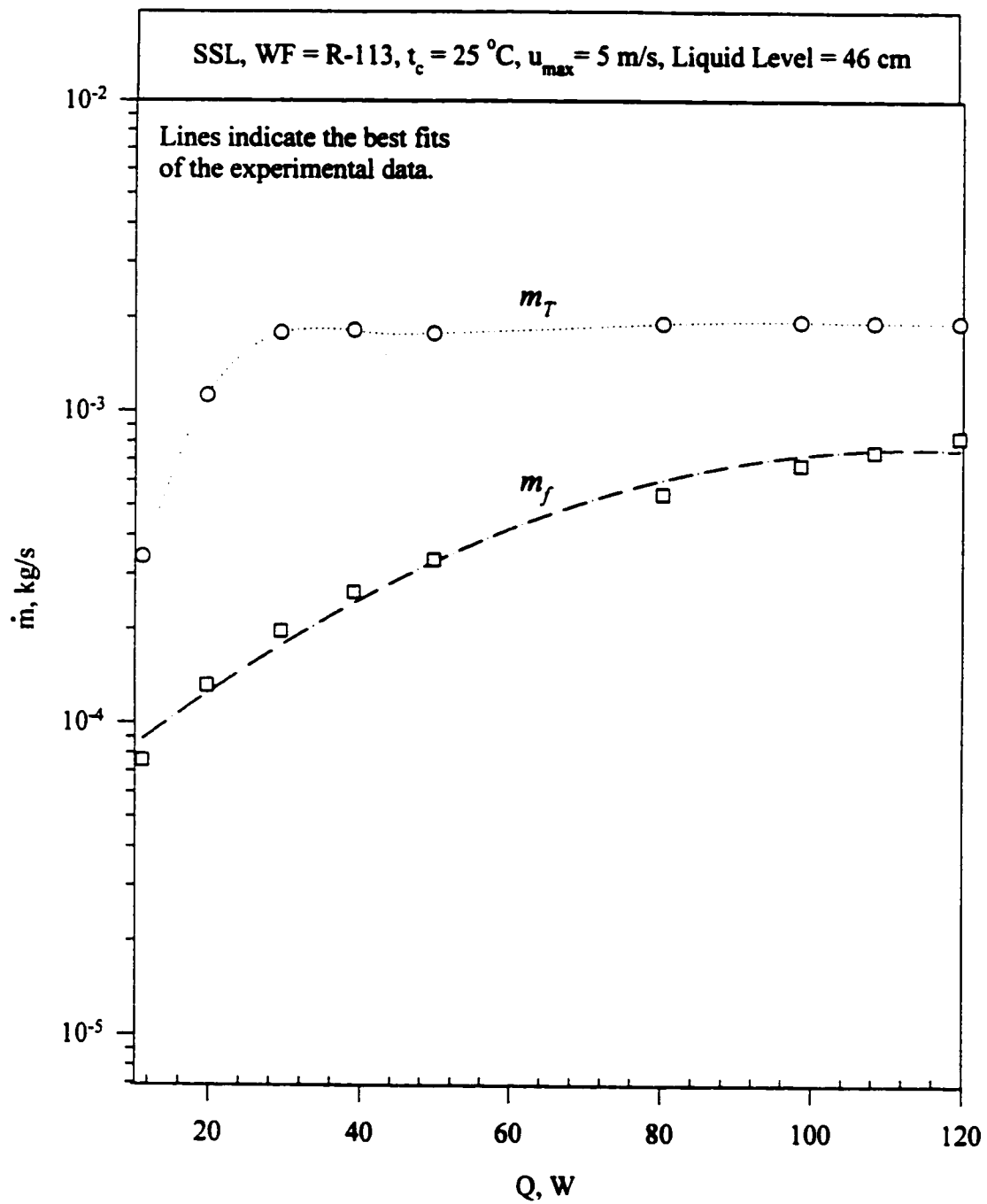
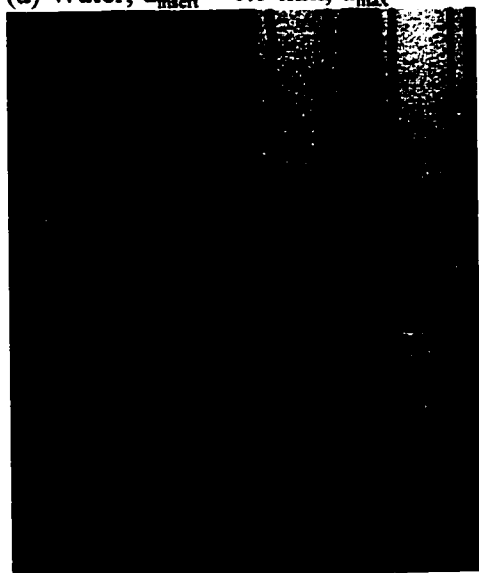


Figure 5.16 (b). Total and Apparent Masses of the Coolant ; SSL

(a) Water, $d_{\text{insert}} = 0.5 \text{ mm}$, $u_{\text{max}} = 5 \text{ m/s}$



1	2.1	2.2	3
Q = 20 W	Q = 60 W	Q = 100 W	
$t_w = 41^\circ\text{C}$	$t_w = 50^\circ\text{C}$	$t_w = 60^\circ\text{C}$	

(b) R-113, $u_{\text{max}} = 5 \text{ m/s}$



1.1	1.2	2.1	2.2	3
Q = 20 W	Q = 60 W	Q = 100 W		
$t_w = 30^\circ\text{C}$	$t_w = 36^\circ\text{C}$	$t_w = 46^\circ\text{C}$		

(c) Ethanol, $u_{\text{max}} = 0 \text{ m/s}$



1.1	1.2	2.1	2.2	3.1	3.2
Q = 20 W	Q = 60 W	Q = 100 W			
$t_w = 41^\circ\text{C}$	$t_w = 63^\circ\text{C}$	$t_w = 73^\circ\text{C}$			

(d) FC-87, $u_{\text{max}} = 5 \text{ m/s}$



1.1	1.2	2.1	2.2	3.1	3.2
Q = 20 W	Q = 60 W	Q = 100 W			
$t_w = 34^\circ\text{C}$	$t_w = 44^\circ\text{C}$	$t_w = 52^\circ\text{C}$			

Figure 5.17. Flow Patterns; $V_{\text{WF}} = 40 \text{ ml}$ (SSL)

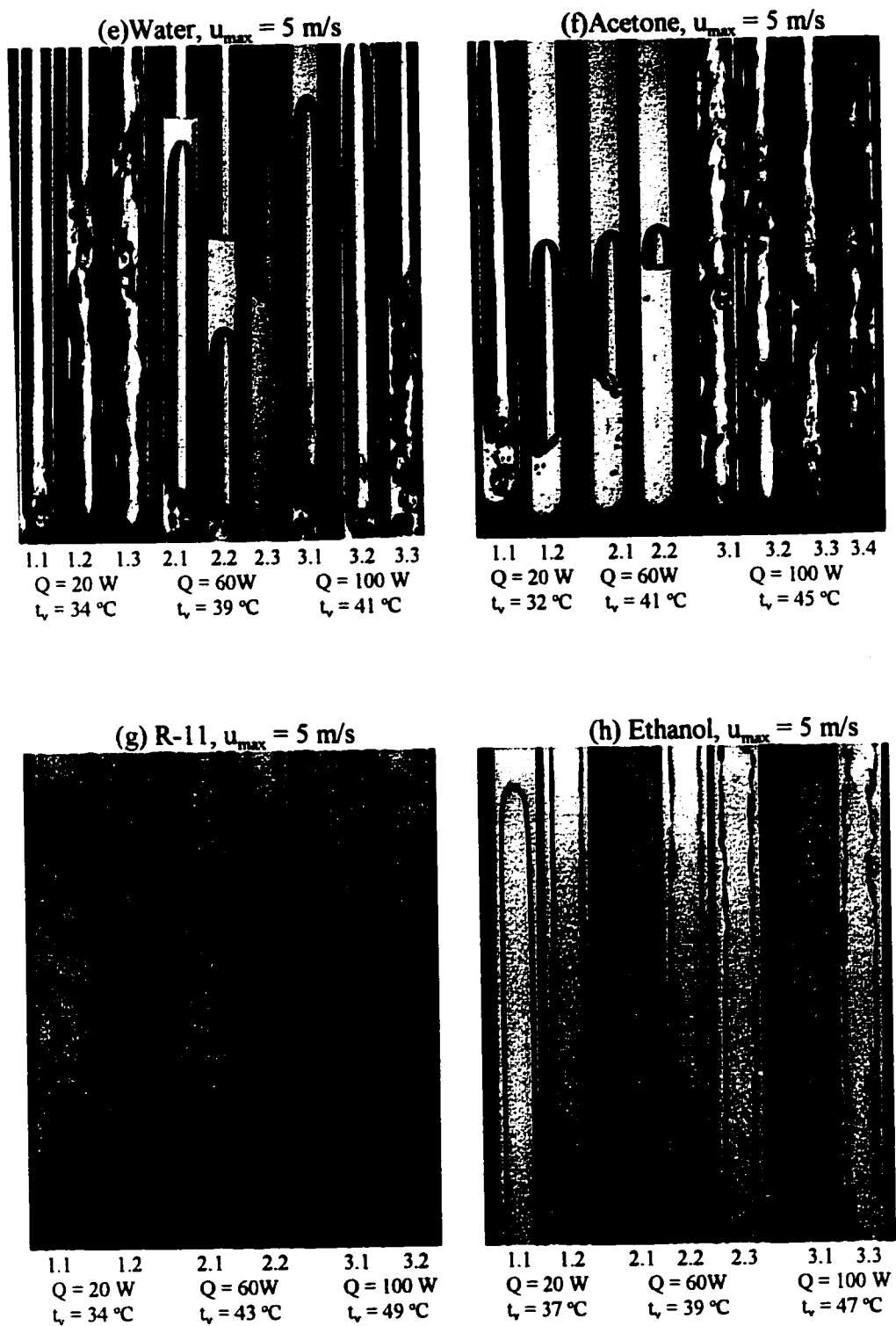


Figure 5.17. Flow Patterns; $V_{WF} = 40 \text{ ml}$ (SSL - Continued)

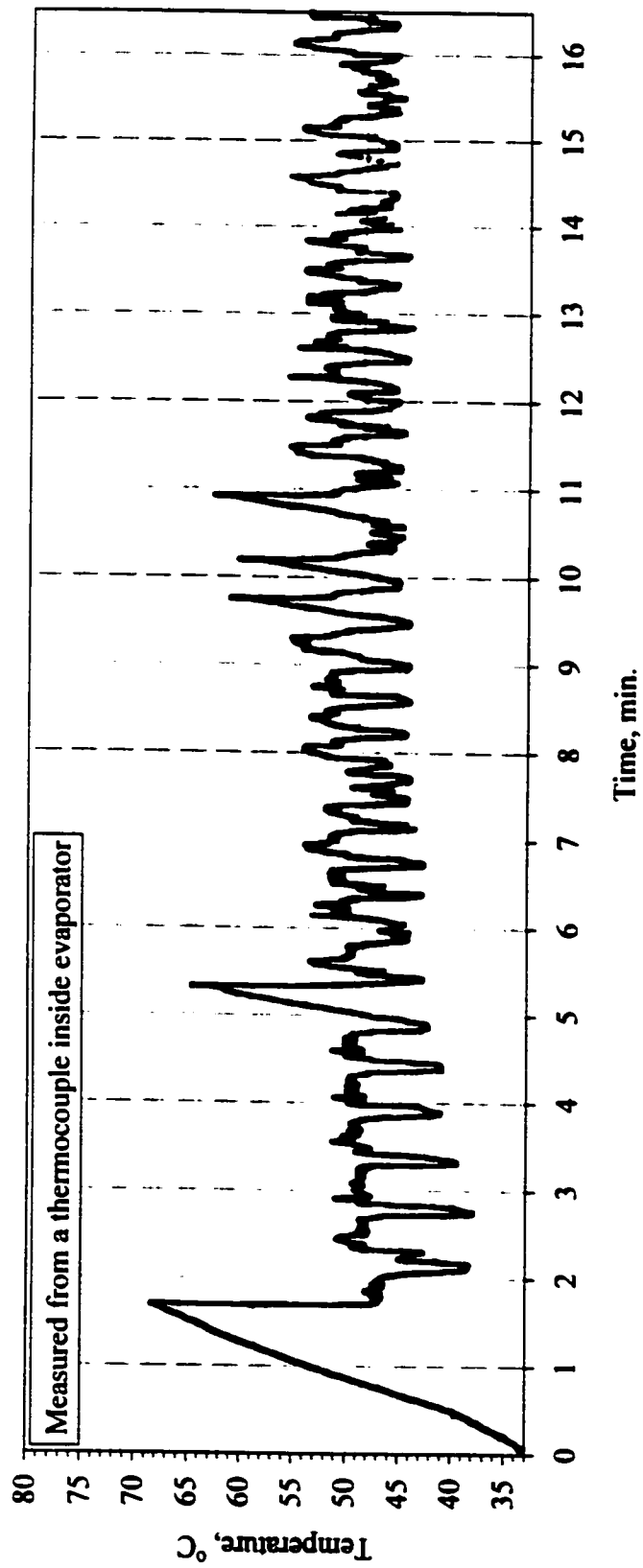


Figure 5.18 (a). Transient Temperature Variation from Startup
 WF = Water, $V_{WF} = 40$ ml, $l_c = 200$ mm, $t_c = 20.5$ °C,
 1 condenser, $Q = 60$ W, SSL

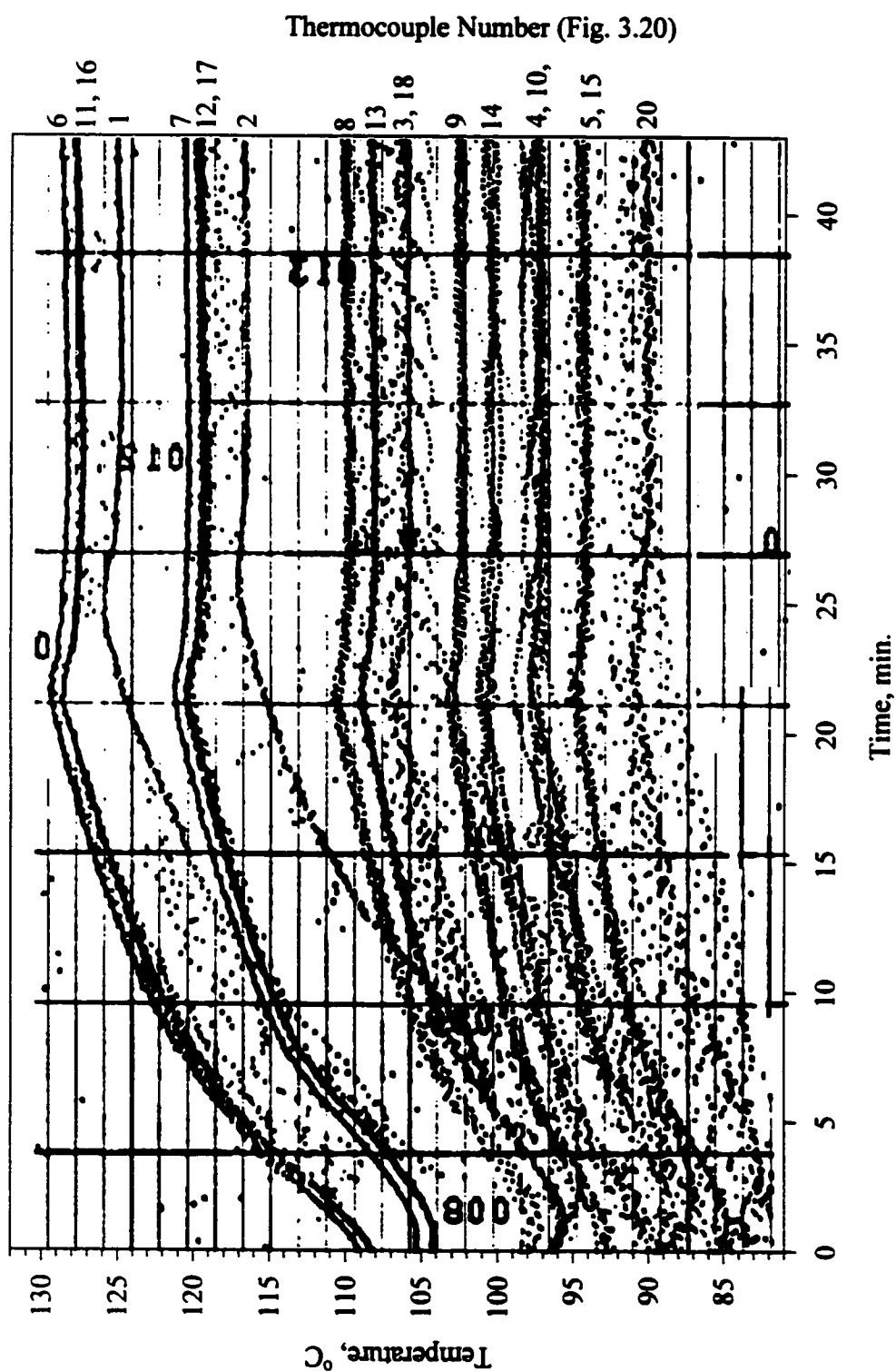


Figure 5.18 (b). Transient Temperature Variation
 WF = Water, $V_{WFR} = 75\%$, $u_c = 2.33$ m/s, $u_c = 3.91$ m/s,
 $t_c = 20^\circ\text{C}$, $Q = 100,000$ W, LSL II

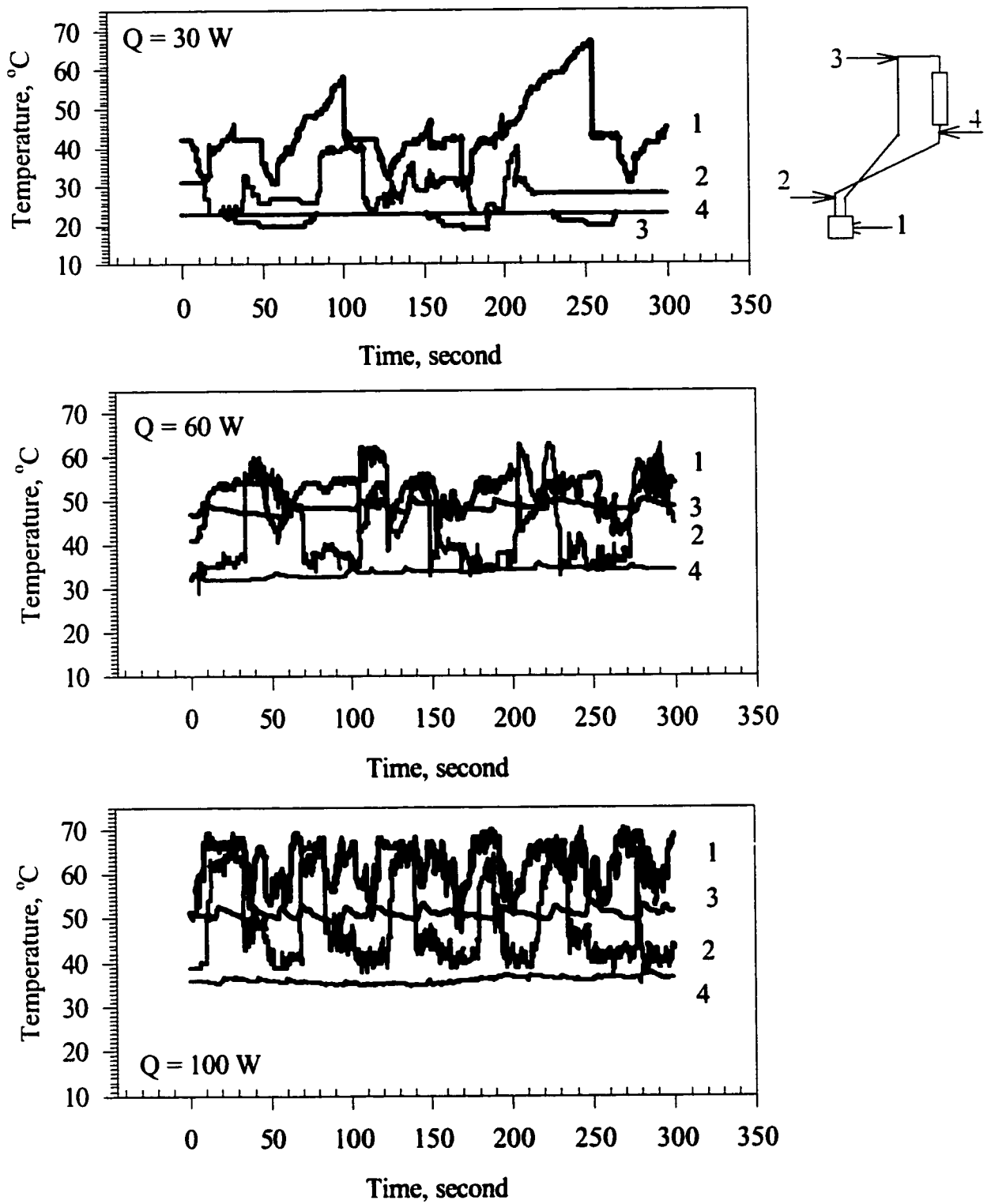


Figure 5.19 (a). Transient Temperature Fluctuation vs. Heat Flux; WF = Water, SSL

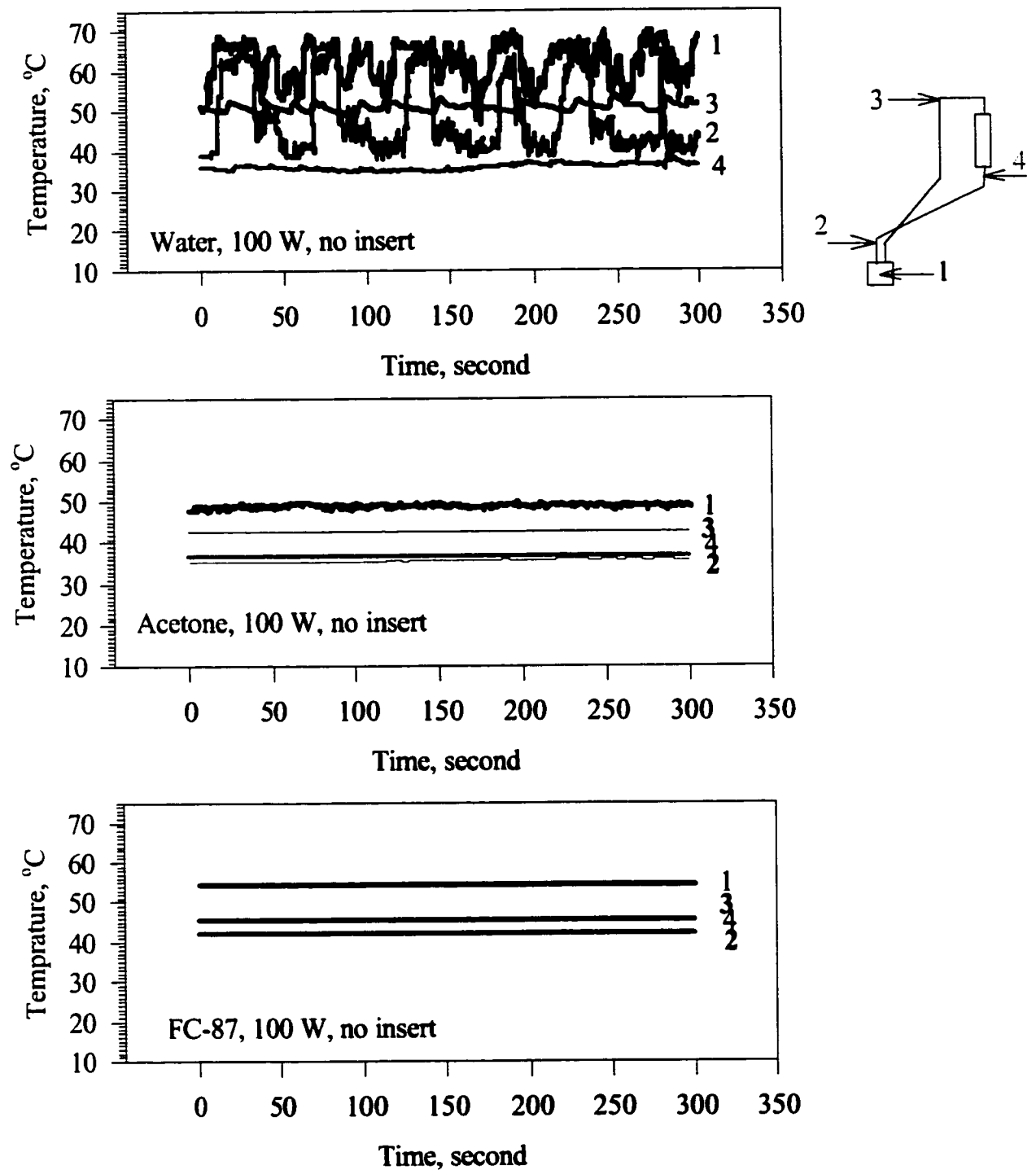


Figure 5.19 (b). Transient Temperature Fluctuation vs. Heat Flux;
 WF = Water, Acetone and FC-87, $V_{WF} = 40$ ml,
 $l_c = 200$ mm, $u_{max} = 5$ m/s (SSL)

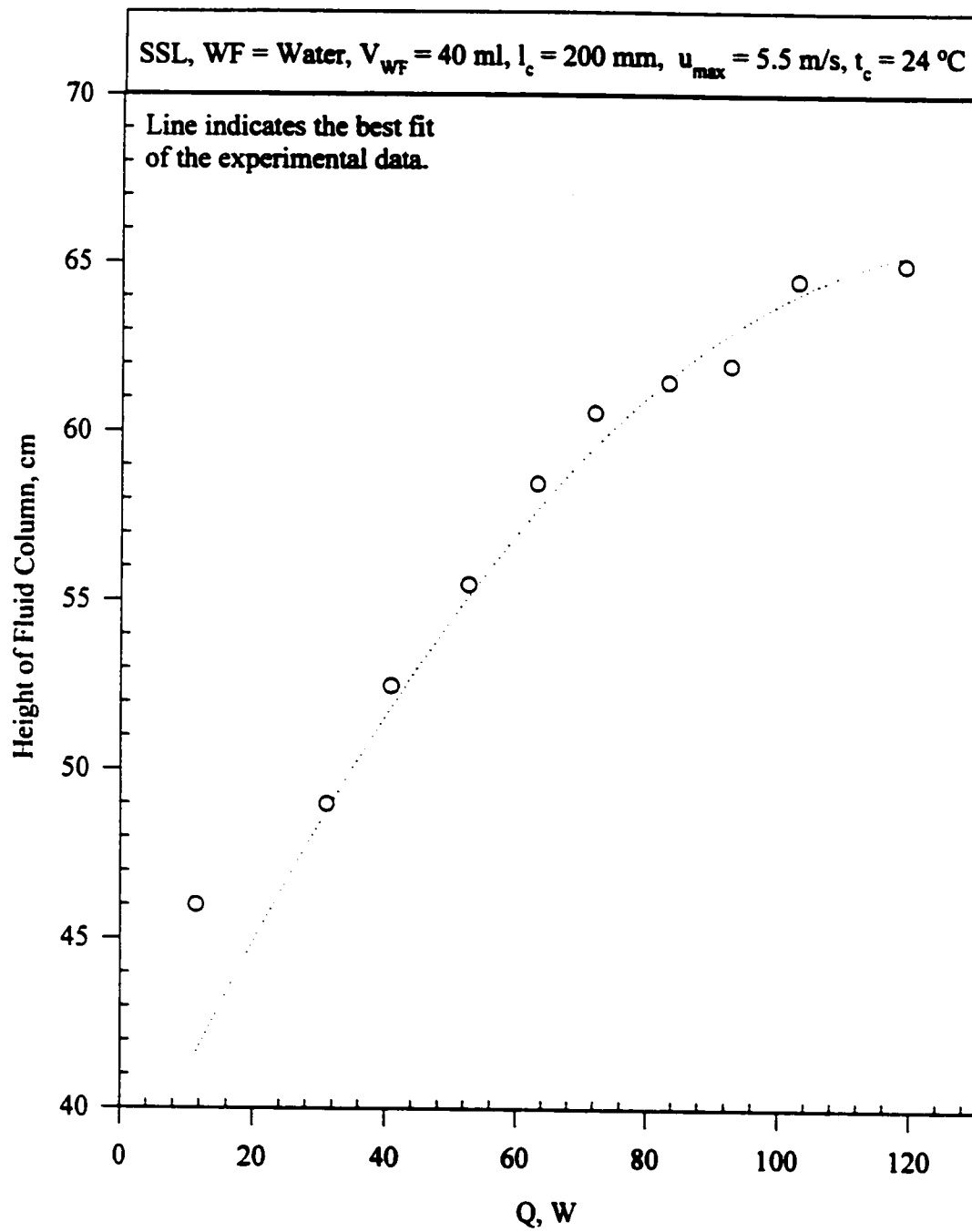


Figure 5.20 (a). Effect of Q on $h_{\text{fluid col.}}$; SSL

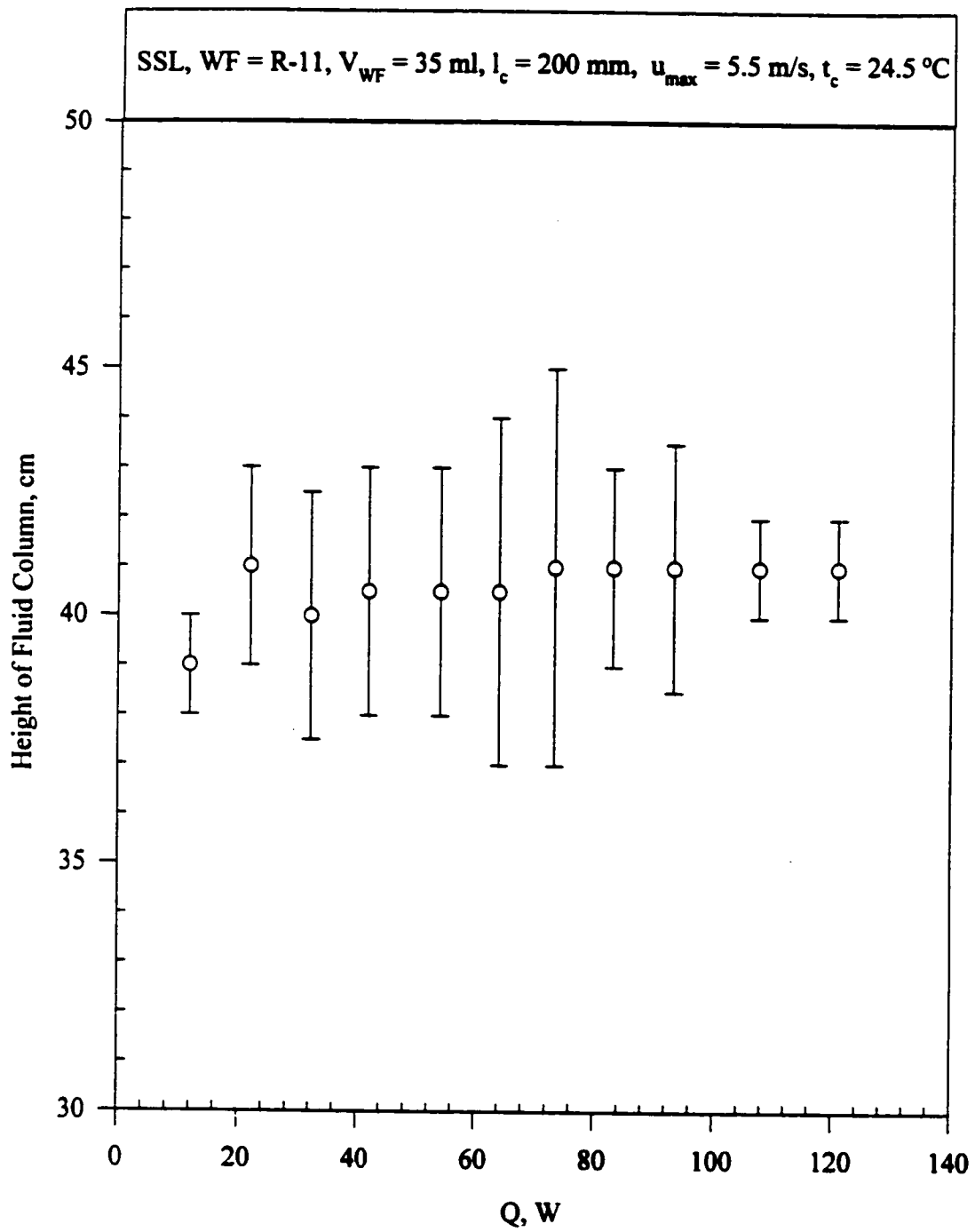


Figure 5.20 (b). Effect of Q on $h_{\text{fluid col.}}$; SSL

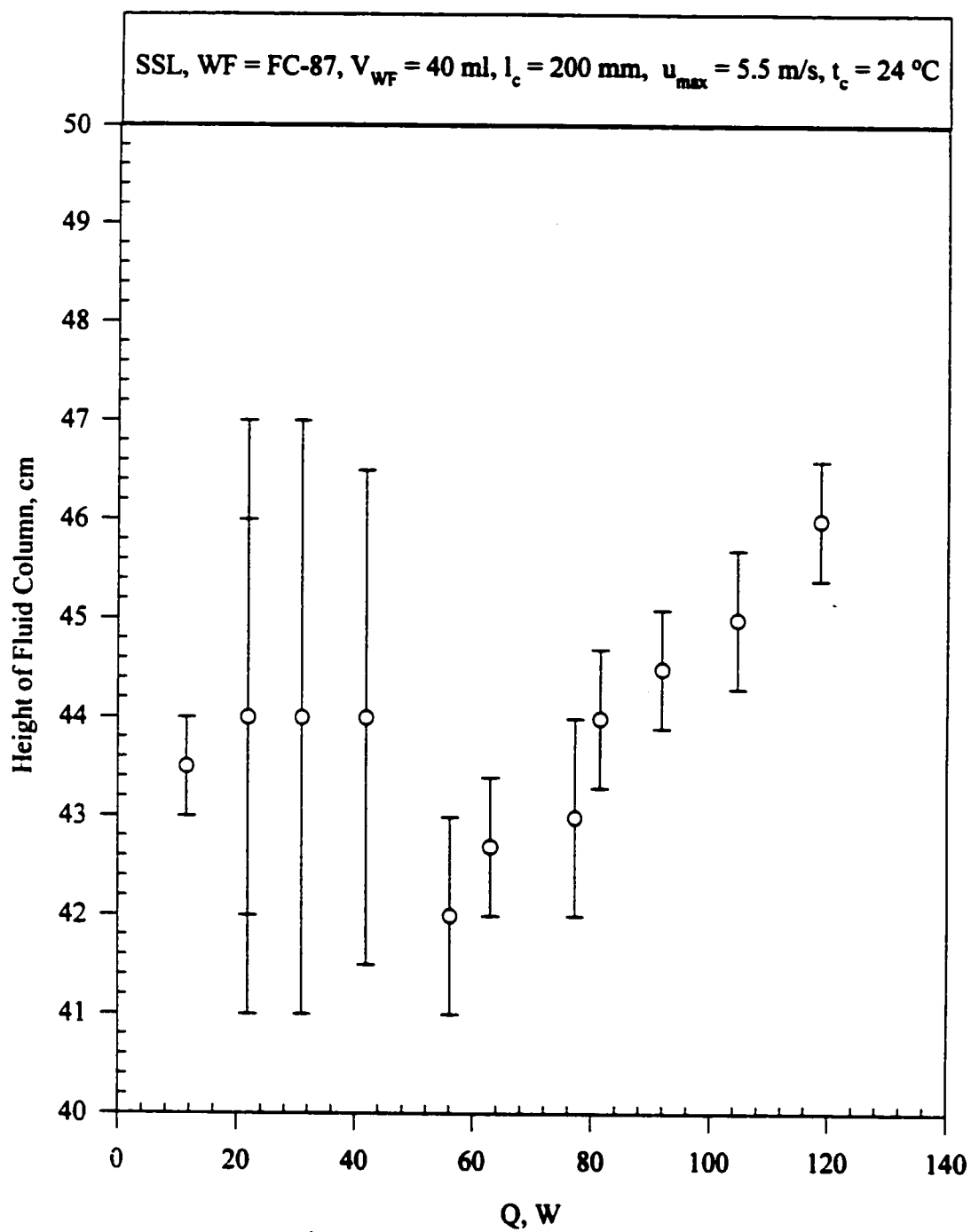


Figure 5.20 (c). Effect of Q on $h_{\text{fluid col.}}$; SSL

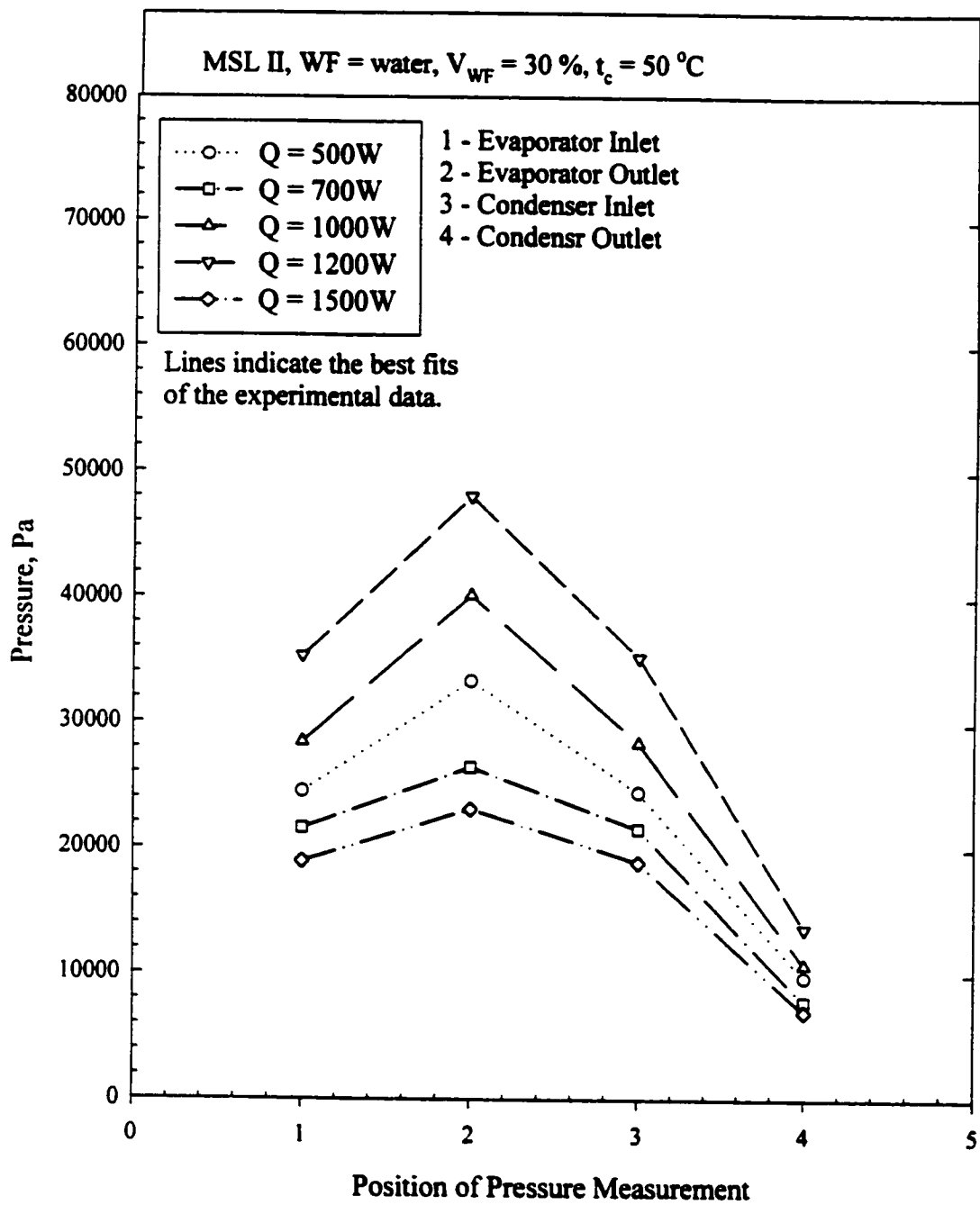


Figure 5.21 (a). Pressure Distribution of TLT ; MSL II

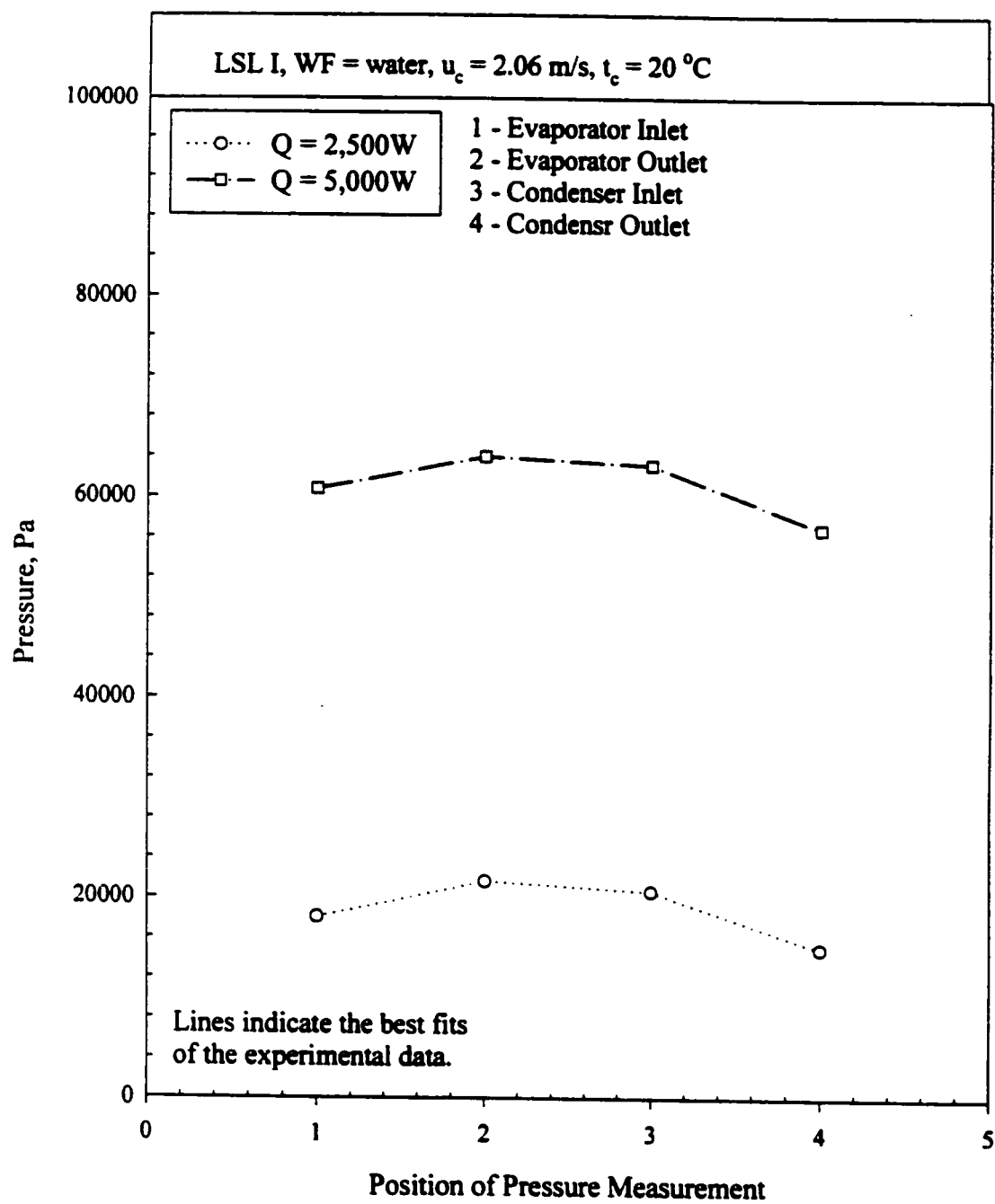


Figure 5.21 (b). Pressure Distribution of TLT ; LSL I

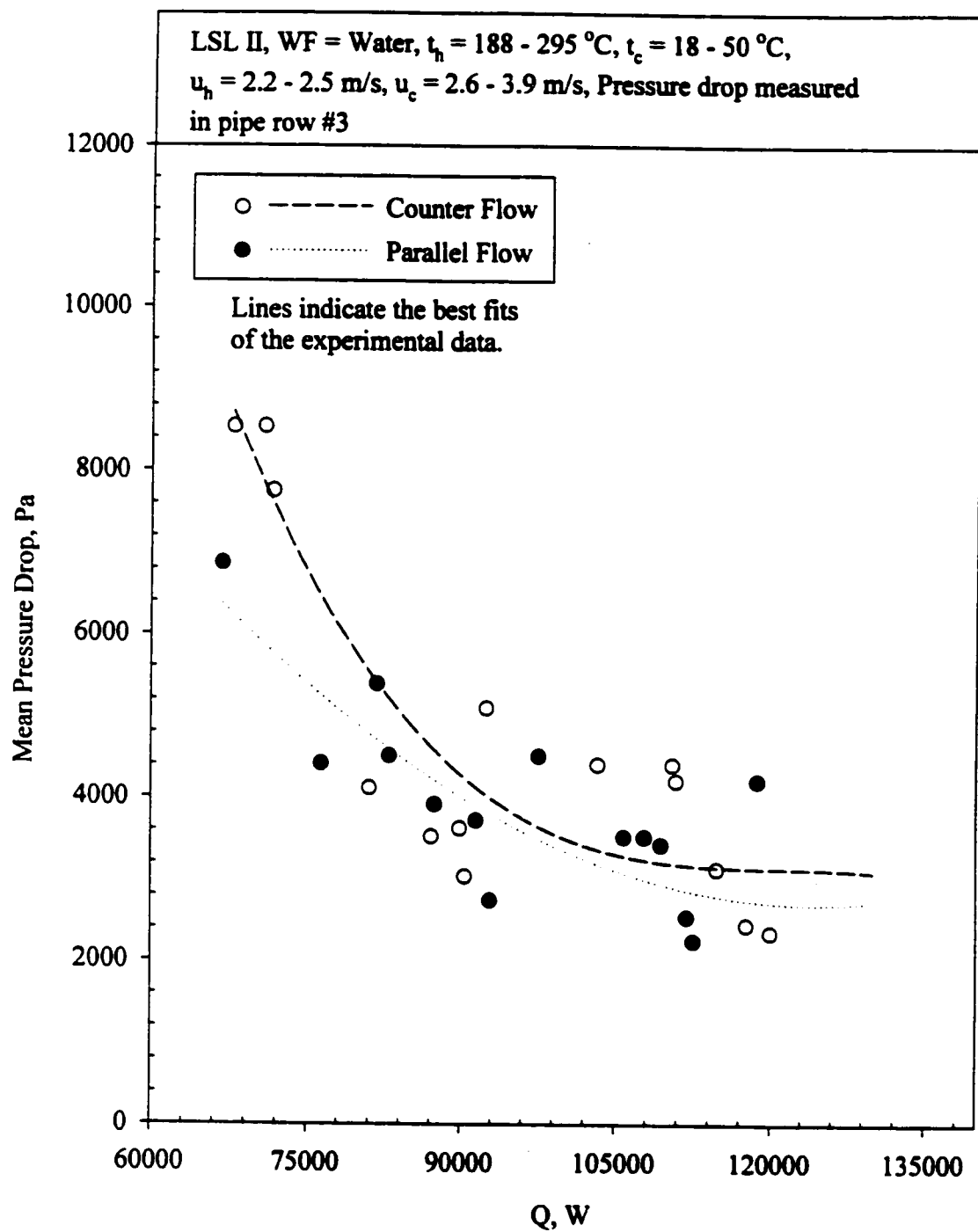


Figure 5.22 (a). Effect of Q on Pressure Drop; LSL II

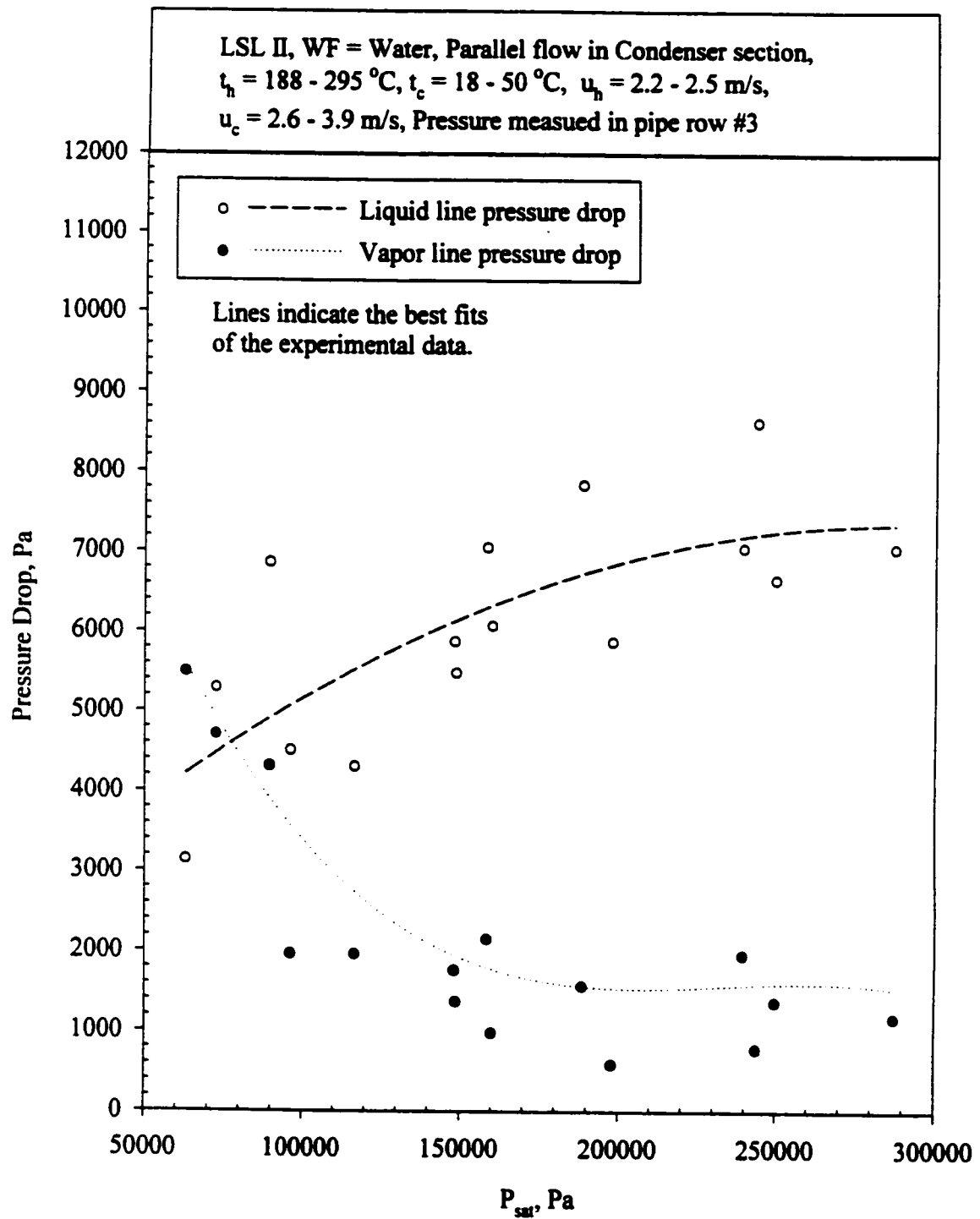


Figure 5.22 (b). Effect of P_{sml} on Pressure Drop ; LSL II with Parallel flow

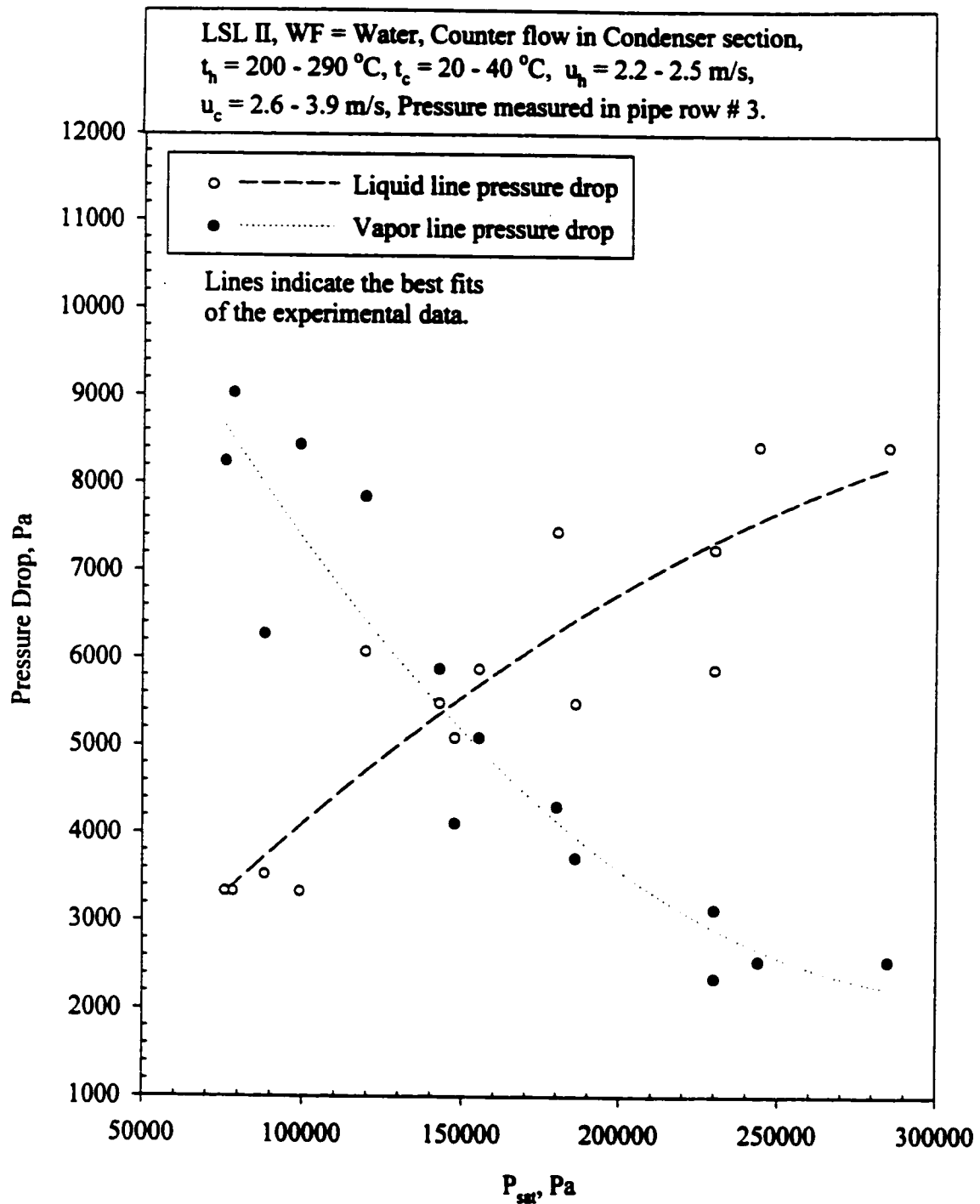


Figure 5.22 (c). Effect of P_{sst} on Pressure Drop ; LSL II with Counter flow

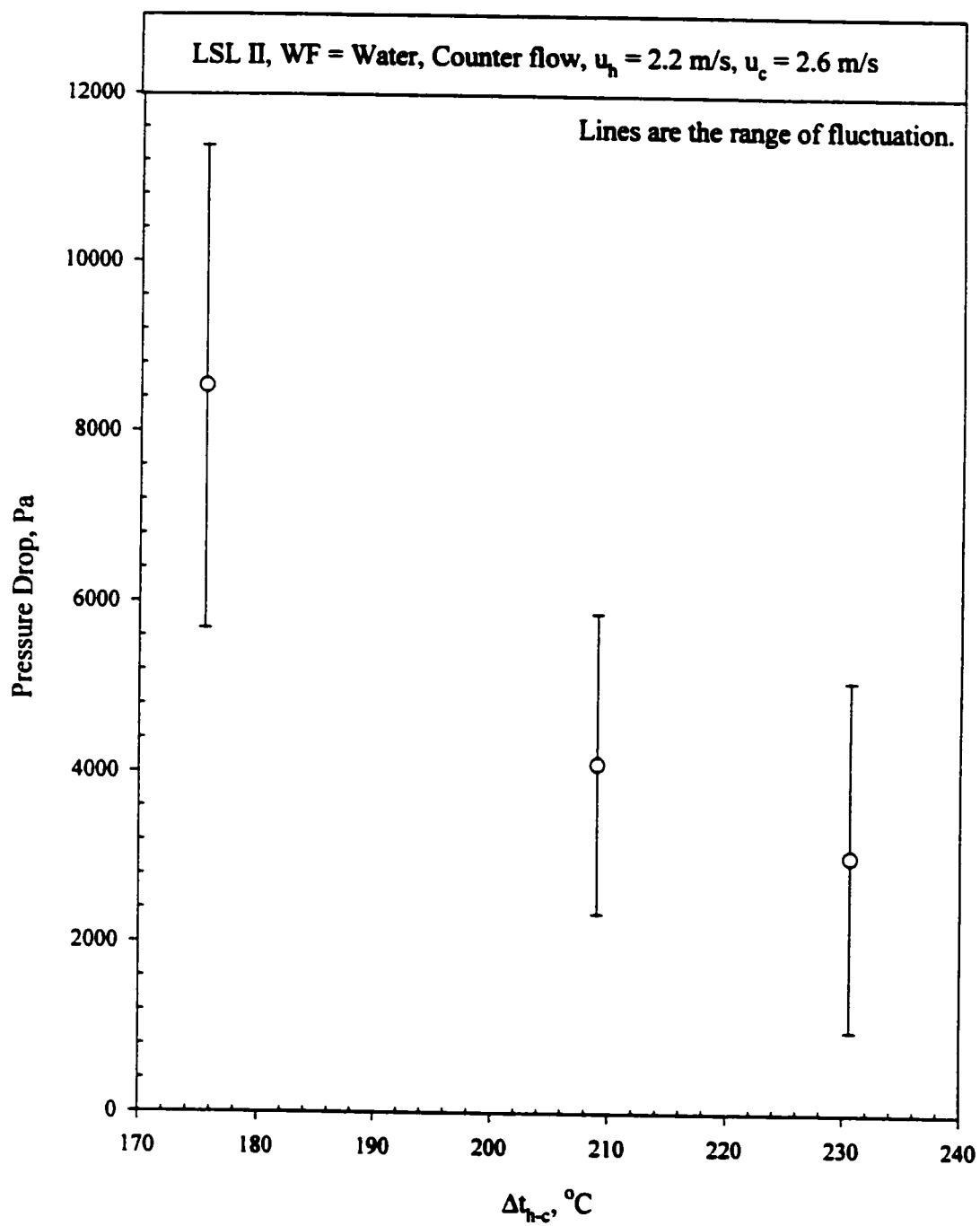


Figure 5.22 (d). Effect of Δt_{h-c} on Pressure Drop Fluctuation; Counter Flow, LSL II

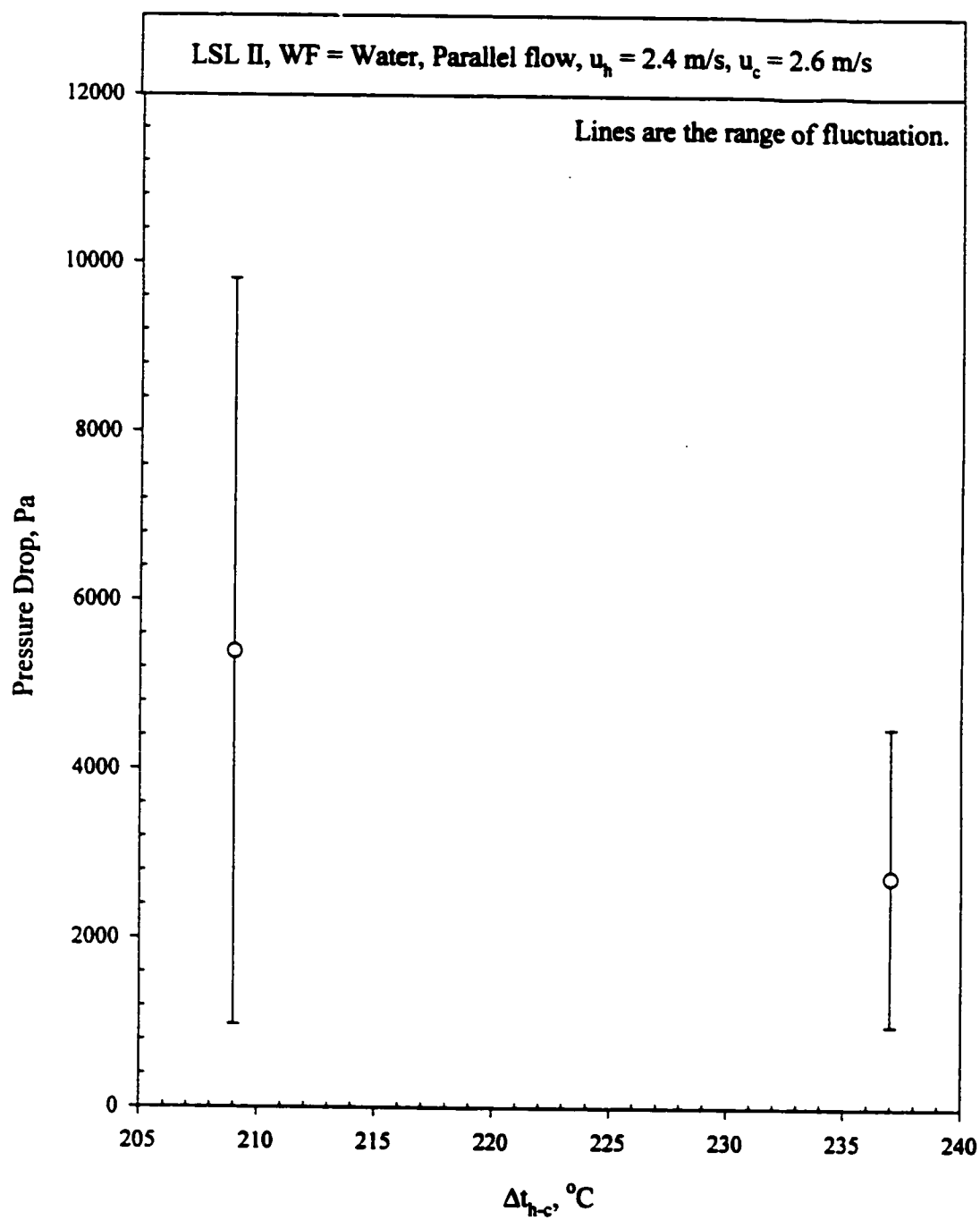


Figure 5.22 (e). Effect of Δt_{h-c} on Pressure Drop Fluctuation;
Parallel Flow, LSL II

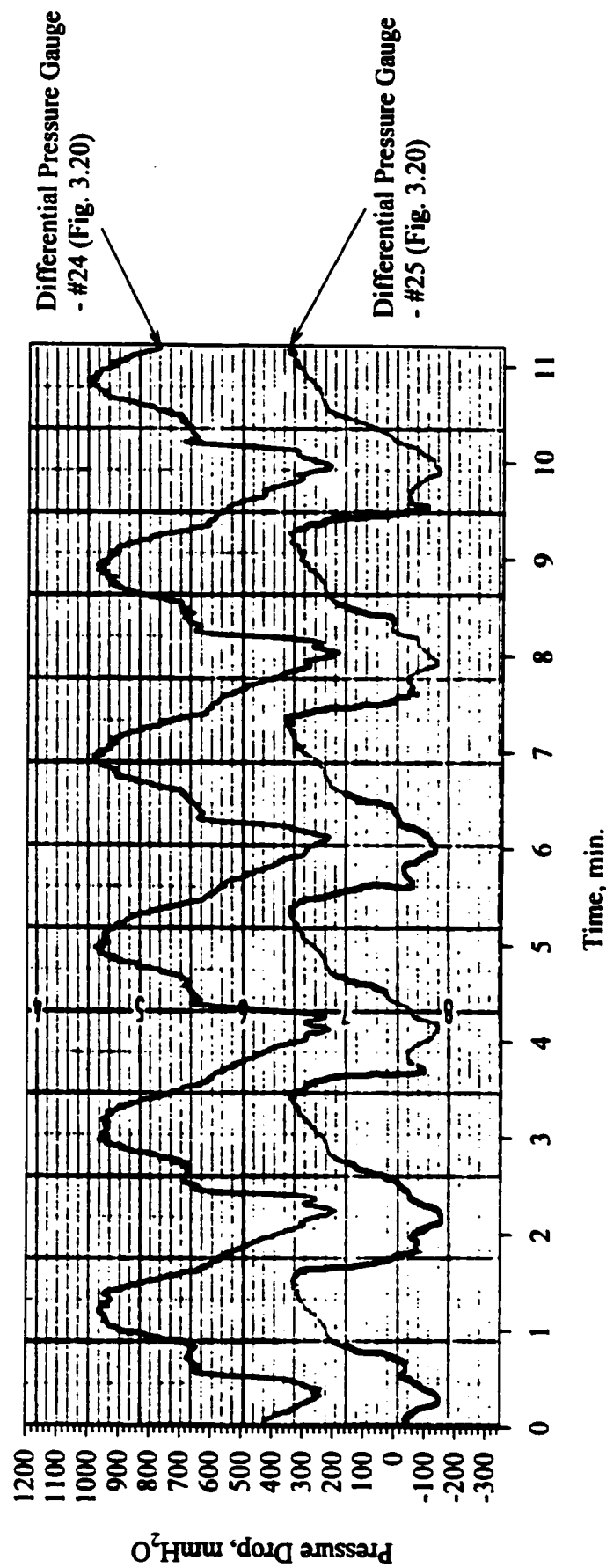


Figure 5.22 (f). Transient Pressure Drop Fluctuation;
 WF = Water, $V_{wFR} = 75\%$, $u_c = 1.47$ m/s,
 $t_c = 20^\circ\text{C}$, $Q = 68,000$ W, LSL II with Counter Flow

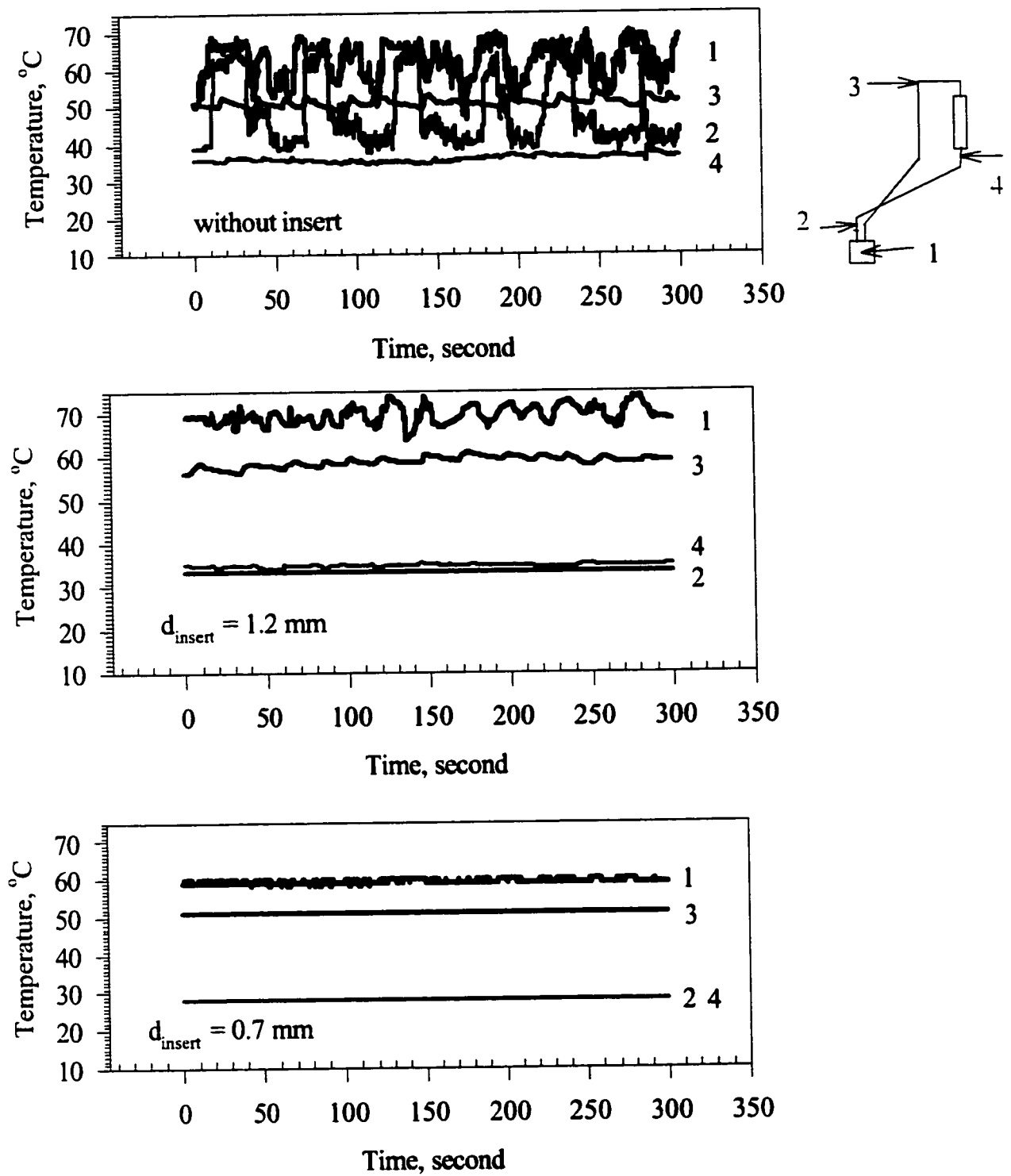


Figure 5.23 (a). Effect of Insert Size on Temperature Fluctuation
Insert Size; WF = Water, $Q = 100 \text{ W}$, SSL

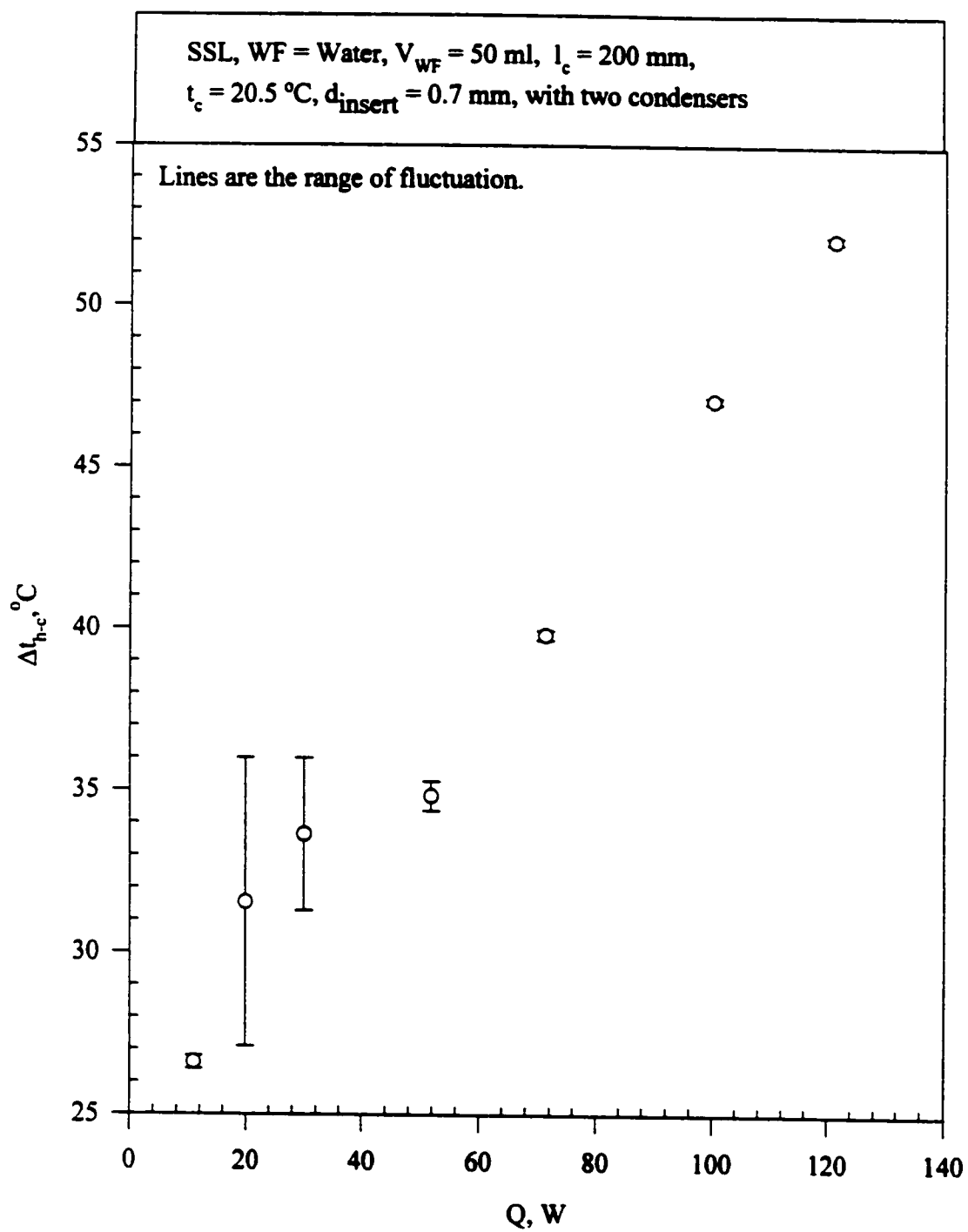


Figure 5.23 (b). Effect of Insert on Q vs. Δt_{h-c} ; SSL

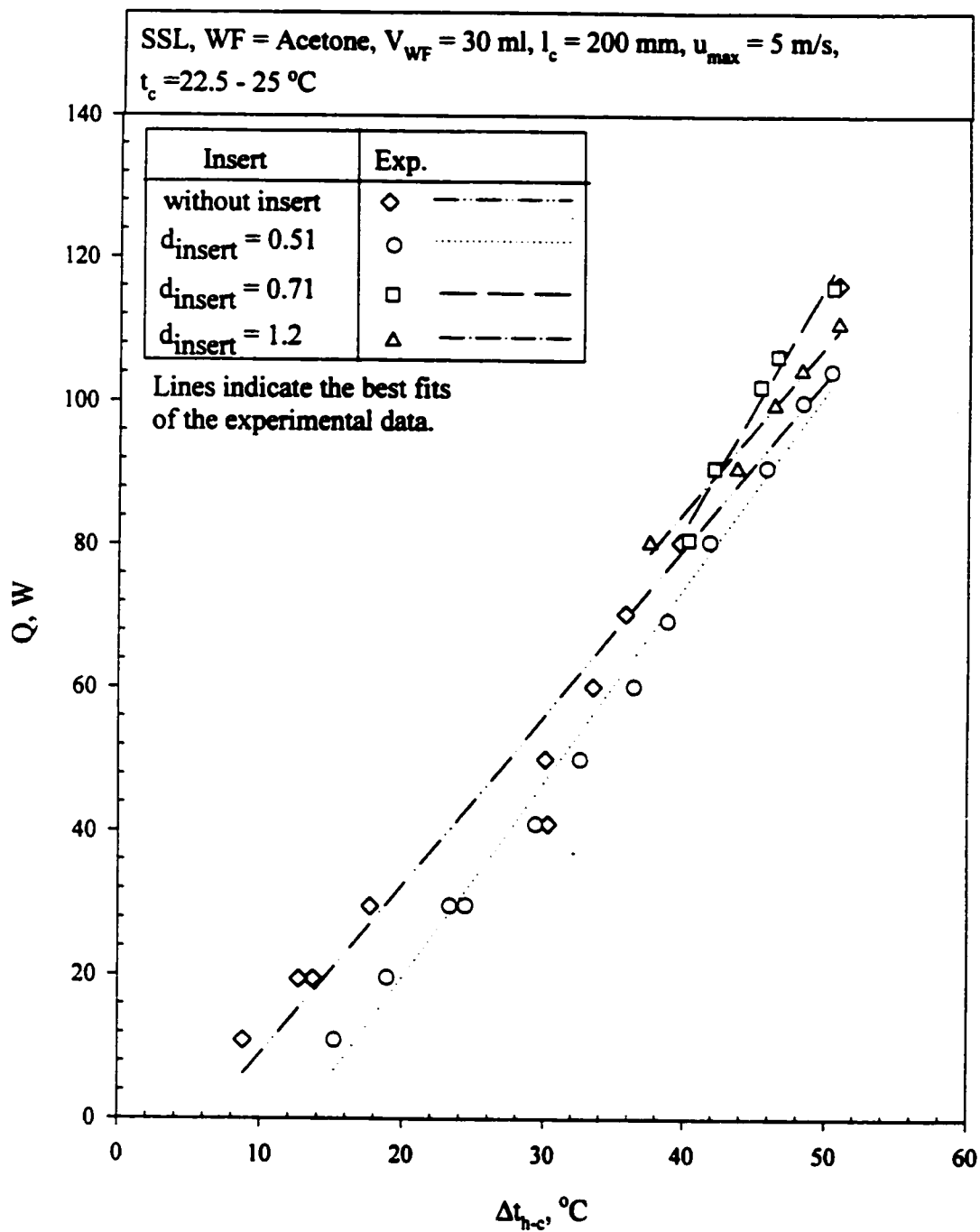


Figure 5.23 (c). Effect of Insert on Δt_{h-c} vs. Q ; SSL

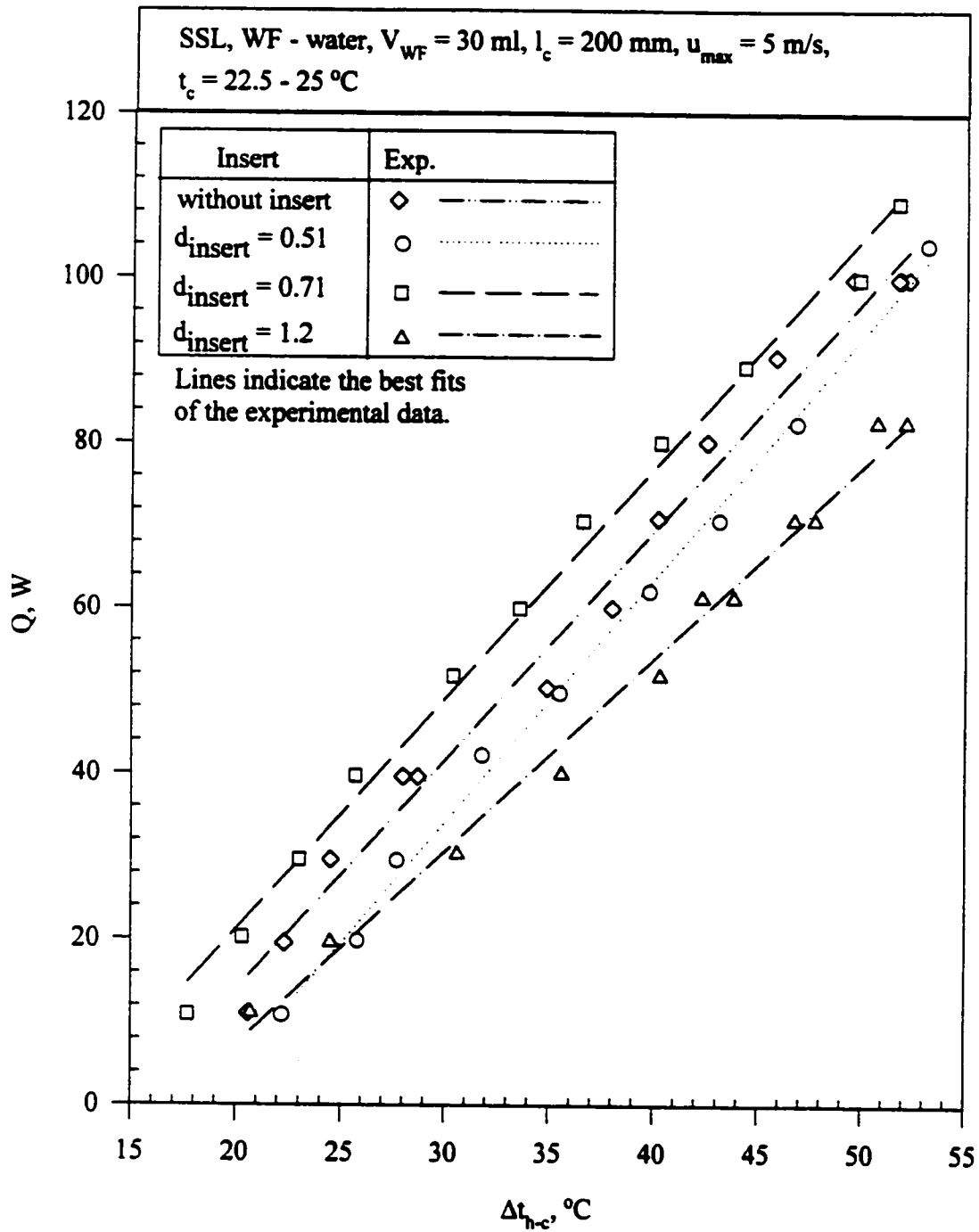


Figure 5.23 (d). Effect of Insert on Δt_{h-c} vs. Q ; SSL

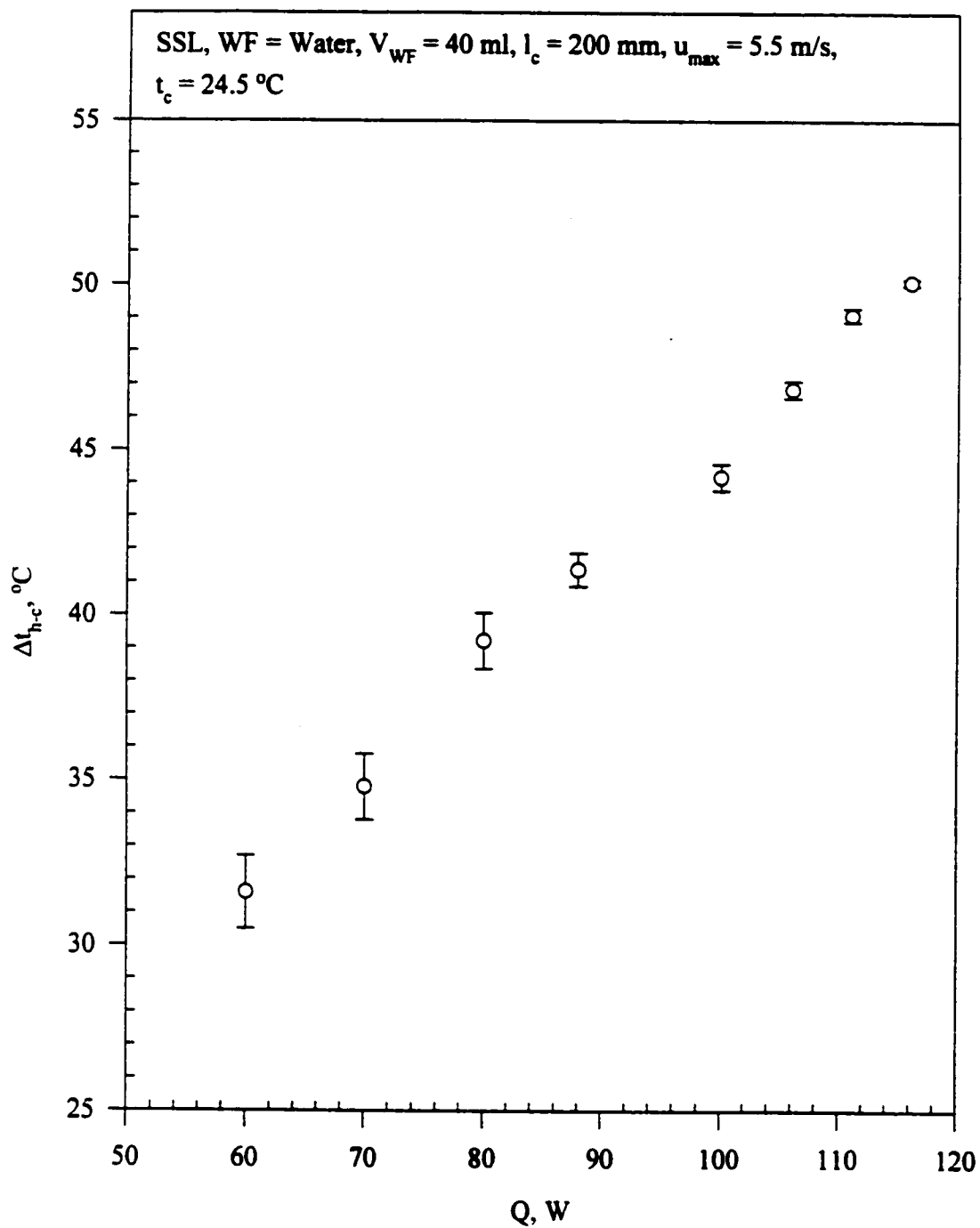


Figure 5.24. Pulsation of Temperature Difference vs. Heat Flux; Forced Convection Mode, SSL

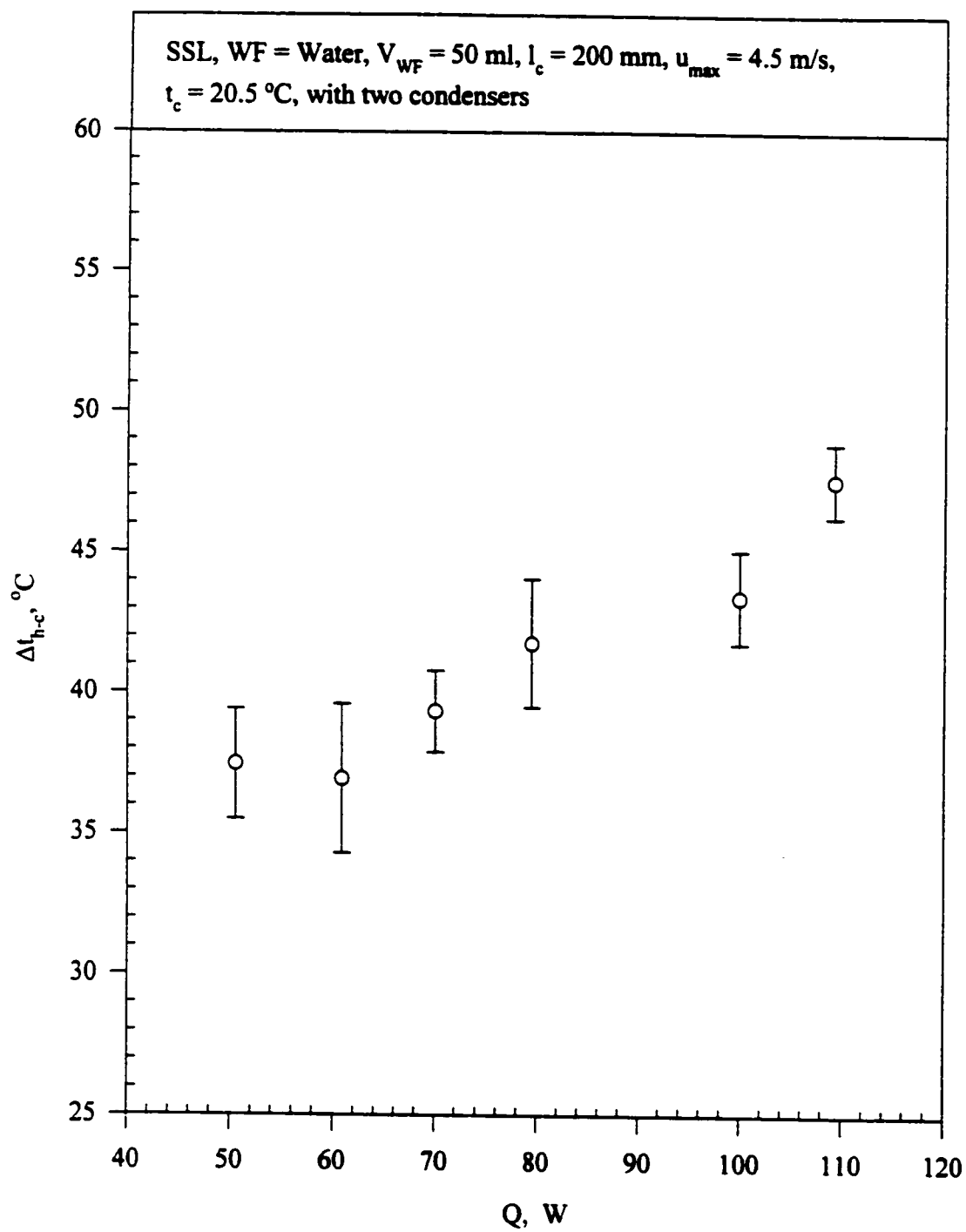


Figure 5.25. Effect of Q on Δt_{h-c} ; Forced Convection Mode, SSL

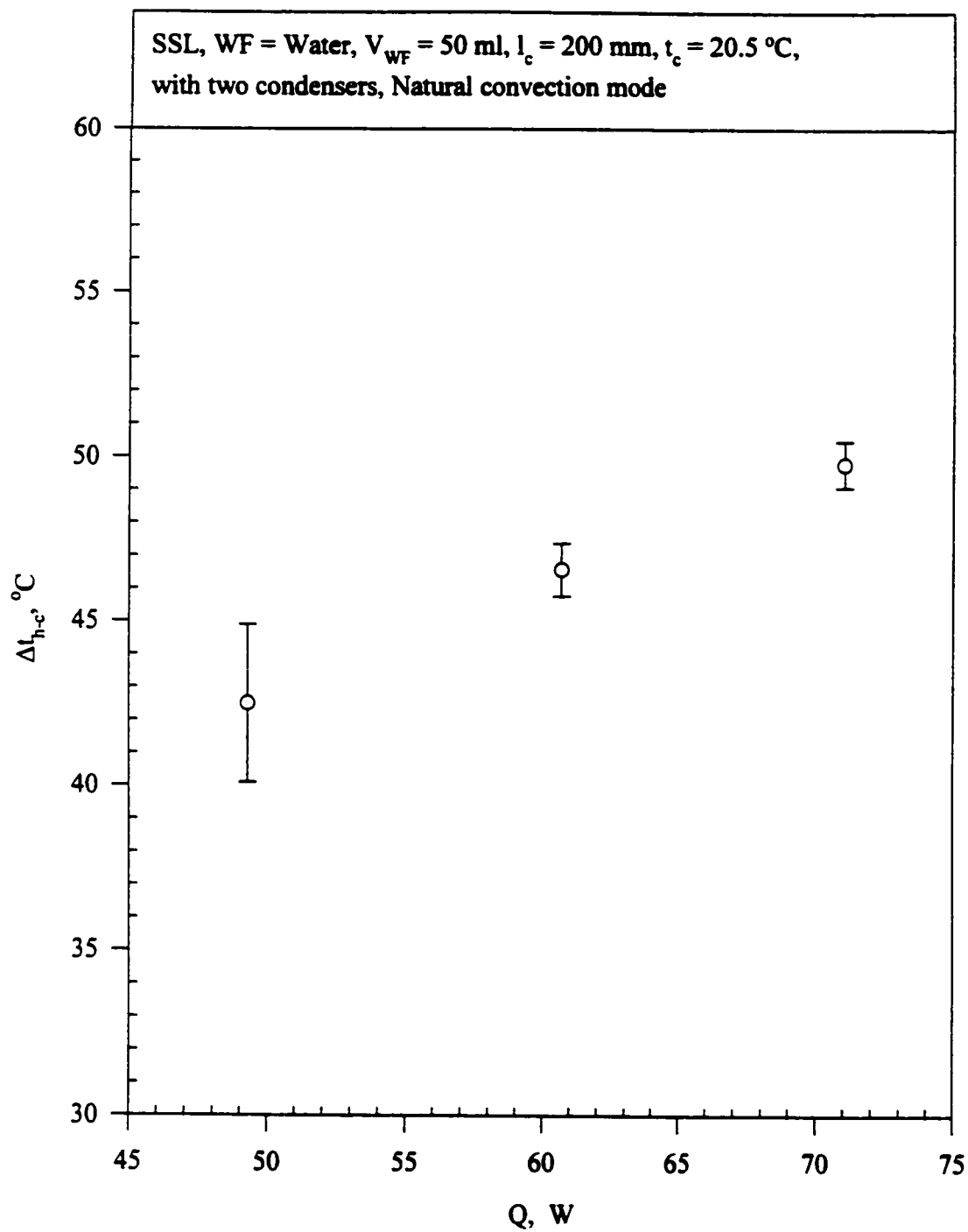


Figure 5.26. Effect of Q on Δt_{h-c} ; Natural Convection Mode, SSL

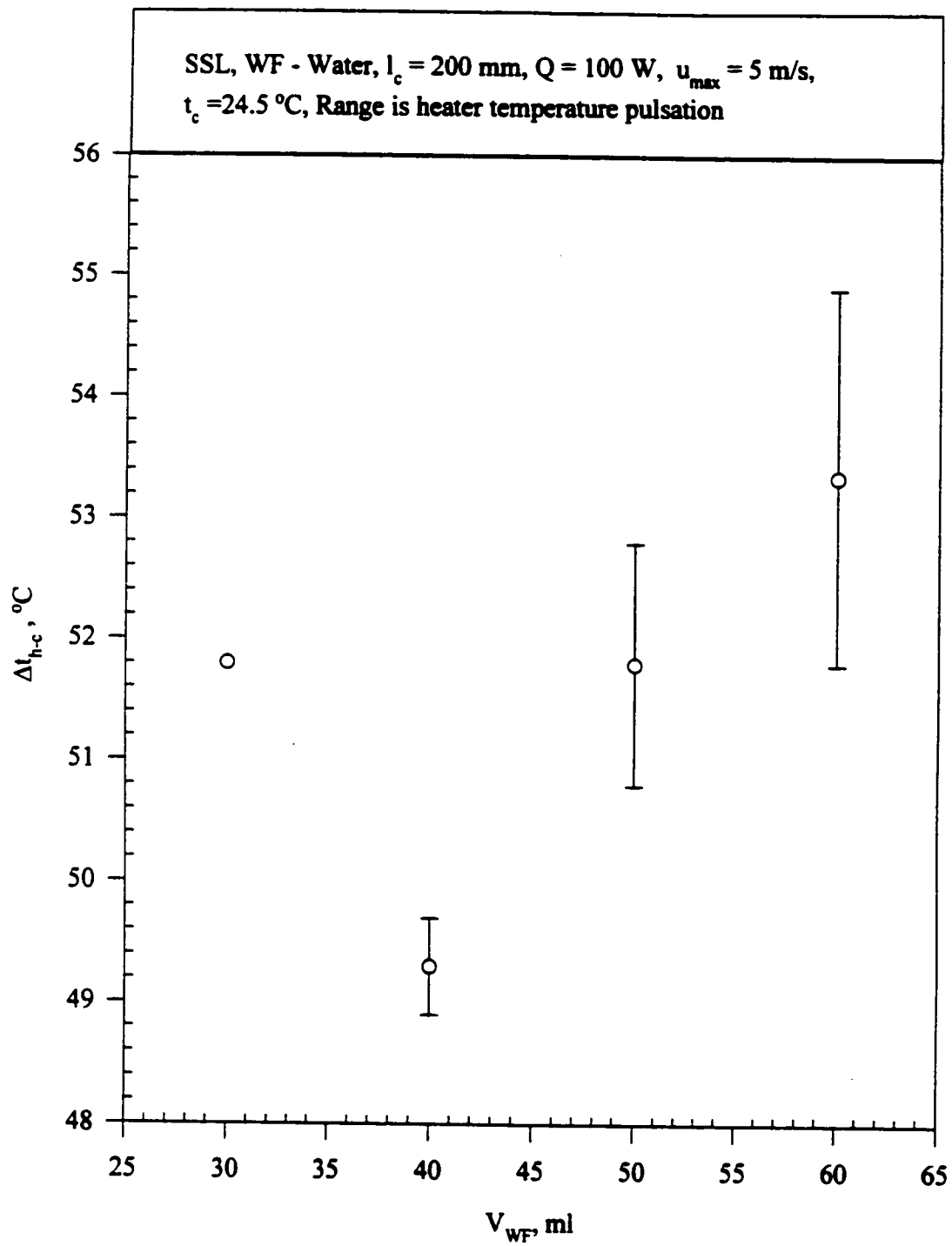


Figure 5.27 (a). Effect of V_{WF} on Δt_{h-c} ; SSL

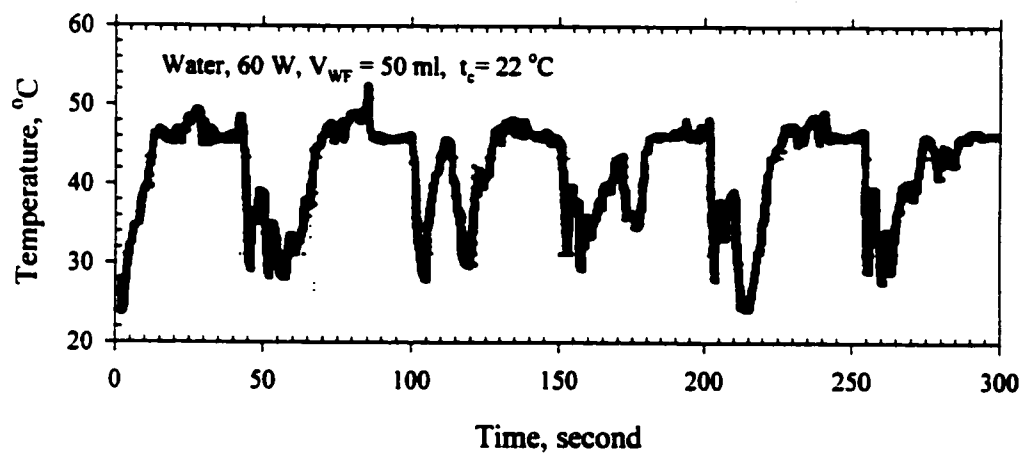
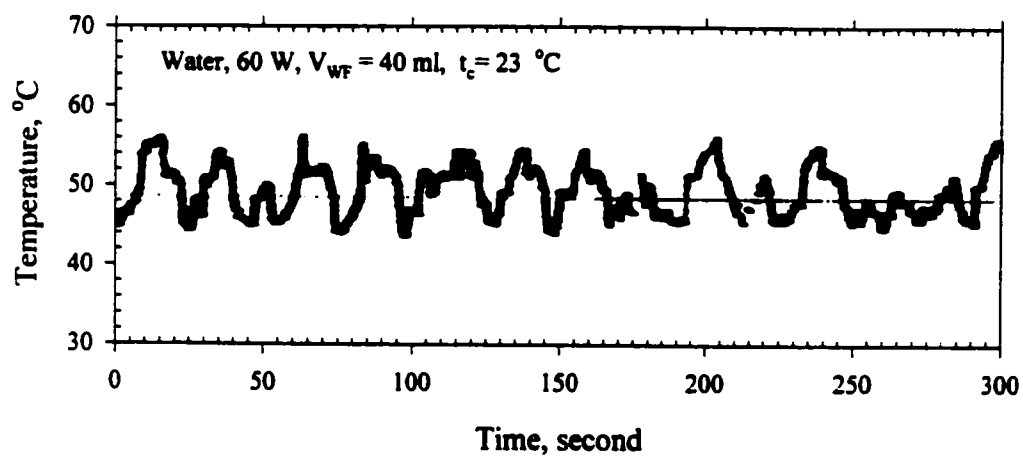
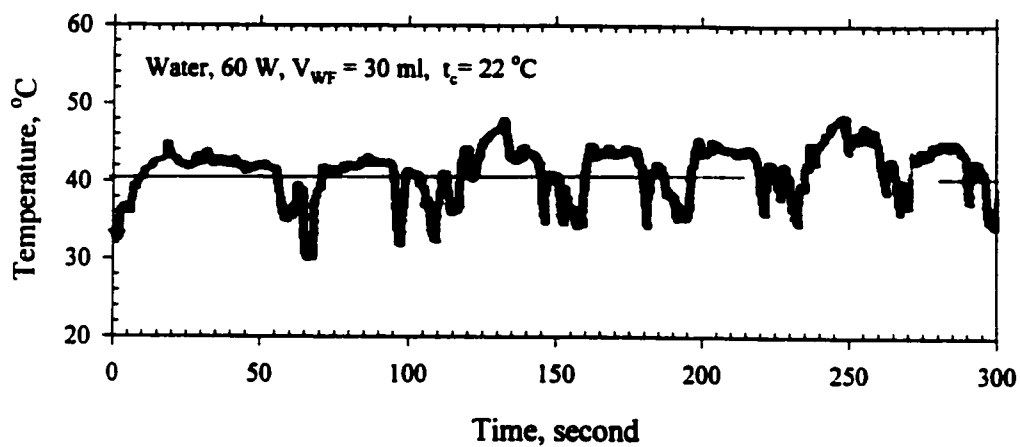


Figure 5.27 (b). Effect of Working Fluid Amount on Instability;
WF = Water, $Q = 50$ W, SSL

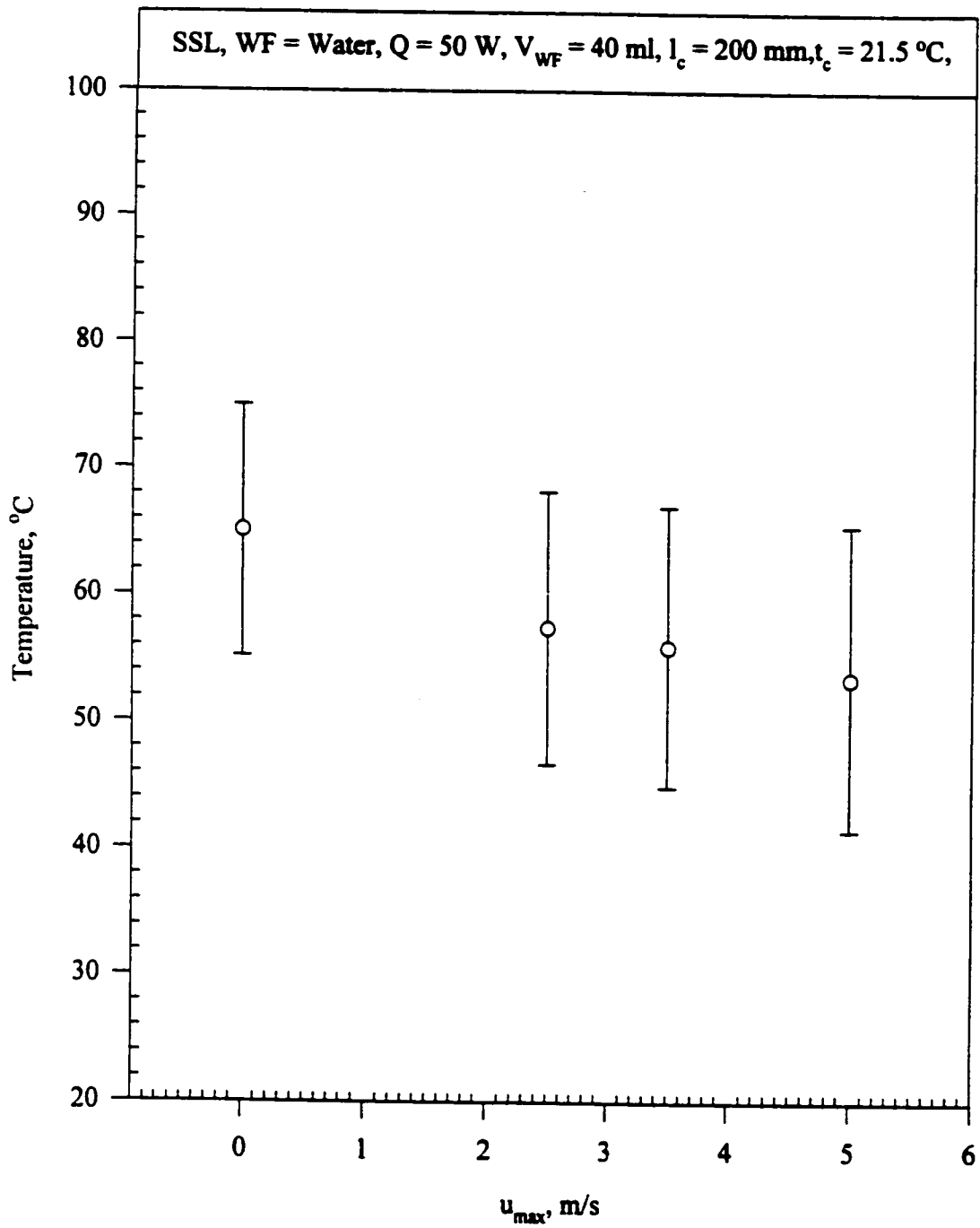


Figure 5.28. Effect of u_{max} on Heater Temperature Fluctuation; SSL

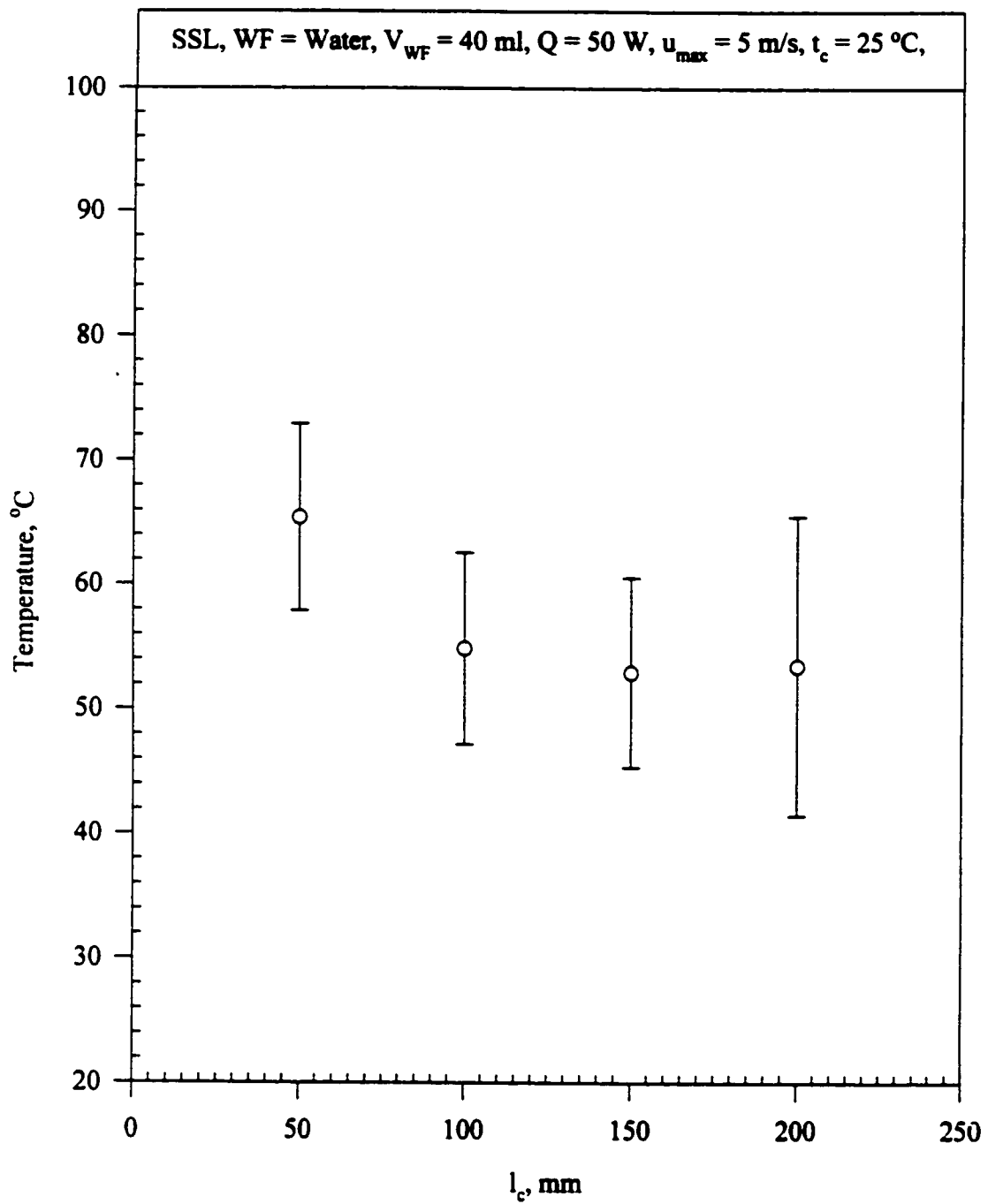
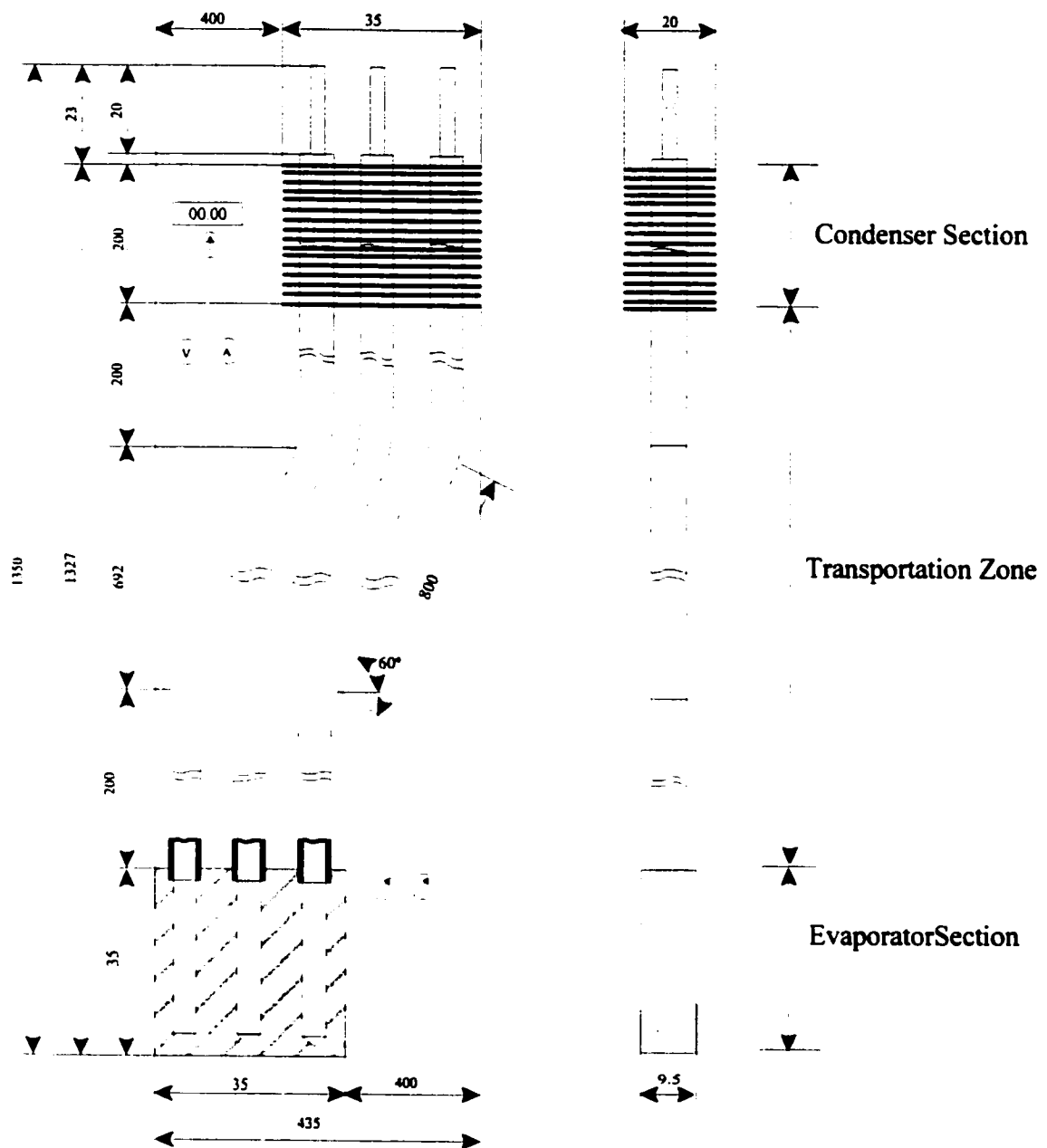


Figure 5.29. Effect of l_c on Heater Temperature Fluctuation; SSL



Unit is millimeter.

Figure 5.30 (a). Two-Phase Closed Thermosyphon (TCT) Test Assembly

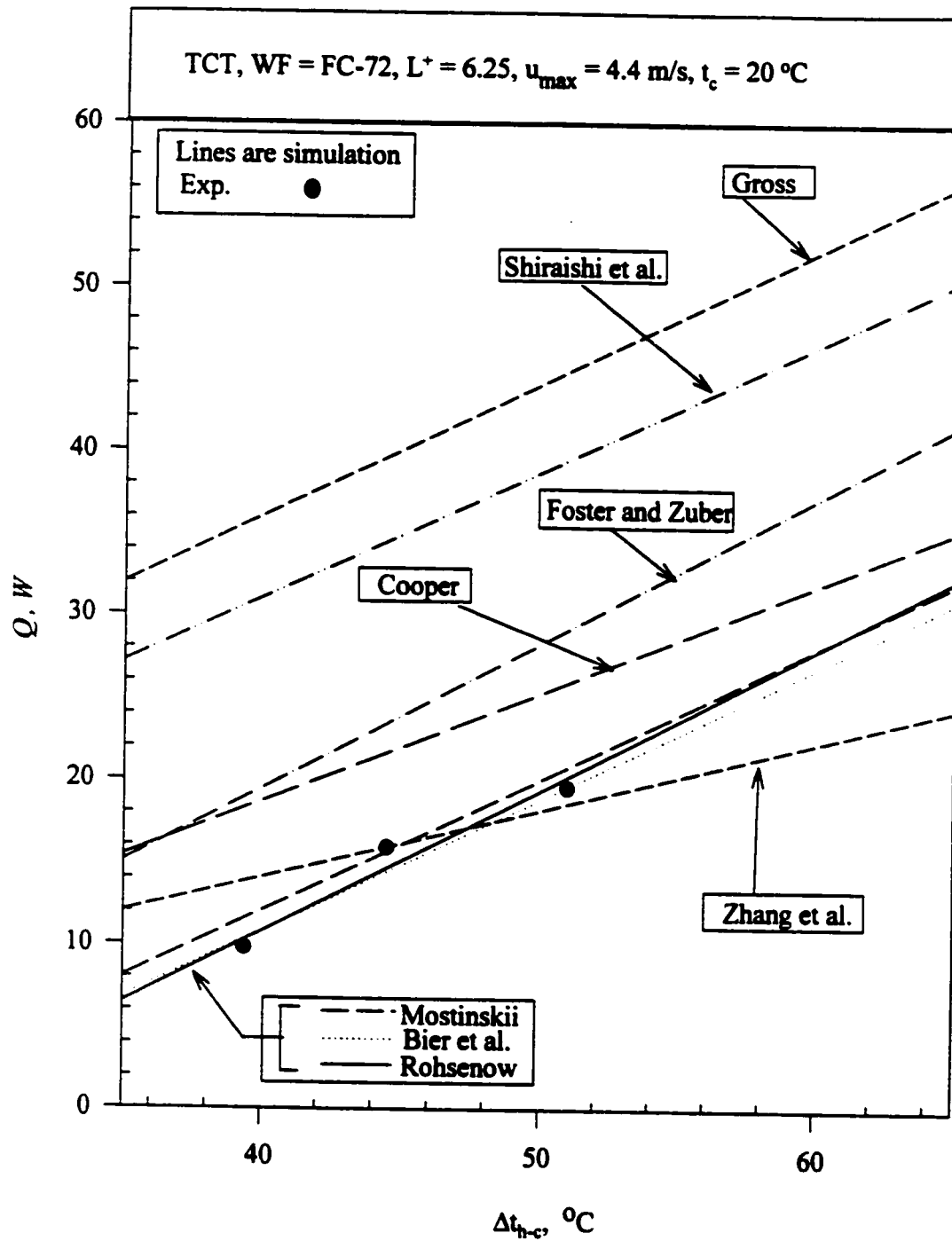


Figure 5.30 (b). Q obtained by Simulation with Various Correlations for h_{fp} ; with h_{cond} of Rohsenow; TCT

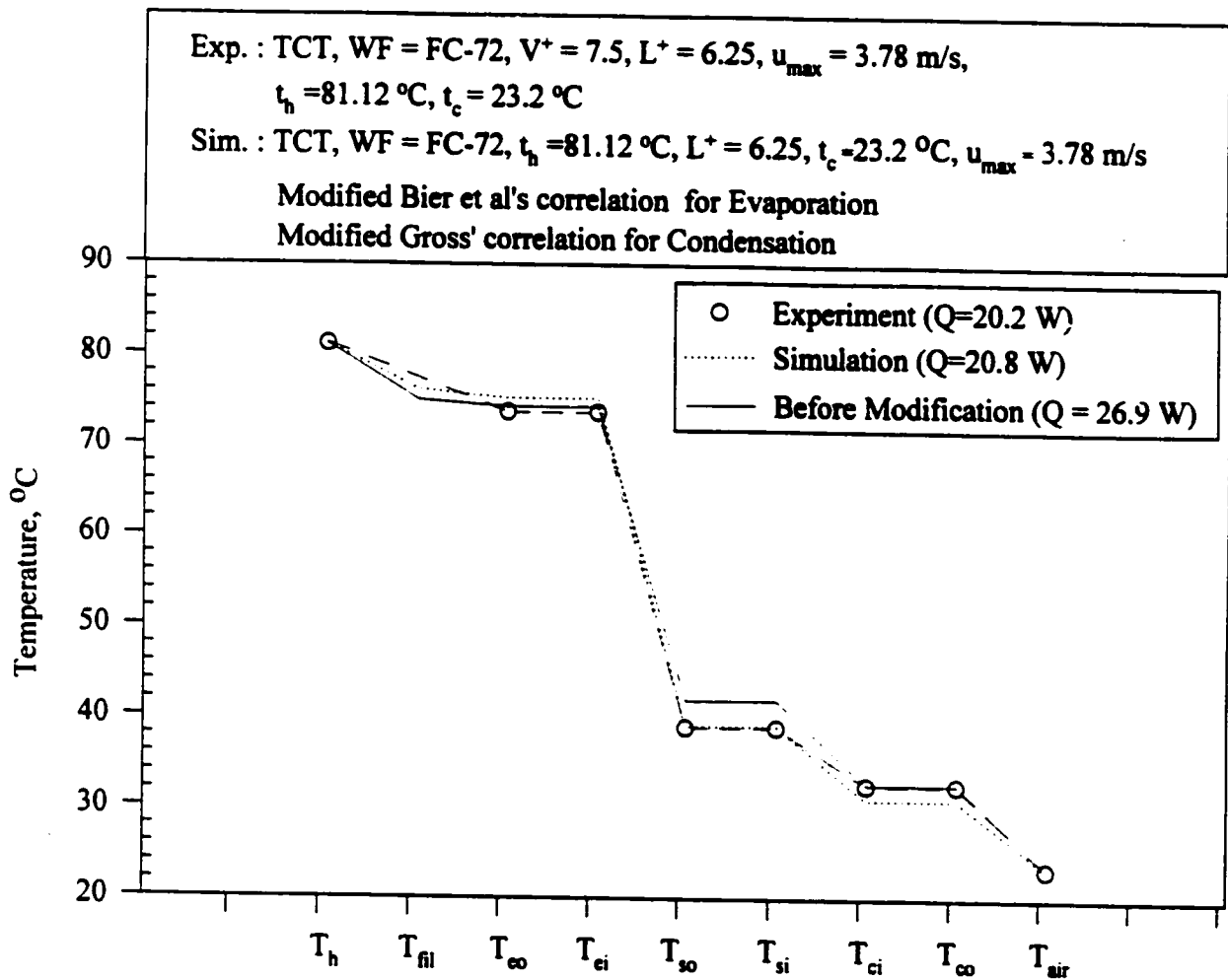


Figure 5.30 (c). Comparison between Experiment and Simulation on Temperature Distribution of TCT Assembly; TCT

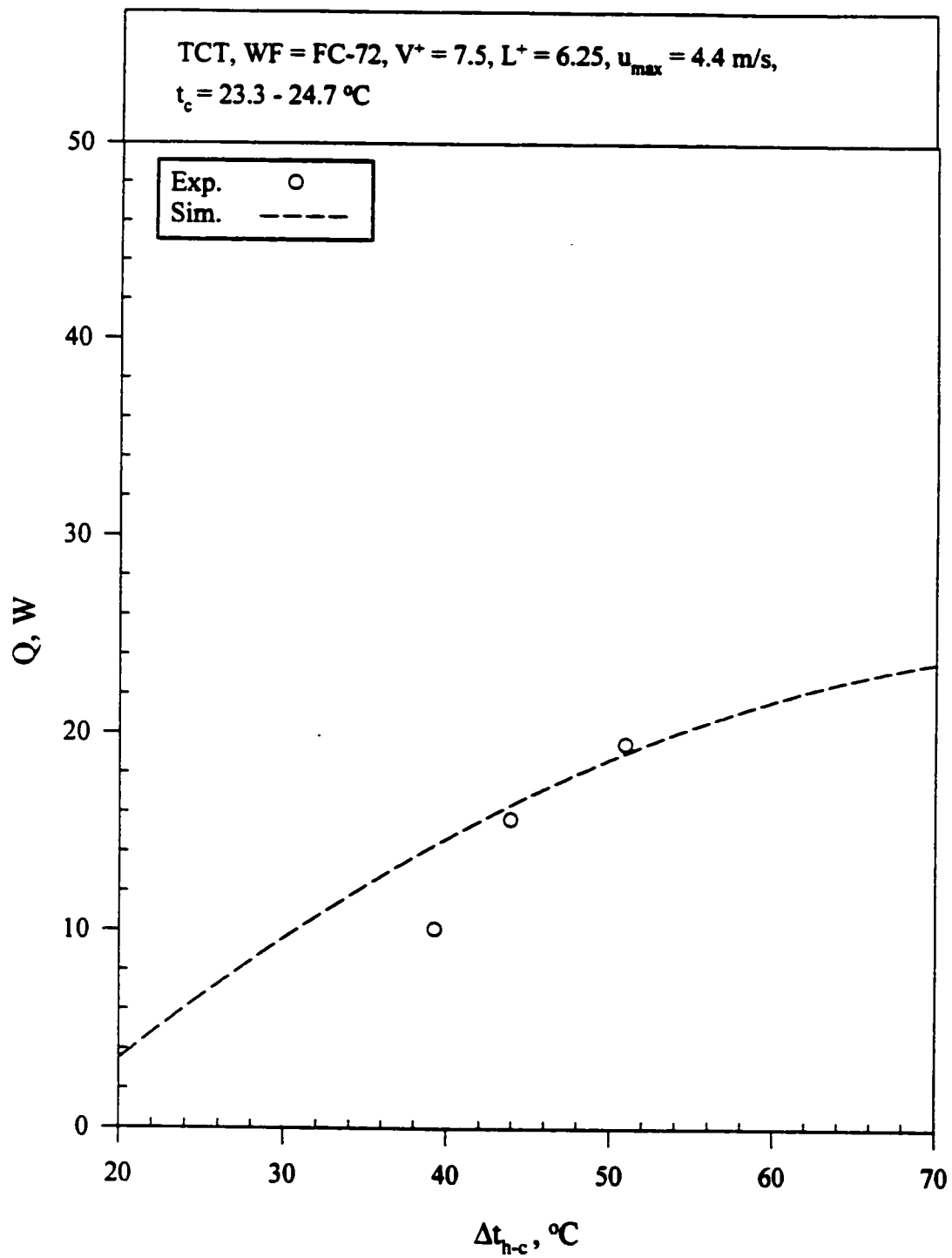


Figure 5.30 (d). Comparison between Experiment and Simulation,
Effect of Δt_{h-c} on Q; TCT

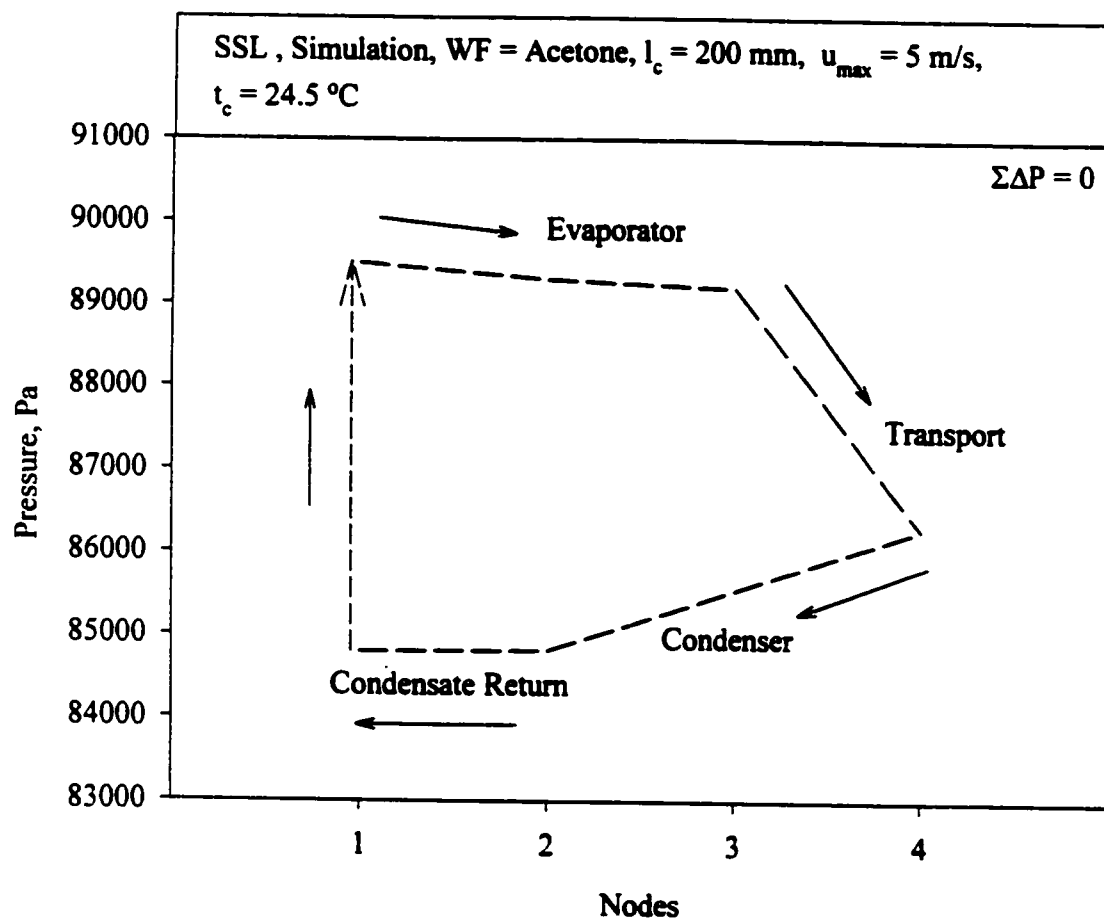


Figure 5.31. Pressure Variation of Loop; SSL

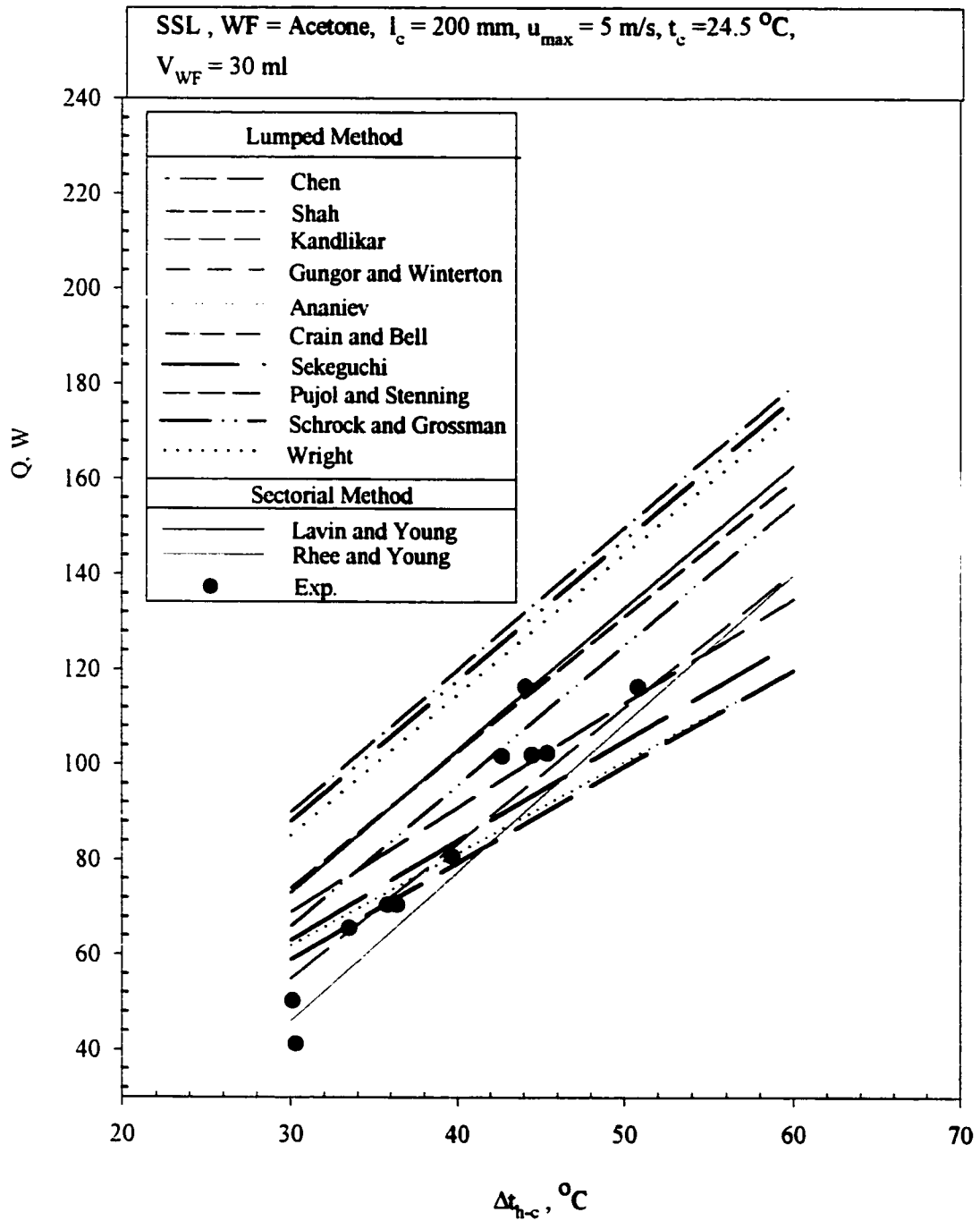


Figure 5.32 (a). Comparison between Experiment and Simulation ; SSL

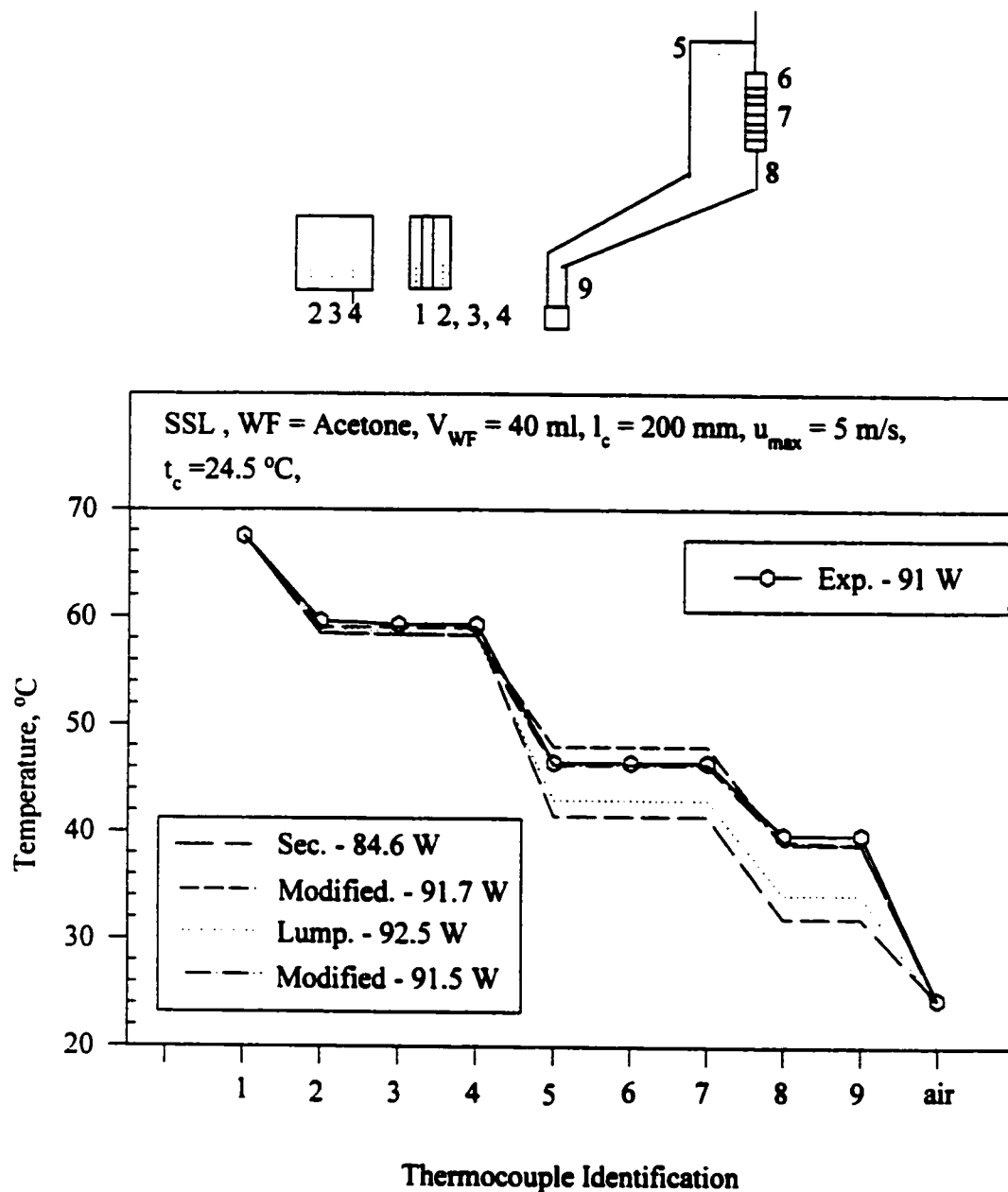


Figure 5.32 (b). Comparison of Temperature Distribution, Simulation and Experiment; SSL

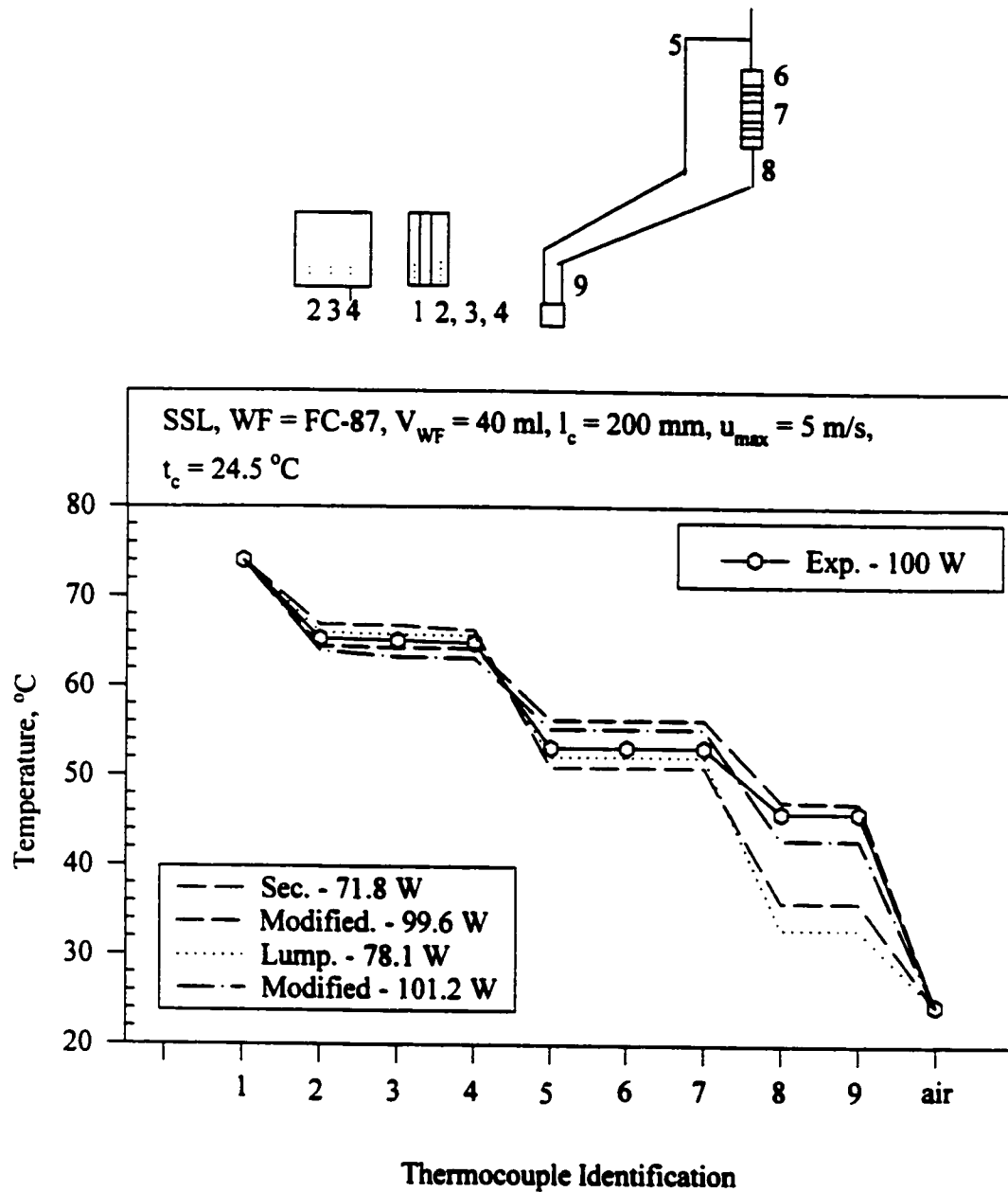


Figure 5.32 (c). Comparison of Temperature Distribution, Simulation and Experiment; SSL

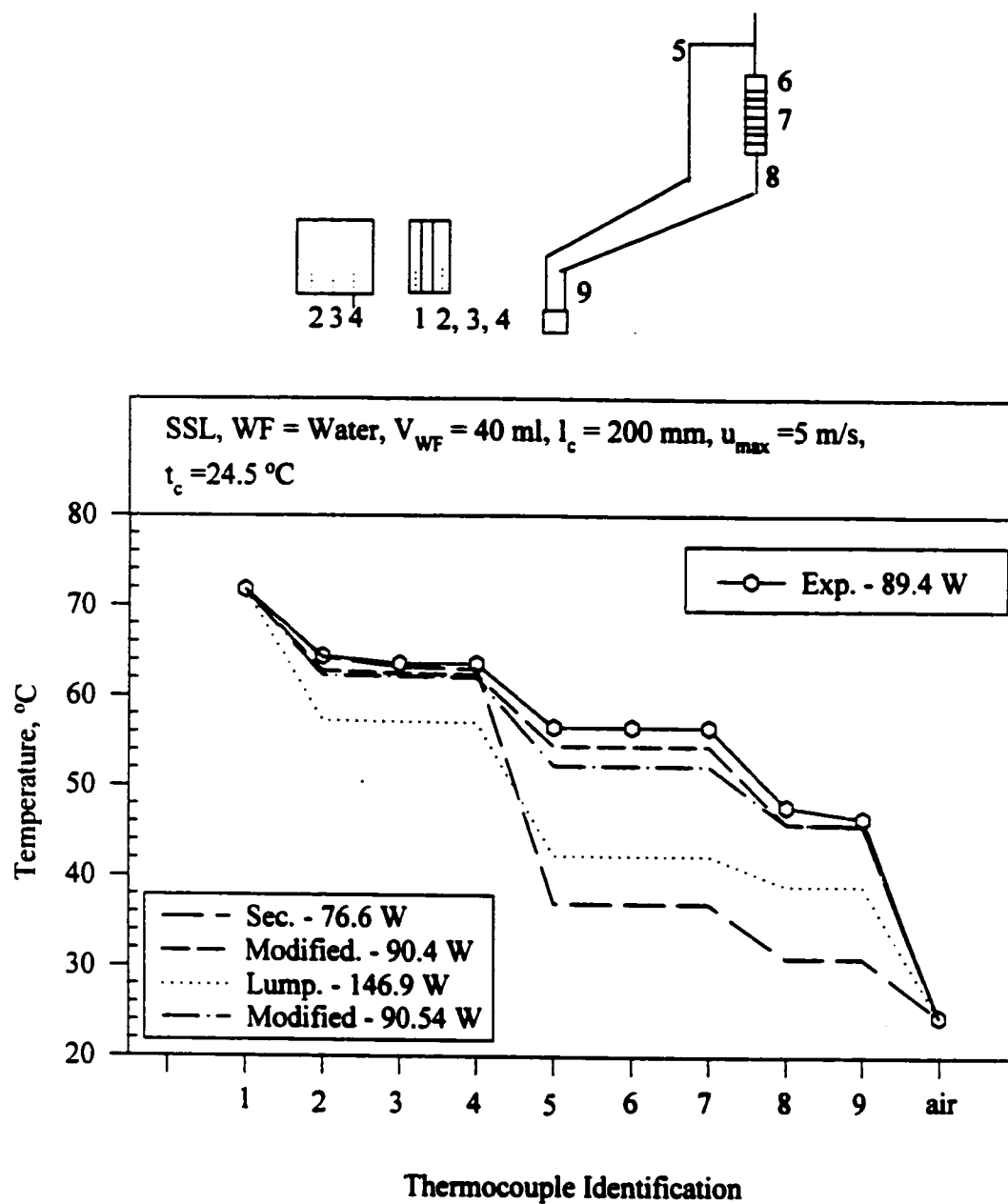


Figure 5.32 (d). Comparison of Temperature Distribution, Simulation and Experiment; SSL

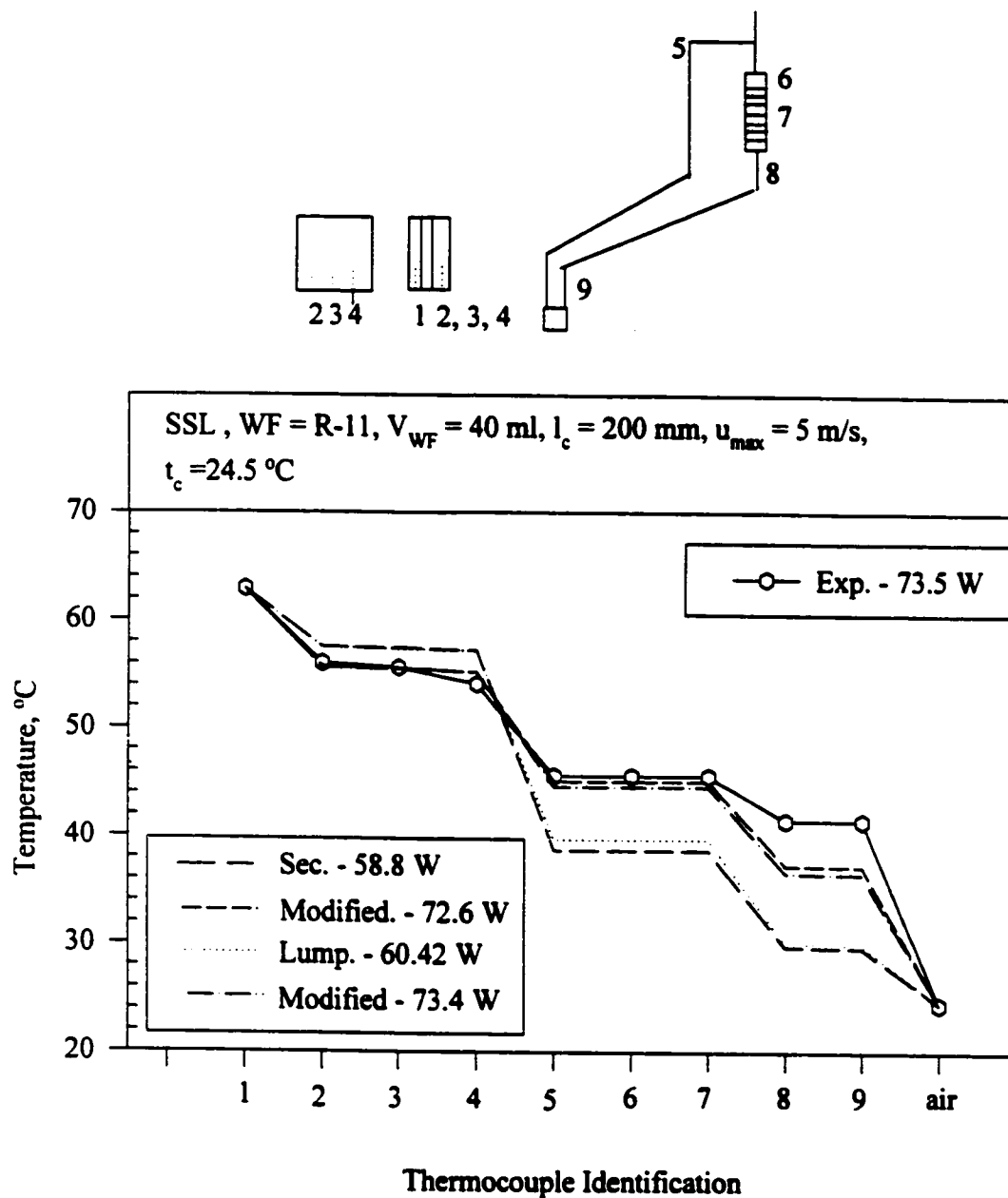


Figure 5.32 (e). Comparison of Temperature Distribution, Simulation and Experiment; SSL

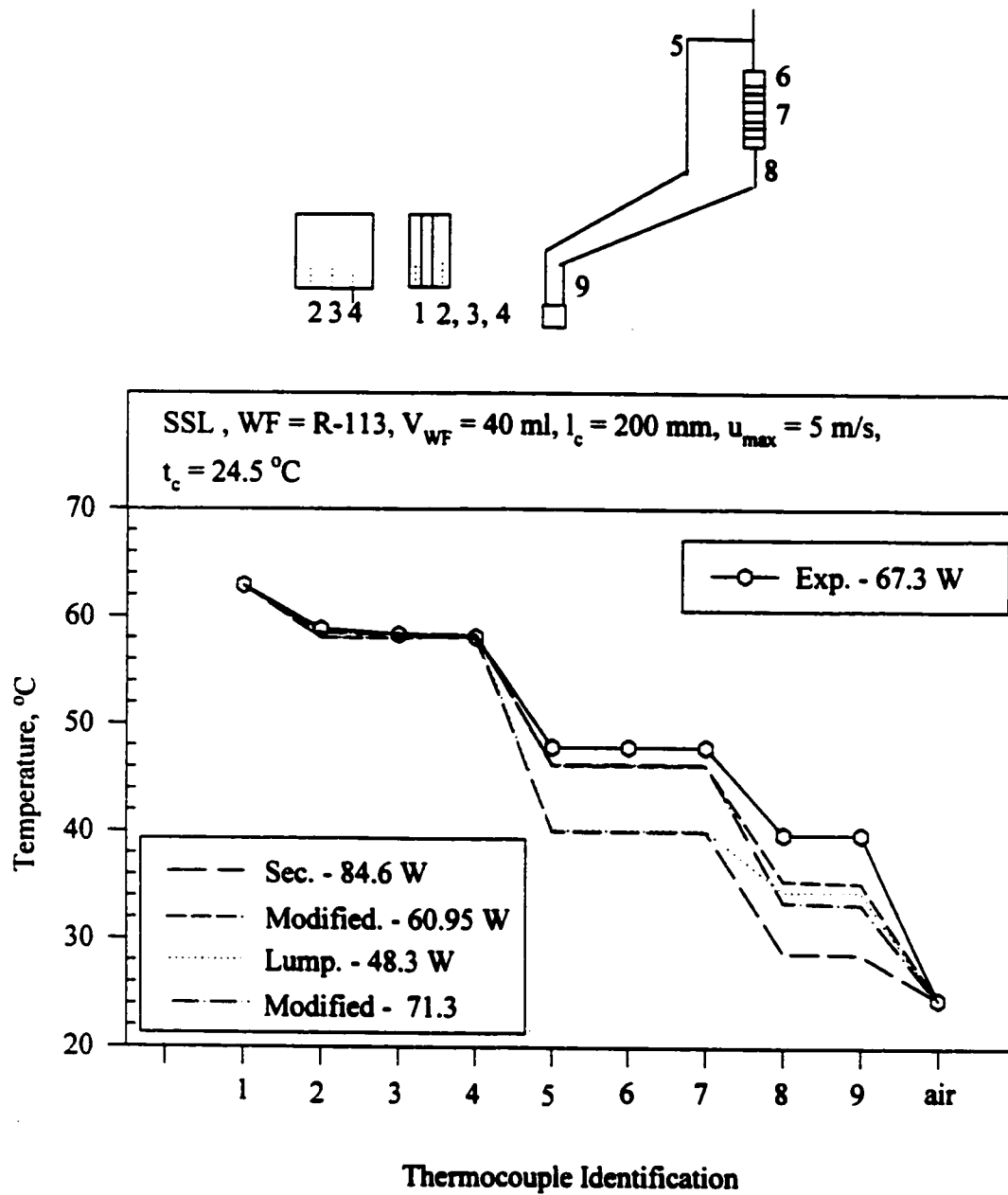


Figure 5.32 (f). Comparison of Temperature Distribution, Simulation and Experiment; SSL

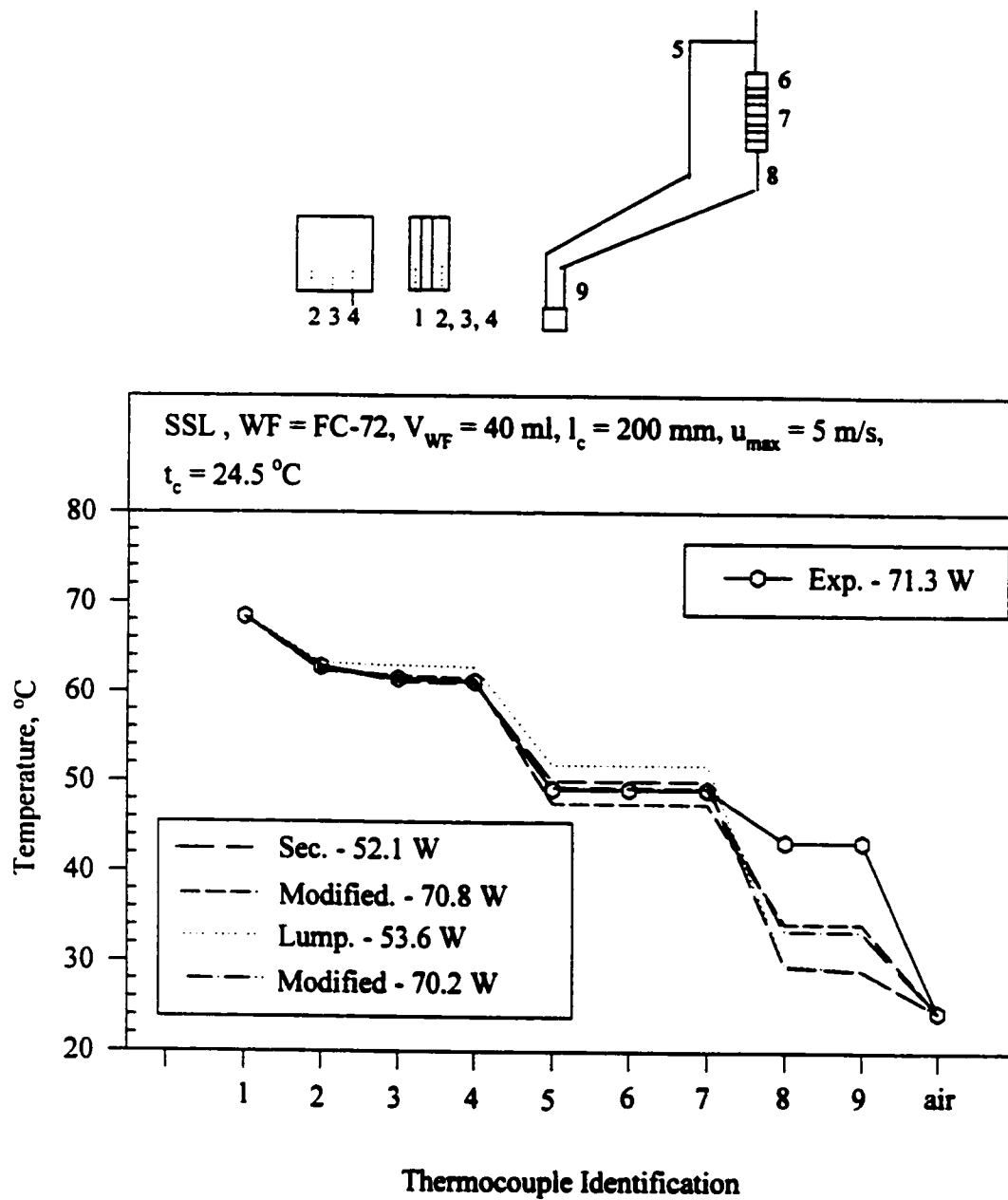


Figure 5.32 (g). Comparison of Temperature Distribution, Simulation and Experiment; SSL

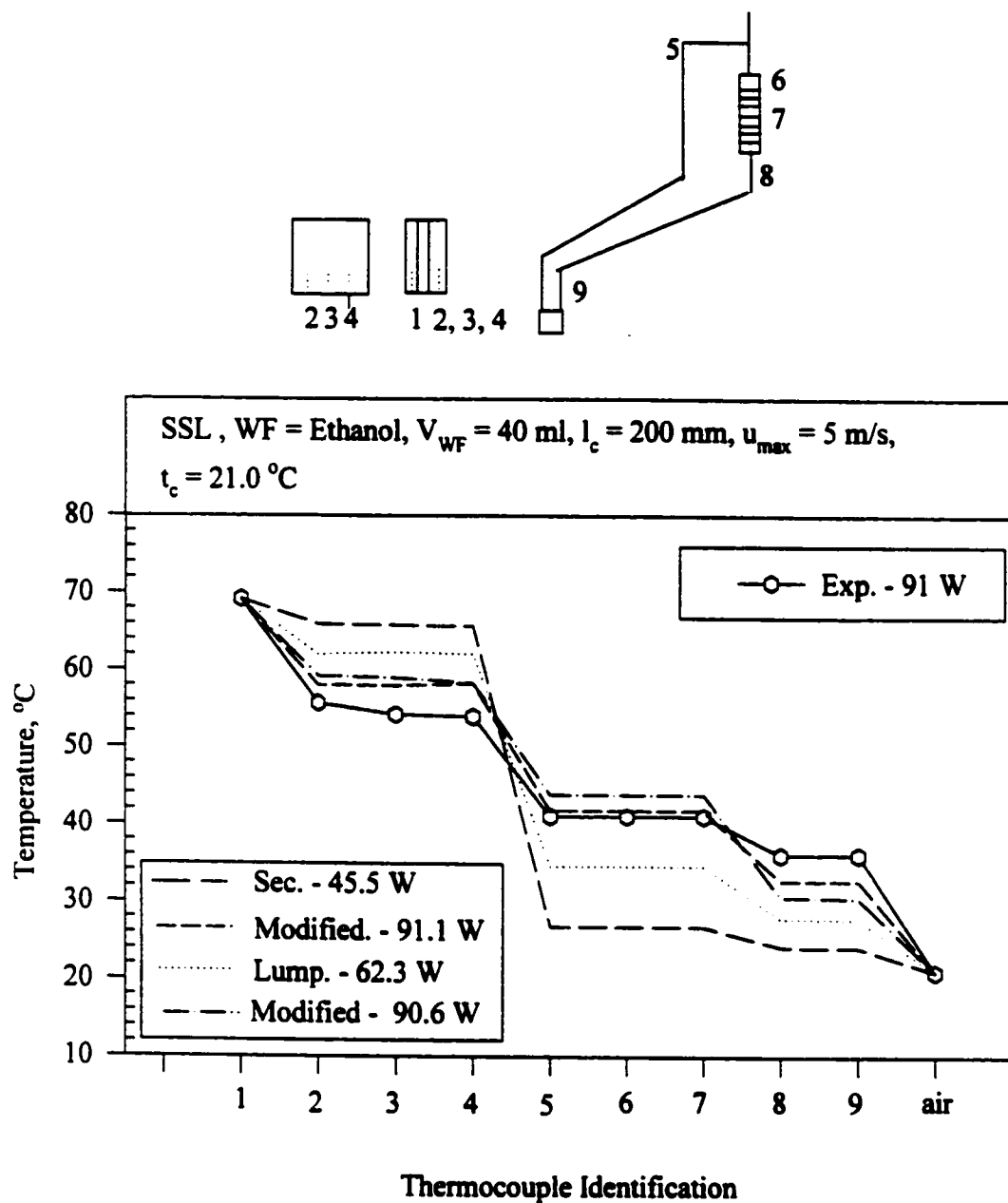


Figure 5.32 (h). Comparison of Temperature Distribution, Simulation and Experiment; SSL

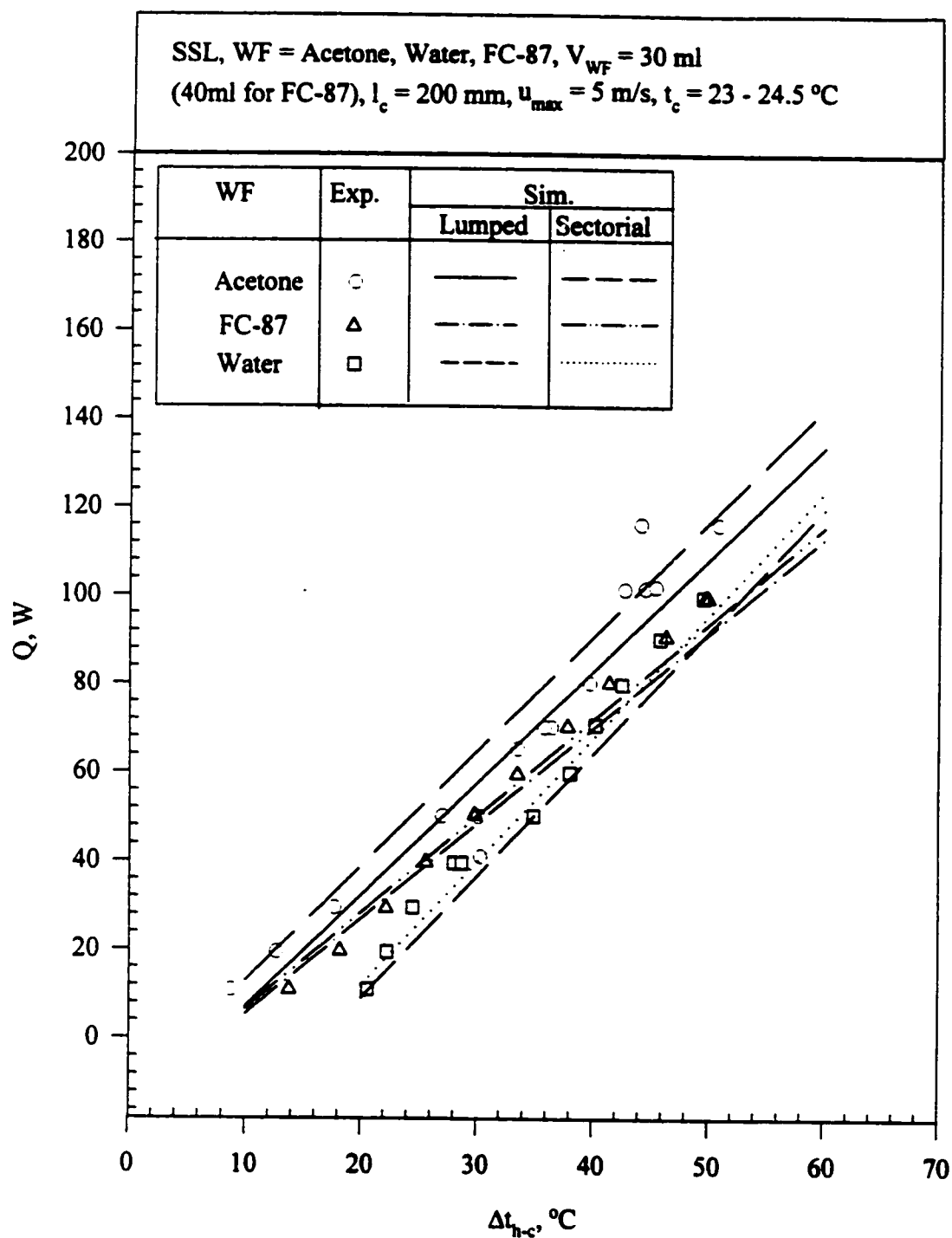


Figure 5.32 (i). Comparison Between Experiments and Simulation; SSL

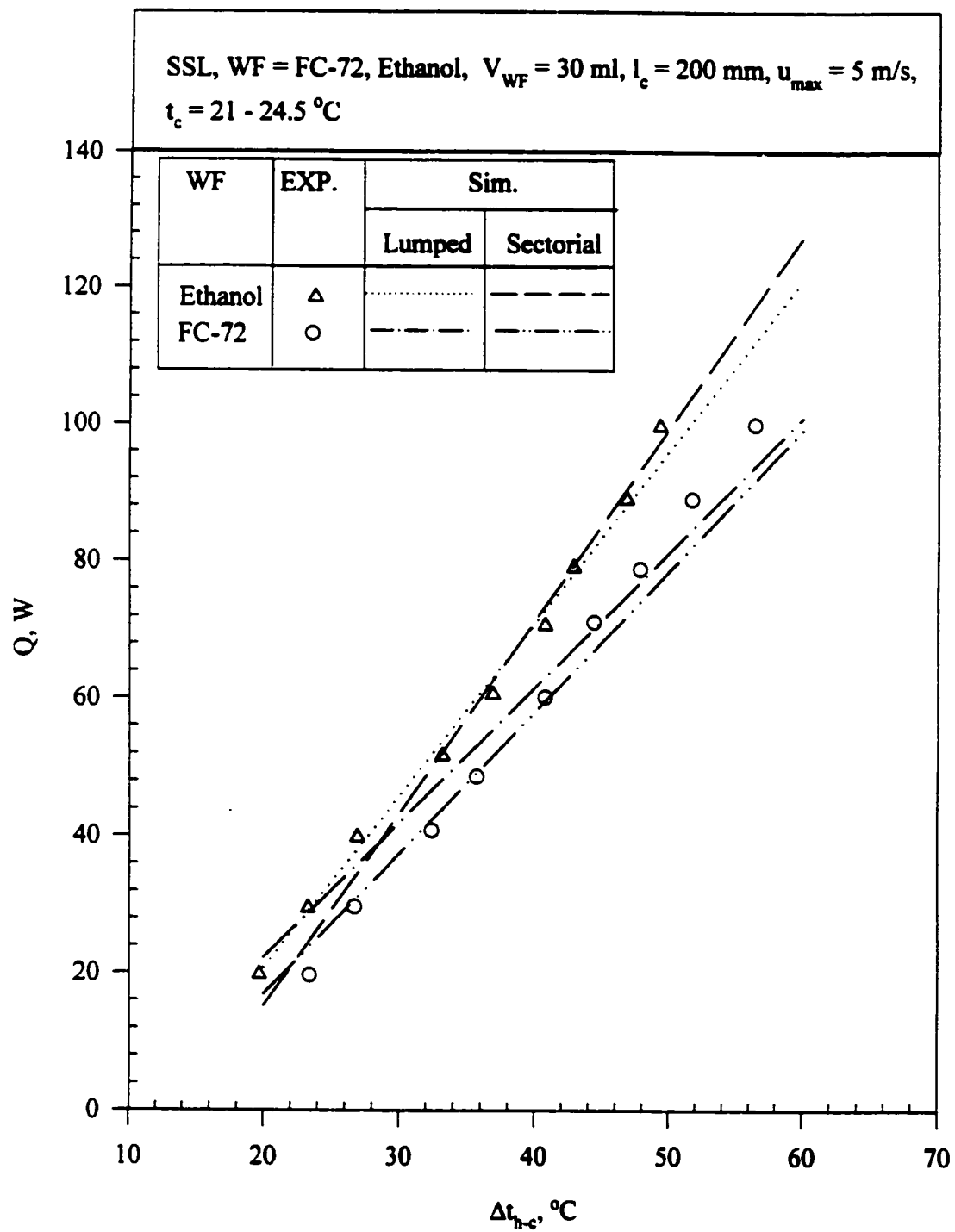


Figure 5.32 (j). Comparison between Experiment and Simulation;
 Forced Convection Mode, SSL

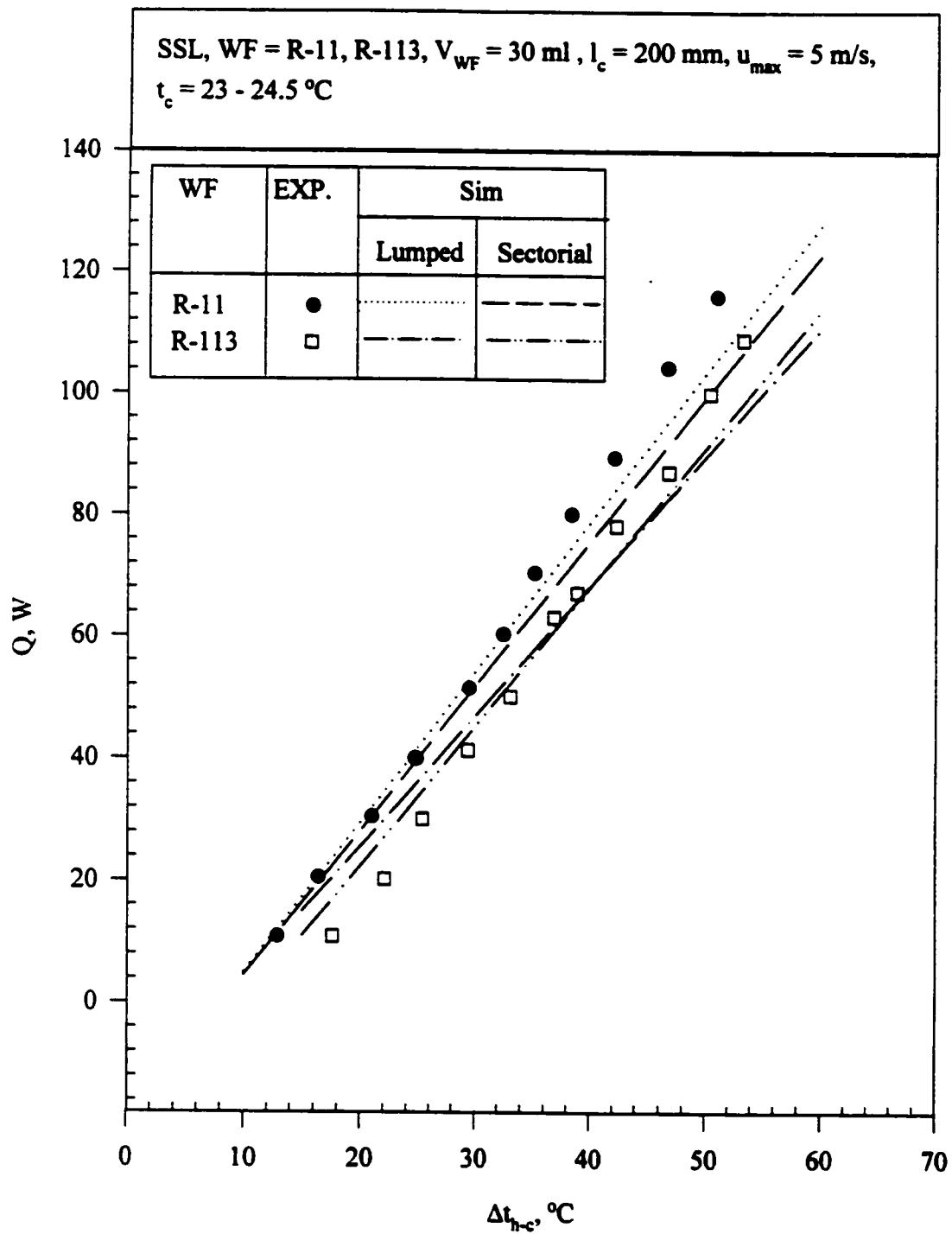


Figure 5.32 (k). Comparison between Experiment and Simulation;
 Forced Convection Mode, SSL

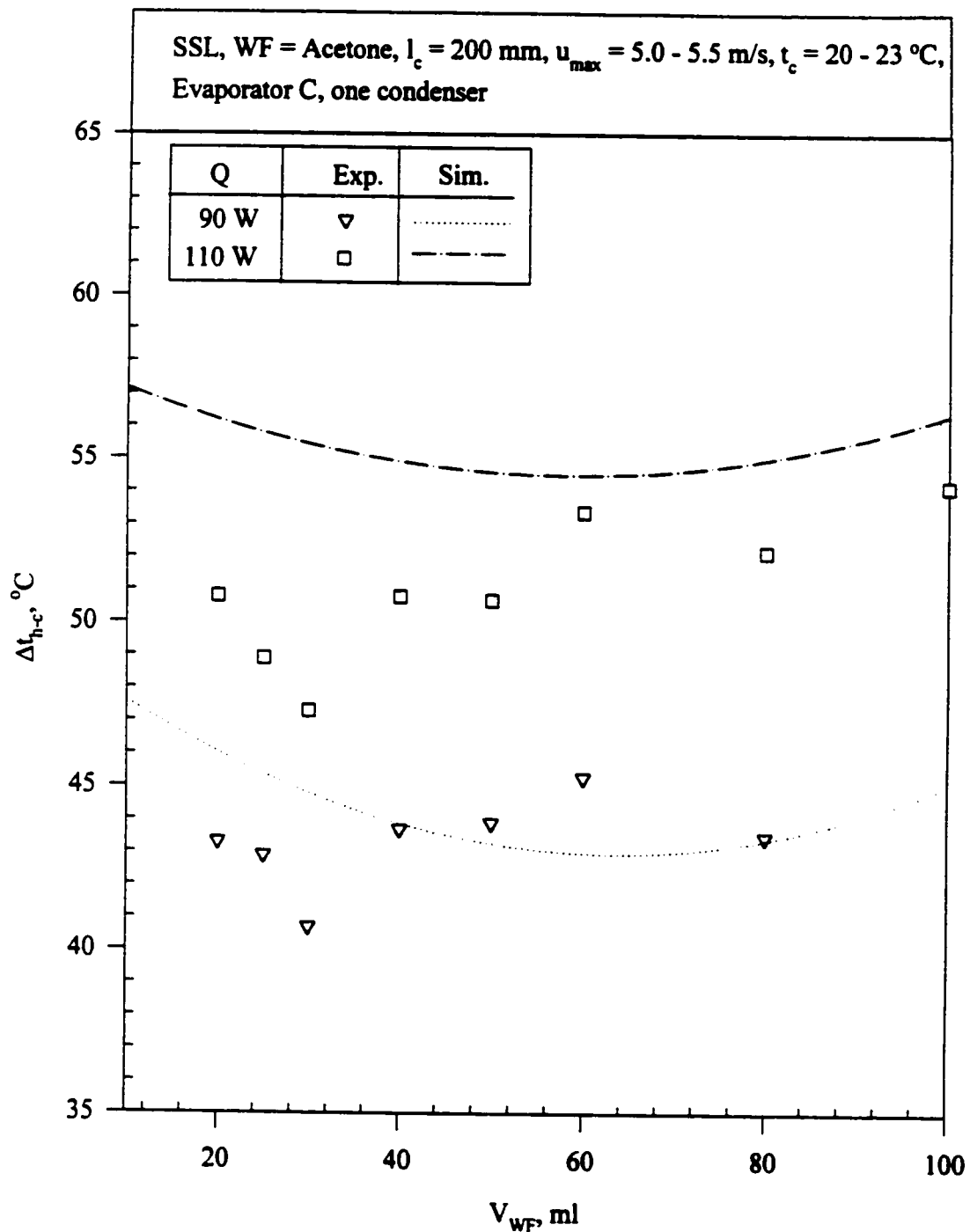


Figure 5.32 (1). Comparison between Experiment and Simulation; SSL
Effect of V_{WF} on Δt_{h-c} , Forced Convection Mode
Sectorial Method : $h_{tp} = \text{Rhee and Young}$, $h_{cond} = \text{Nusselt}$ and,
 $h_{conv} = \text{Knudson}$

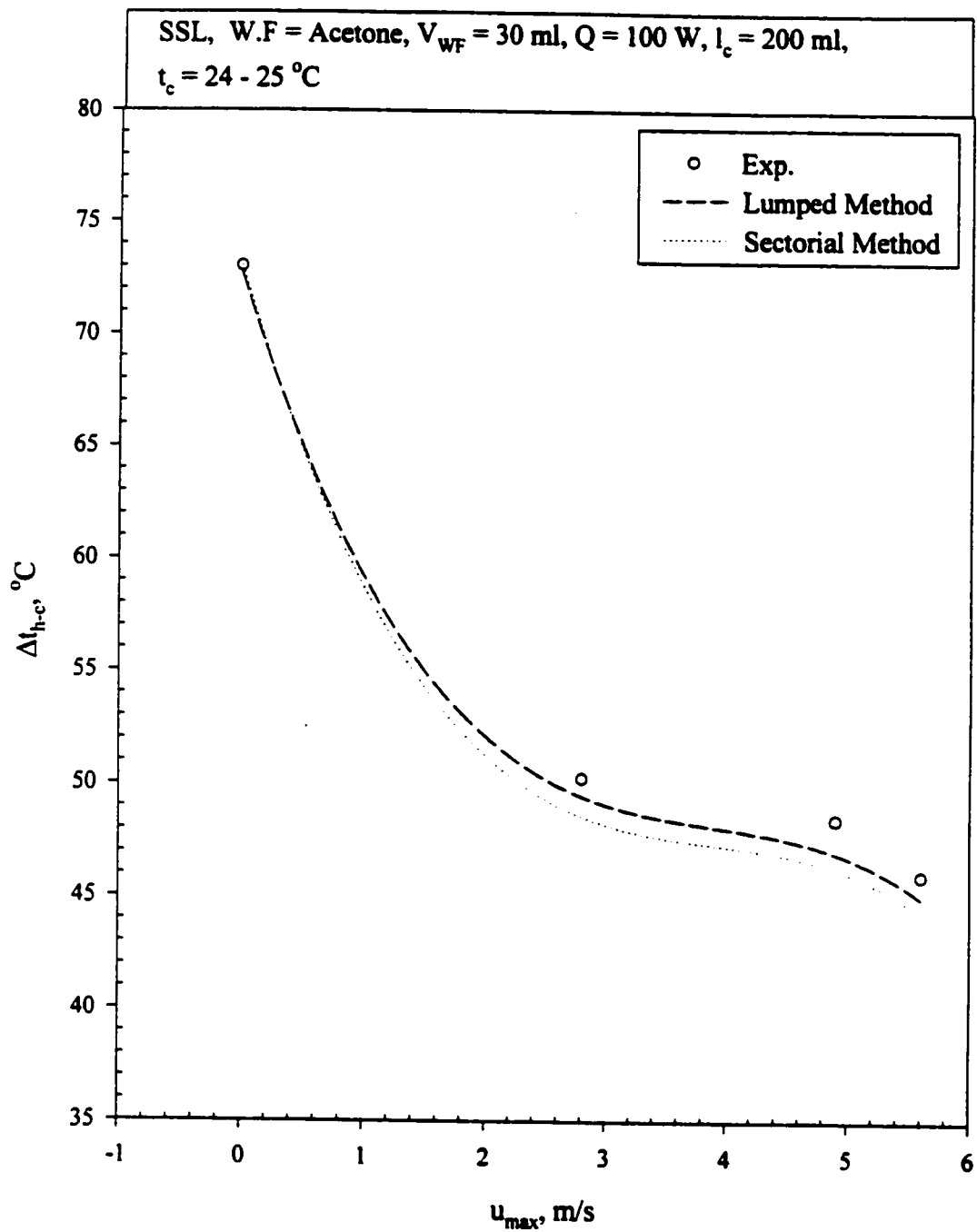


Figure 5.32 (m). Comparison between Experiment and Simulation;
 Effect of u_{max} on Δt_{h-c} , SSL

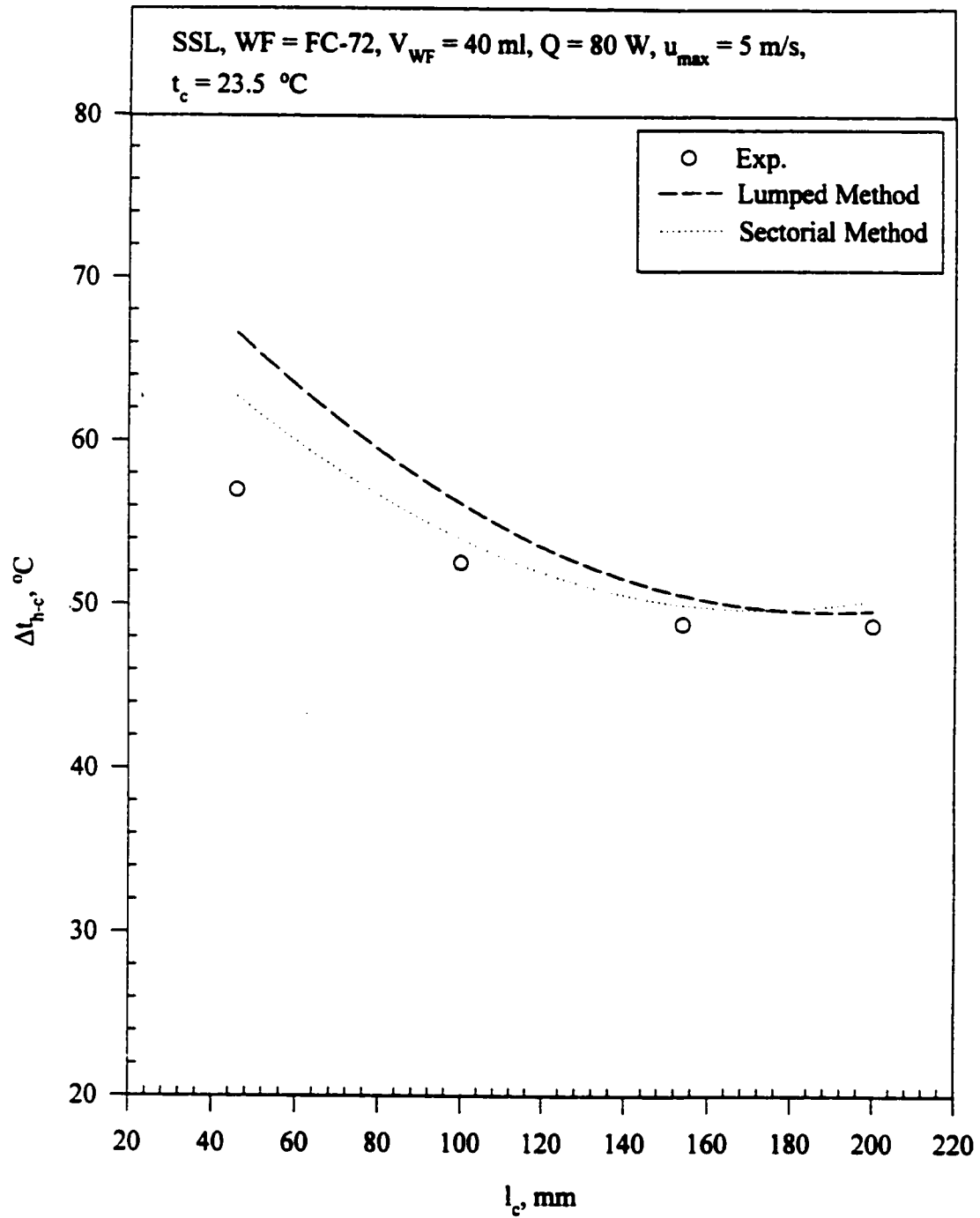


Figure 5.32 (n). Comparison between Experiment and Simulation
 Effect of l_c on Δt_{h-c} ; SSL

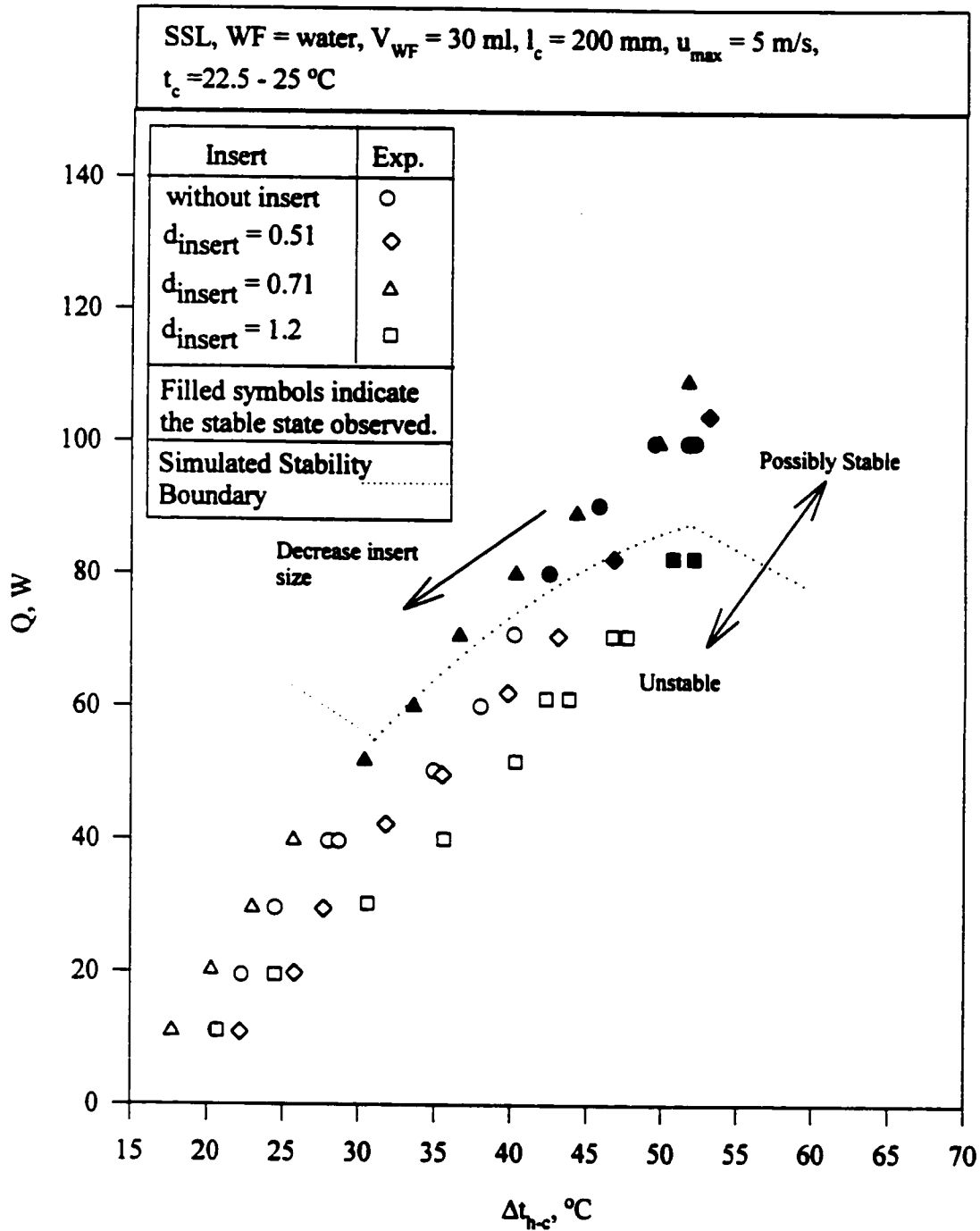


Figure 5.32 (o). Comparison Between Experiment and Simulation, Stability Boundary
 Effect of Insert; SSL
 Simulation: Sectorial Method

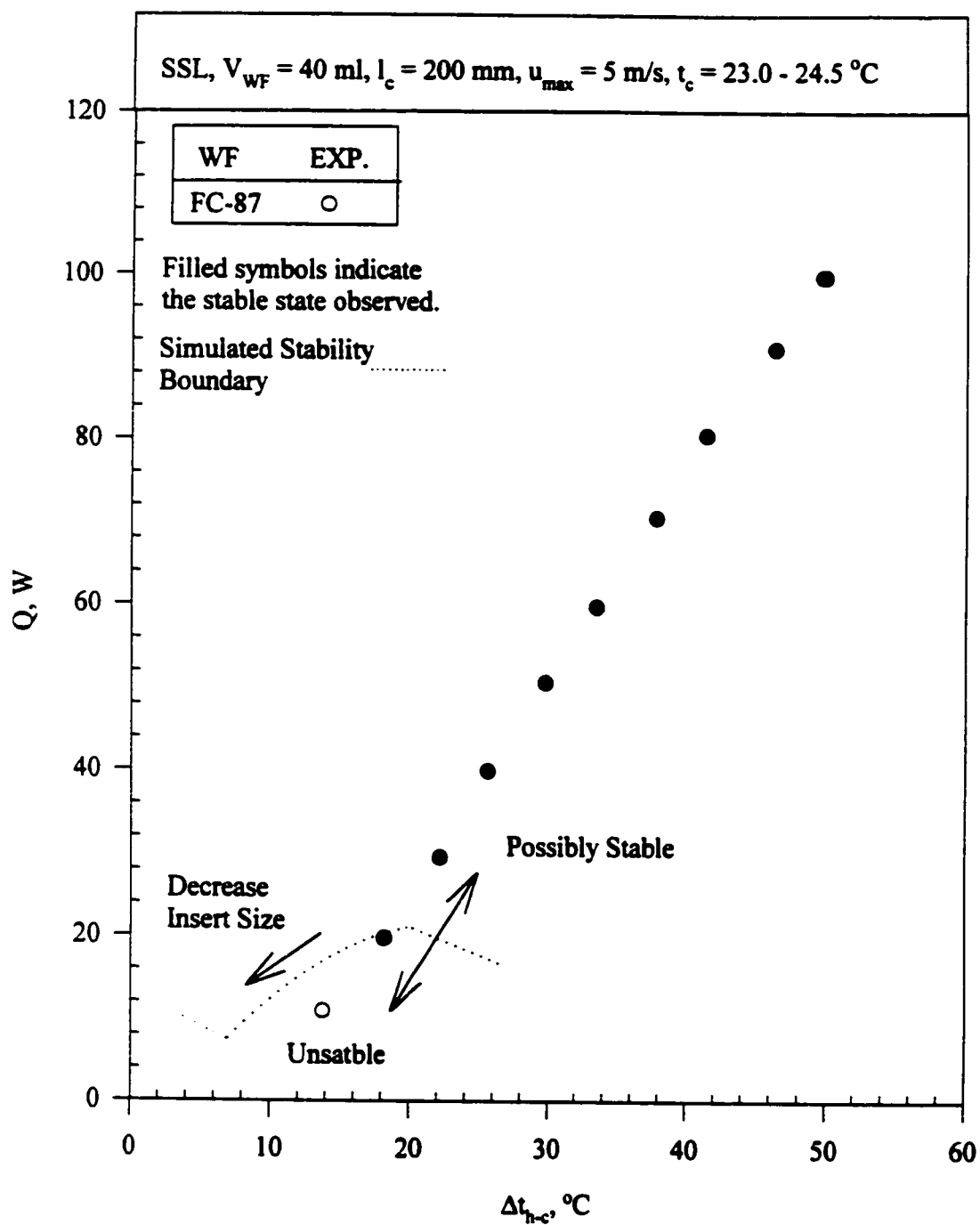


Figure 5.32 (p).

Comparison Between Experiment and Simulation, Stability Boundary
Effect of Insert; SSL
Simulation: Sectorial Method

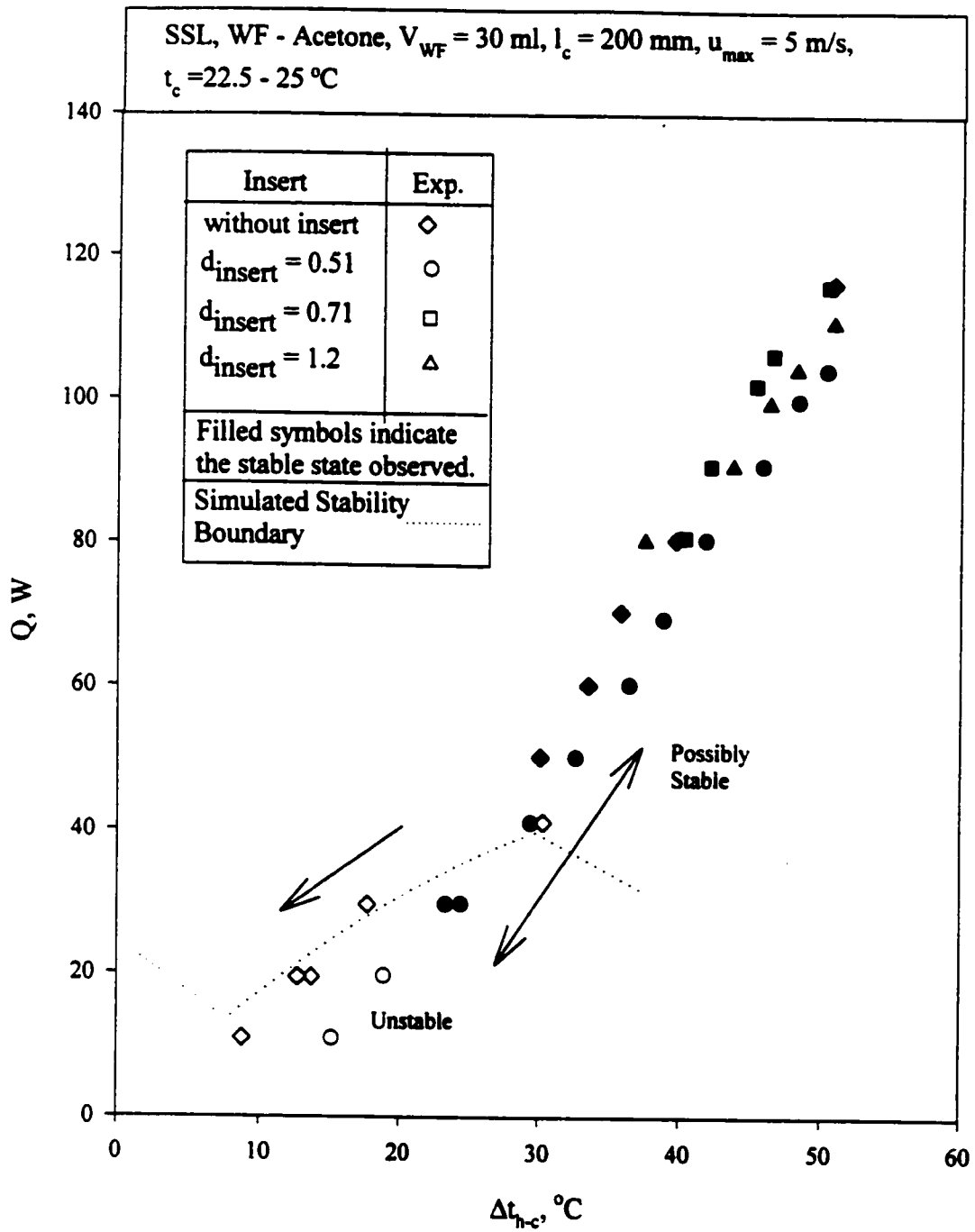


Figure 5.32 (q). Comparison Between Experiment and Simulation, Stability Boundary Effect of Insert; WF = Acetone, SSL
 Simulation: Sectorial Method

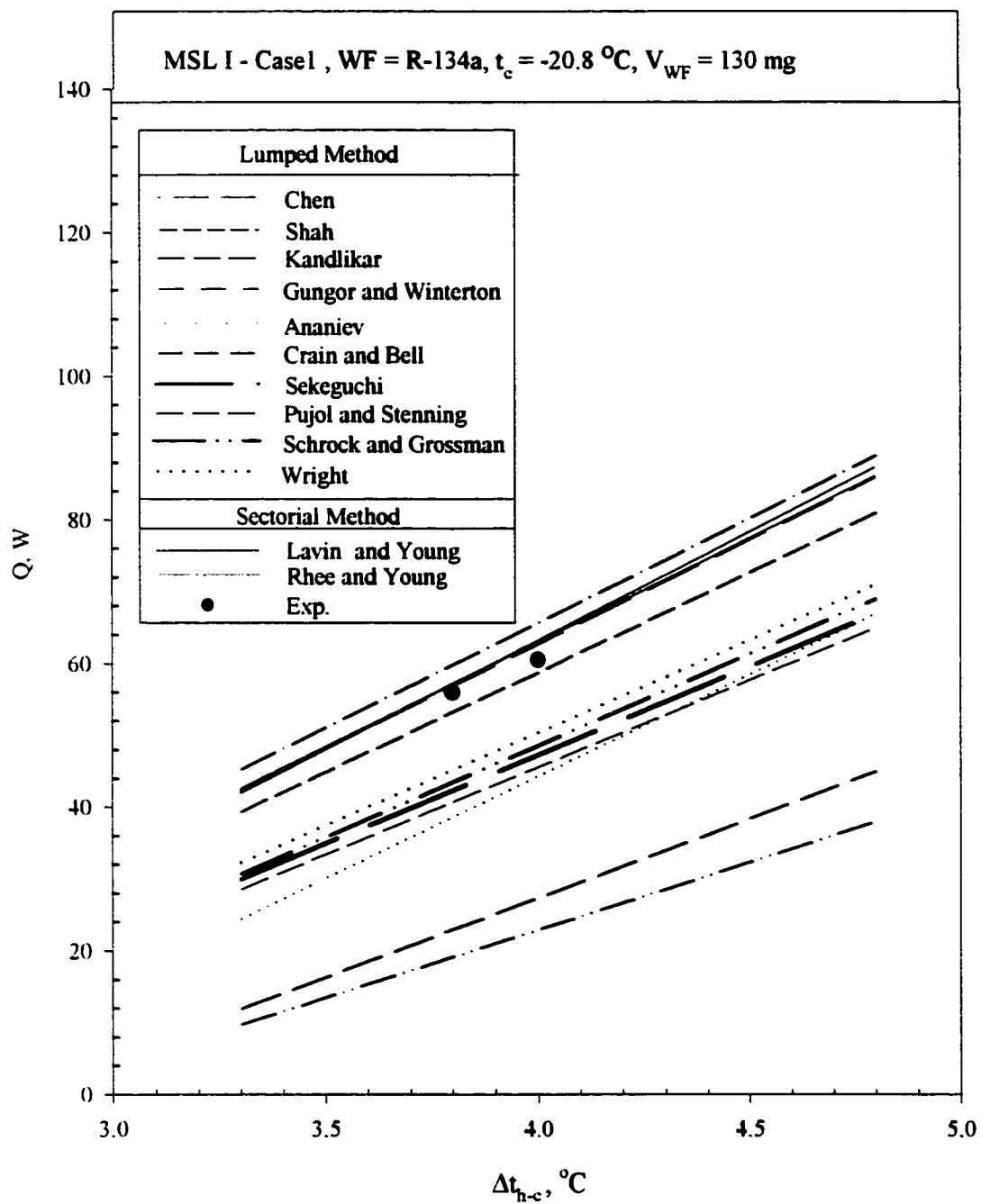


Figure 5.33 (a). Comparison between Experiment and Simulation; MSL I

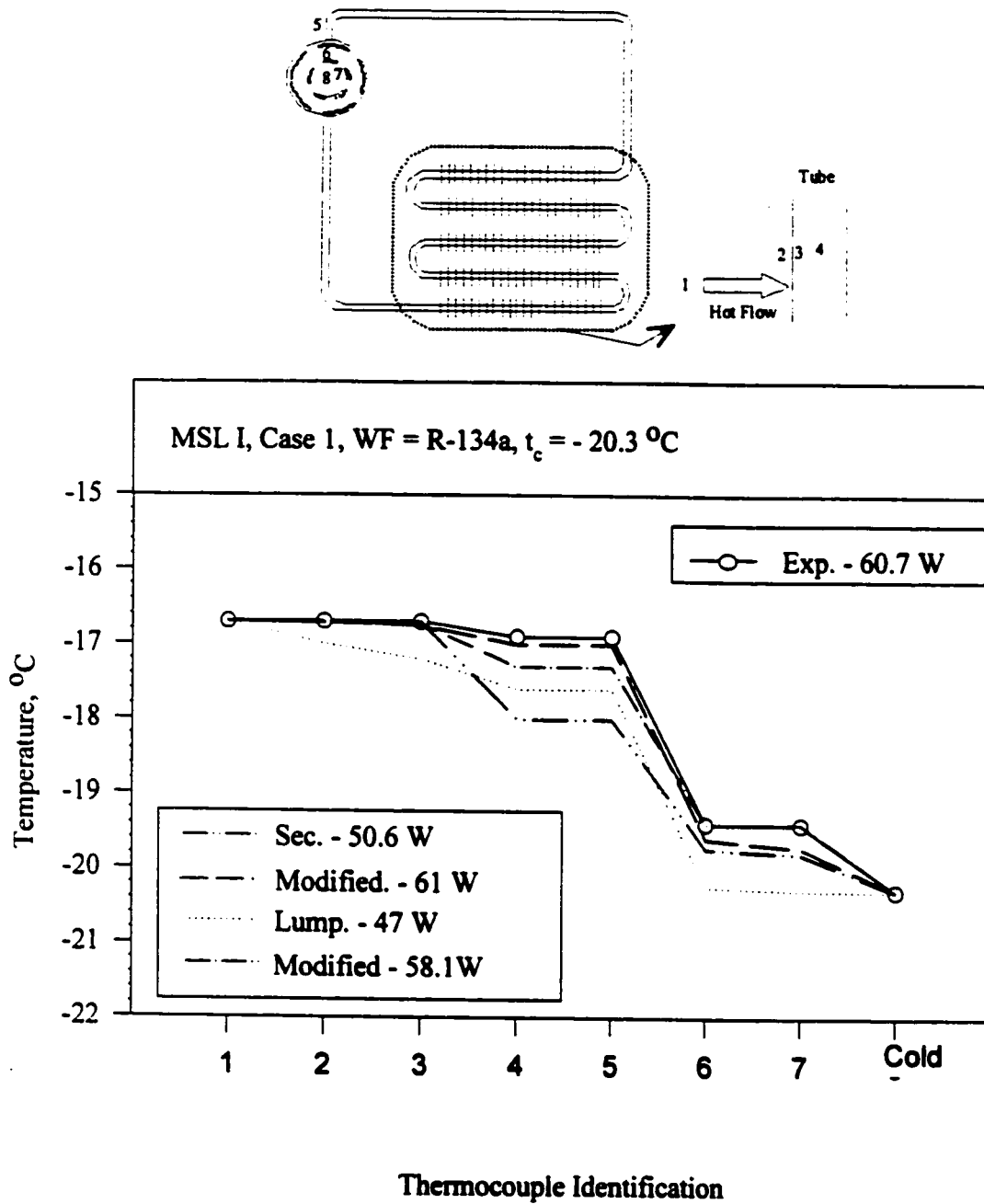


Figure 5.33 (b). Comparison of Temperature Distribution, Simulation and Experiment; Case 1, MSL I

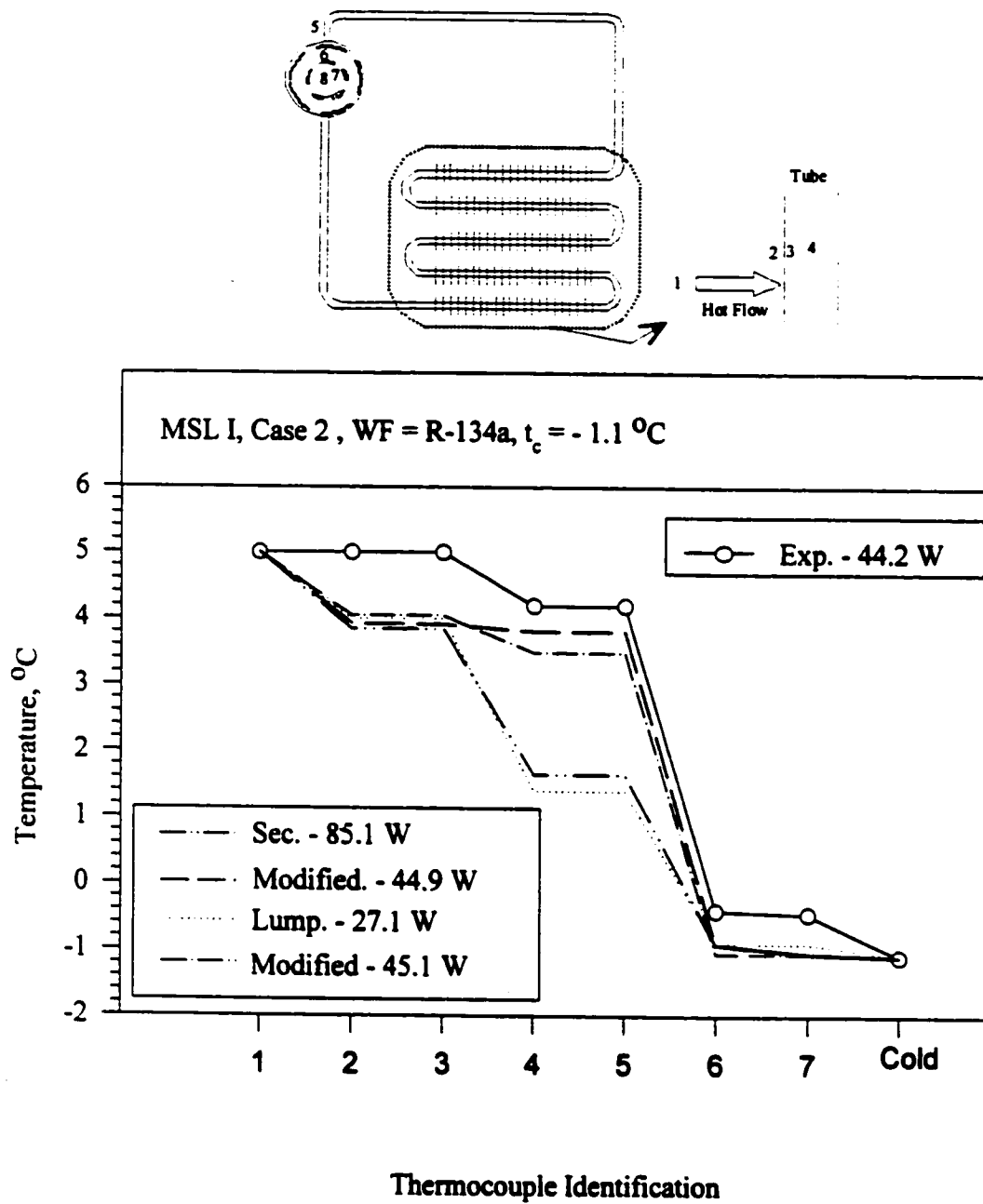


Figure 5.33 (c). Comparison of Temperature Distribution, Simulation and Experiment; Case 2, MSL I

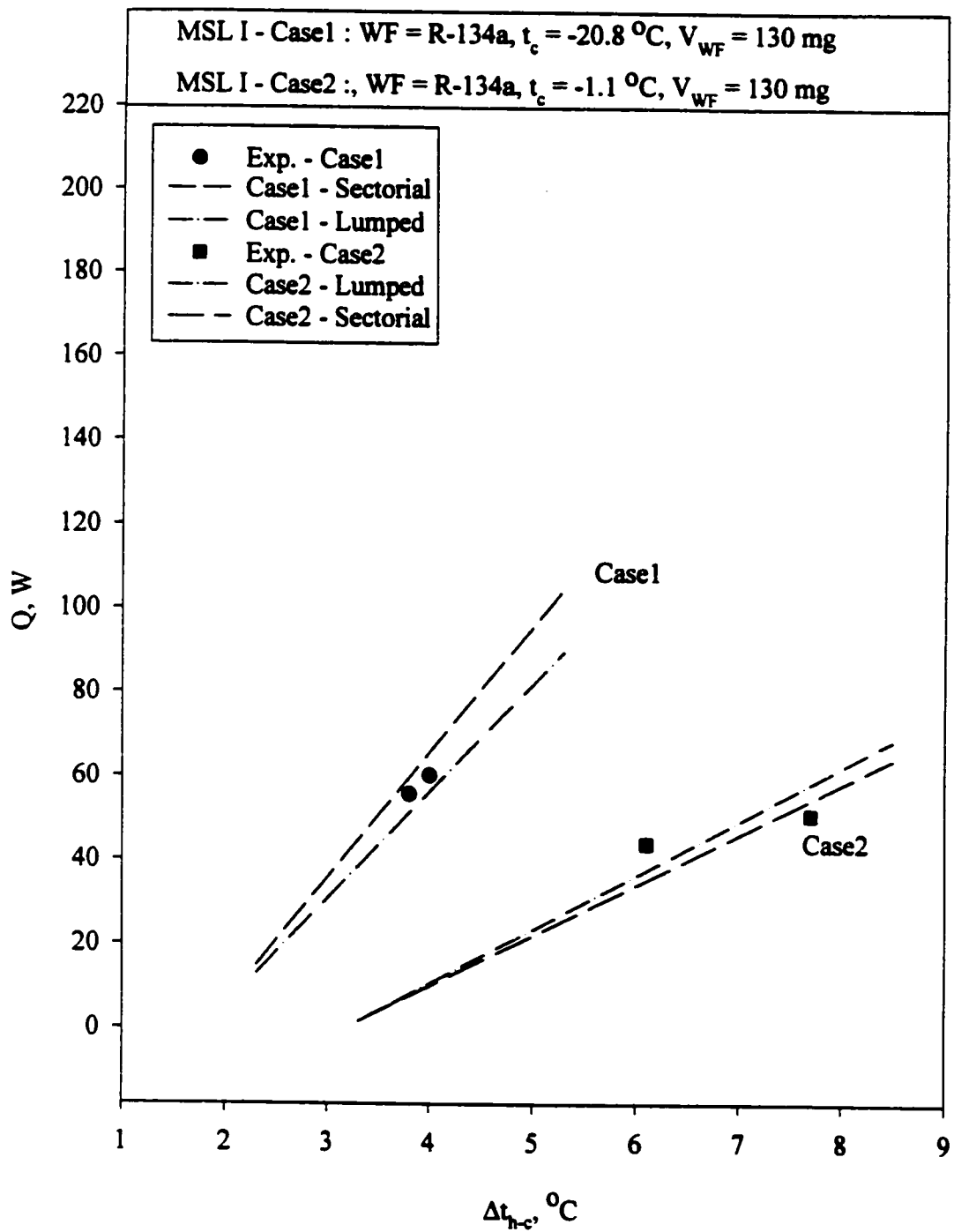


Figure 5.33 (d). Comparison between Experiments and Simulations, MSL I

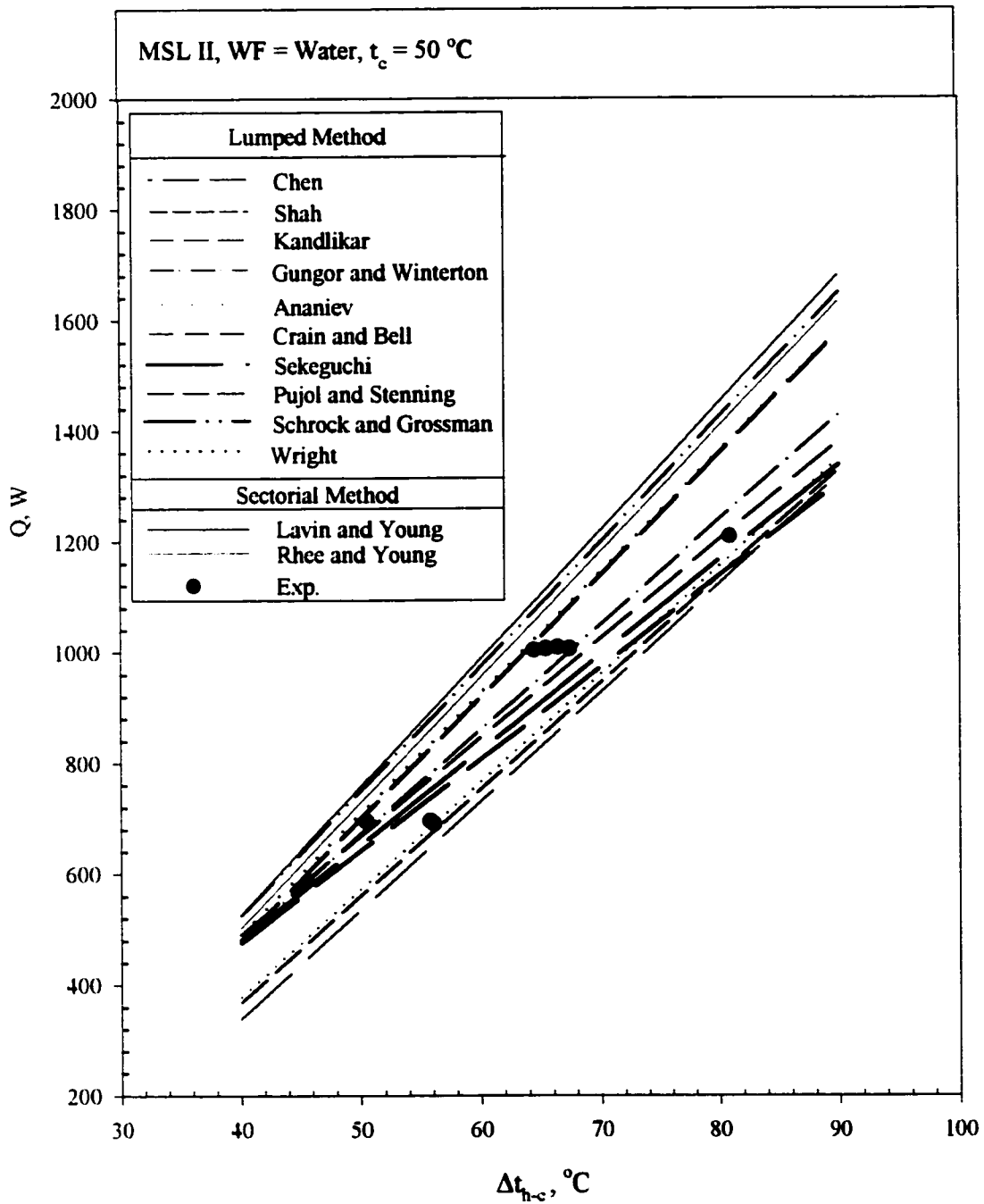


Figure 5.34 (a). Comparison between Experiment and Simulation; MSL II

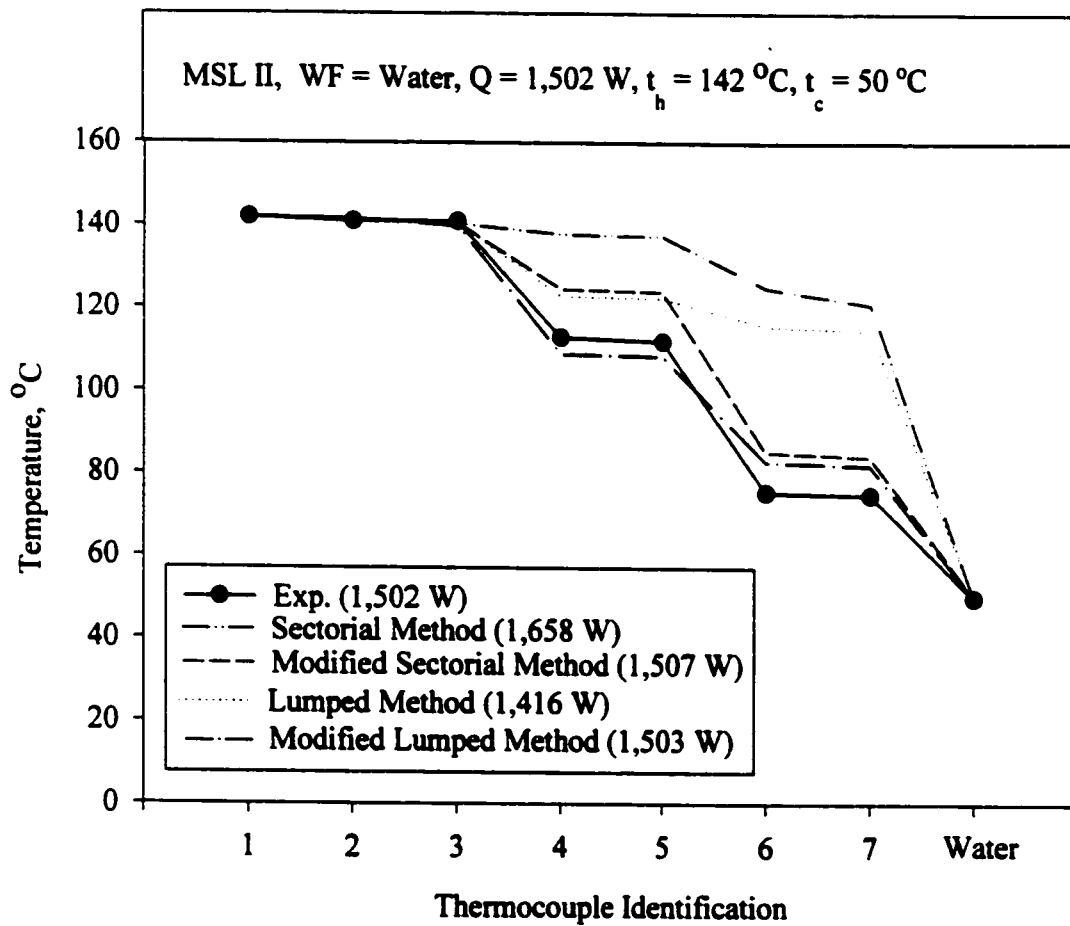
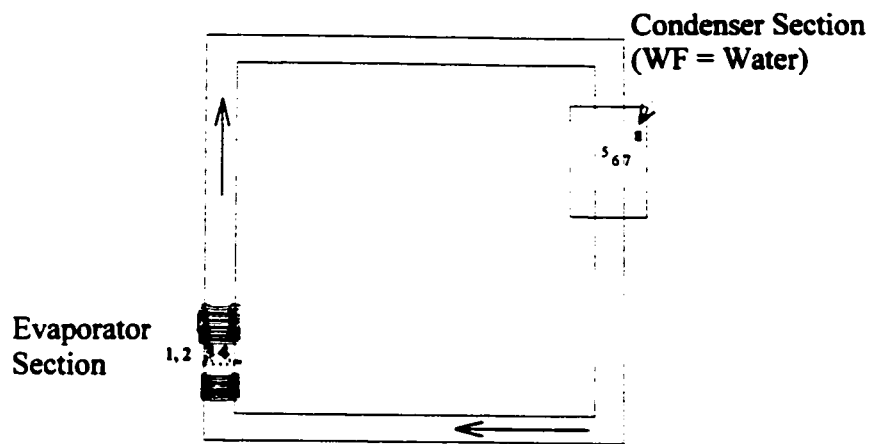


Figure 5.34 (b). Comparison between Experiment and Simulation, Temperature Distribution; MSL II

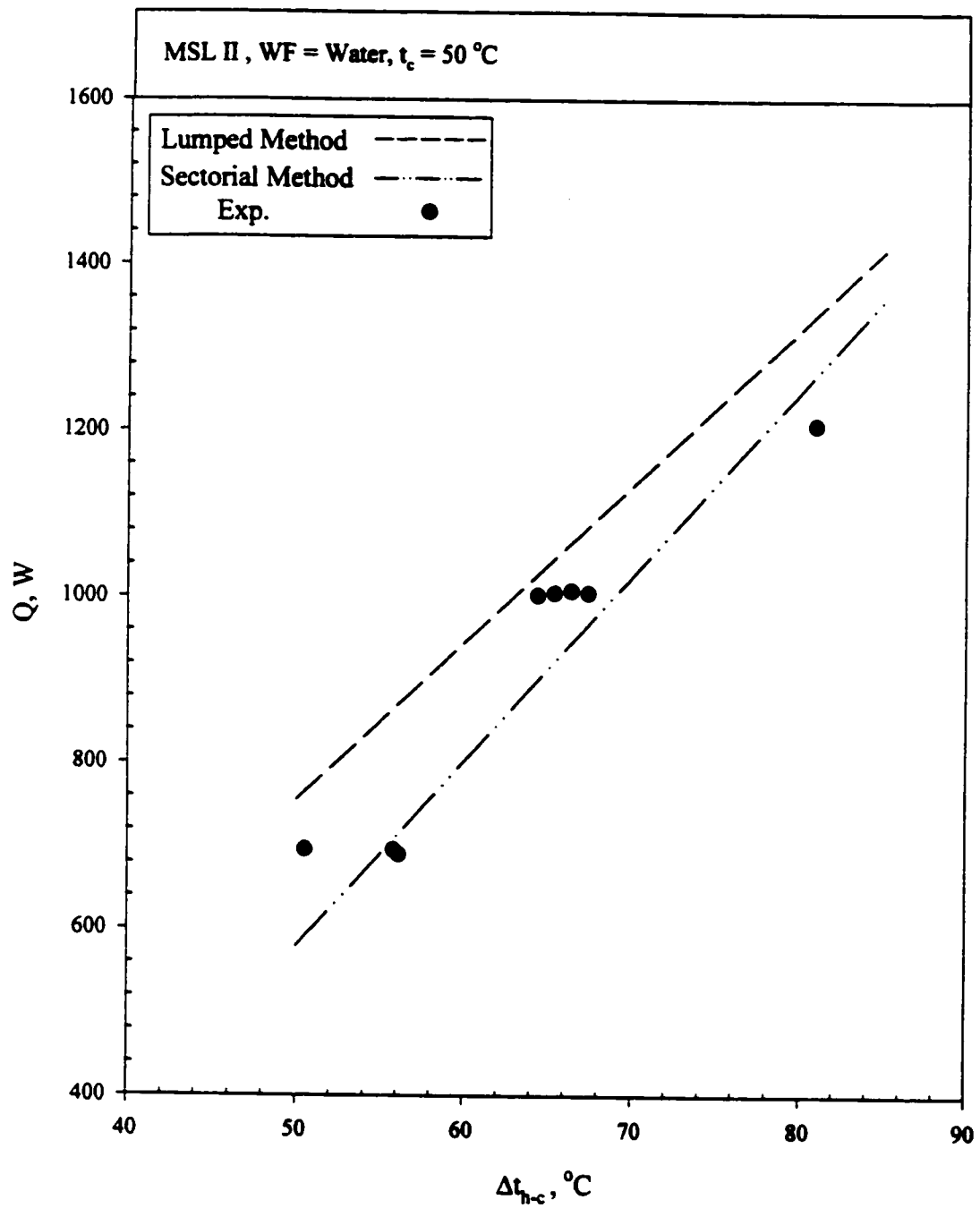


Figure 5.34 (c). Comparison between Experiment and Simulation; MSL II

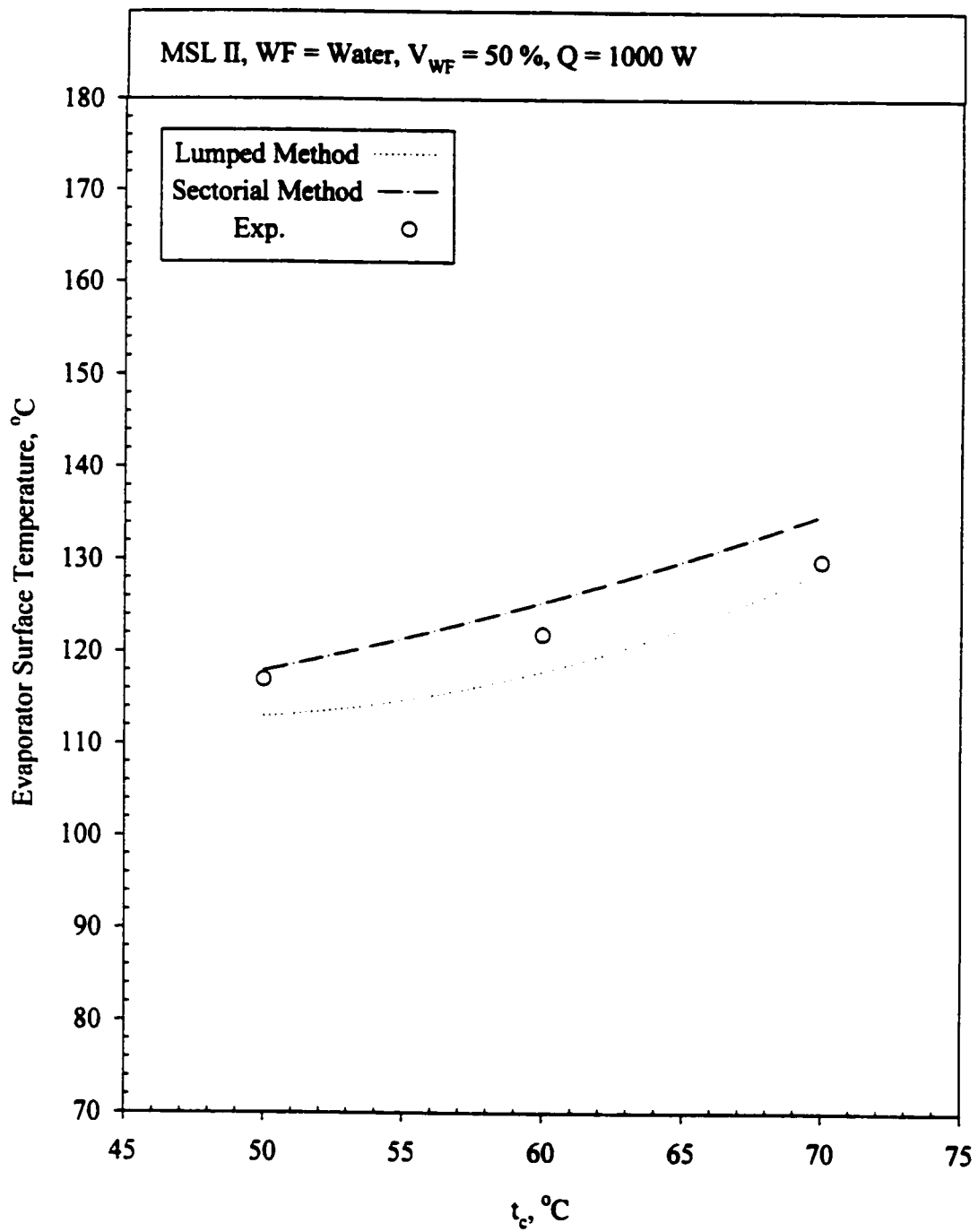


Figure 5.34 (d). Comparison between Experiment and Simulation
Effect of t_c on Evaporator Surface Temperature; MSL II

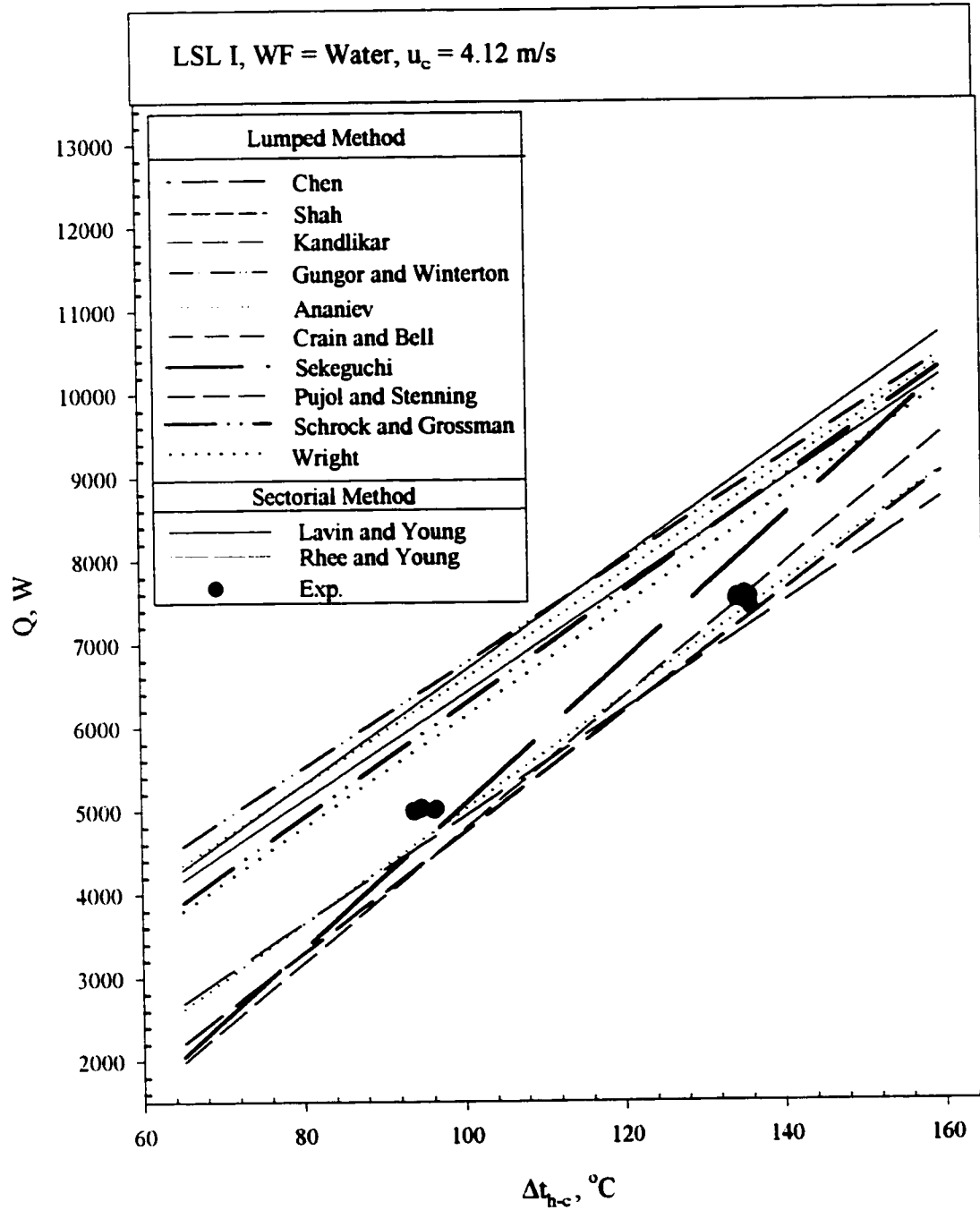


Figure 5.35 (a). Comparison between Experiment and Simulation; LSL I

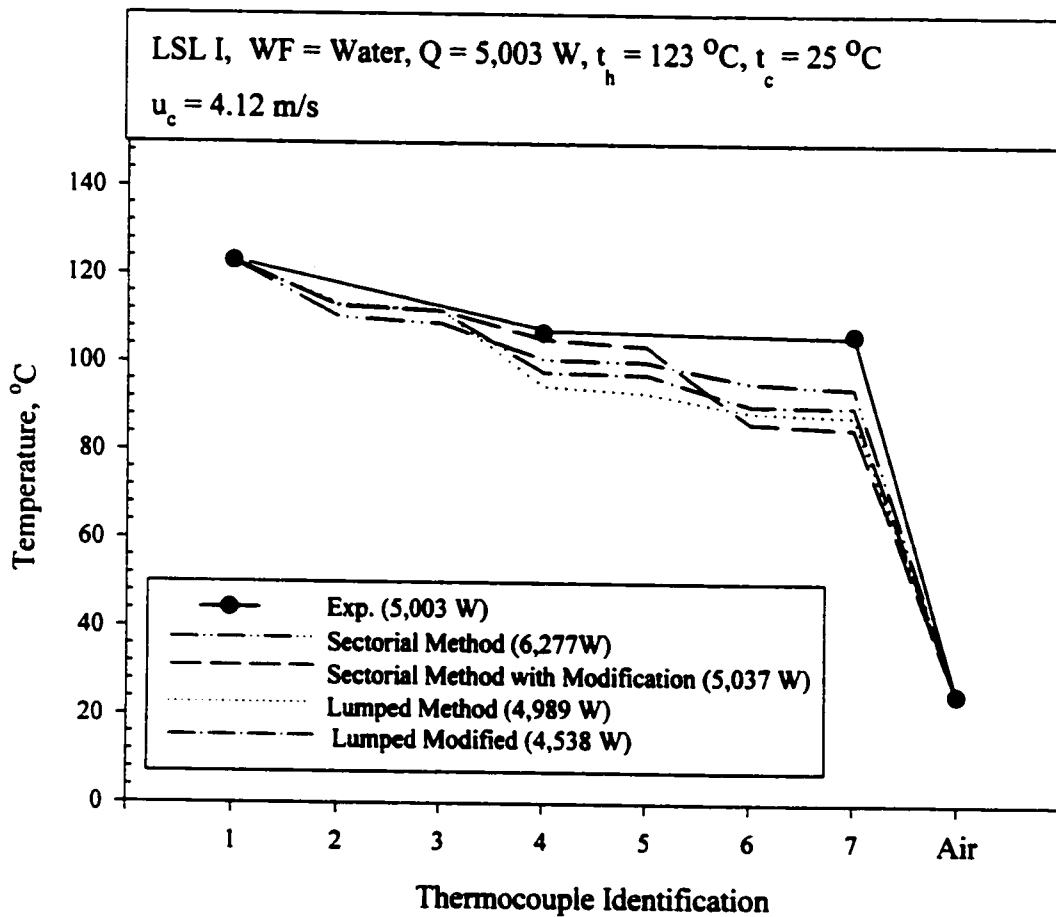
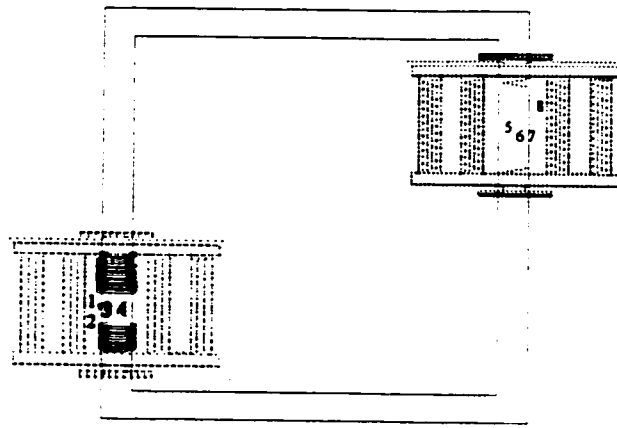


Figure 5.35 (b). Comparison of Temperature Distribution between Experiment and Simulation; LSL I

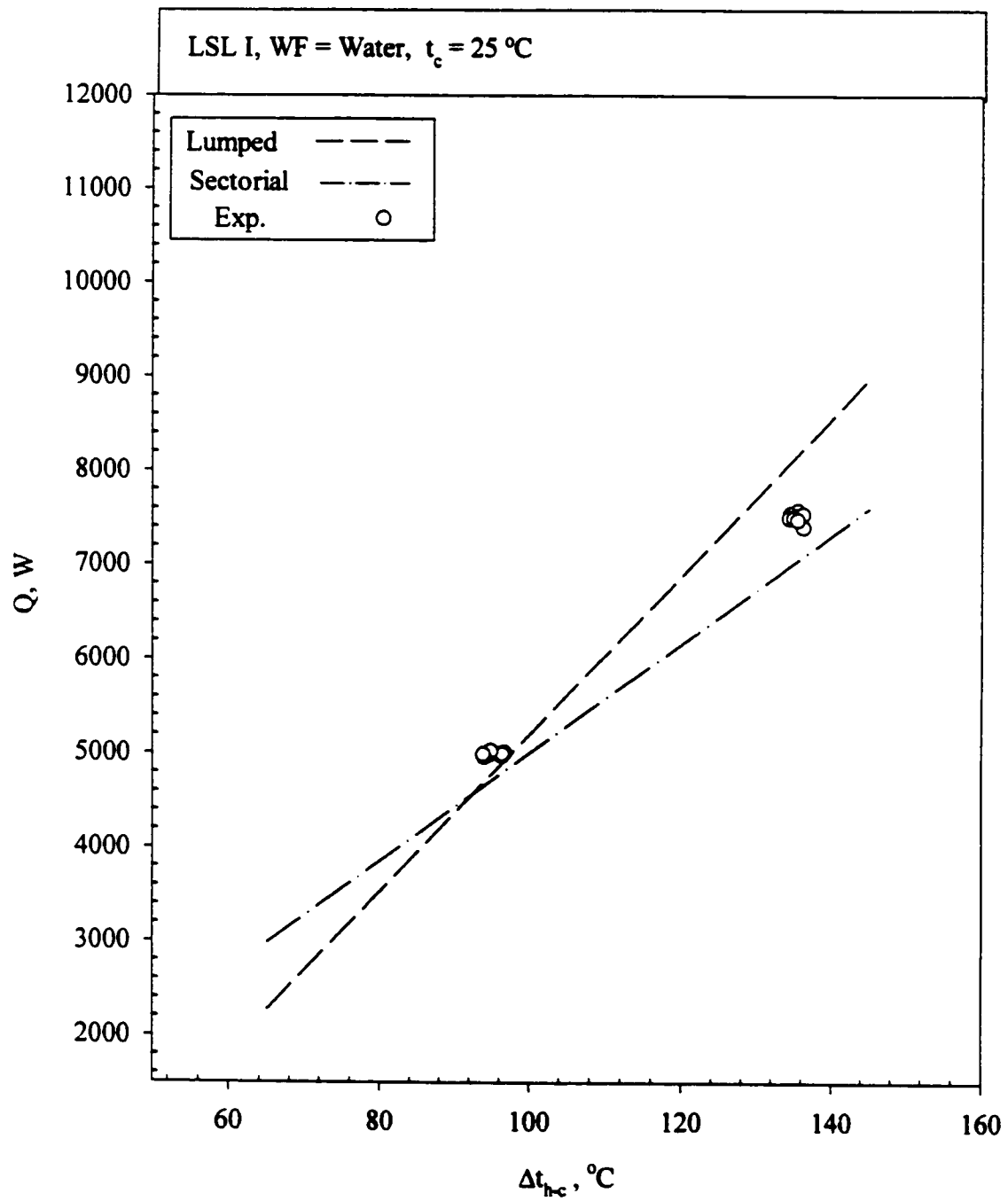


Figure 5.35 (c). Comparison between Experiment and Simulation; LSL I

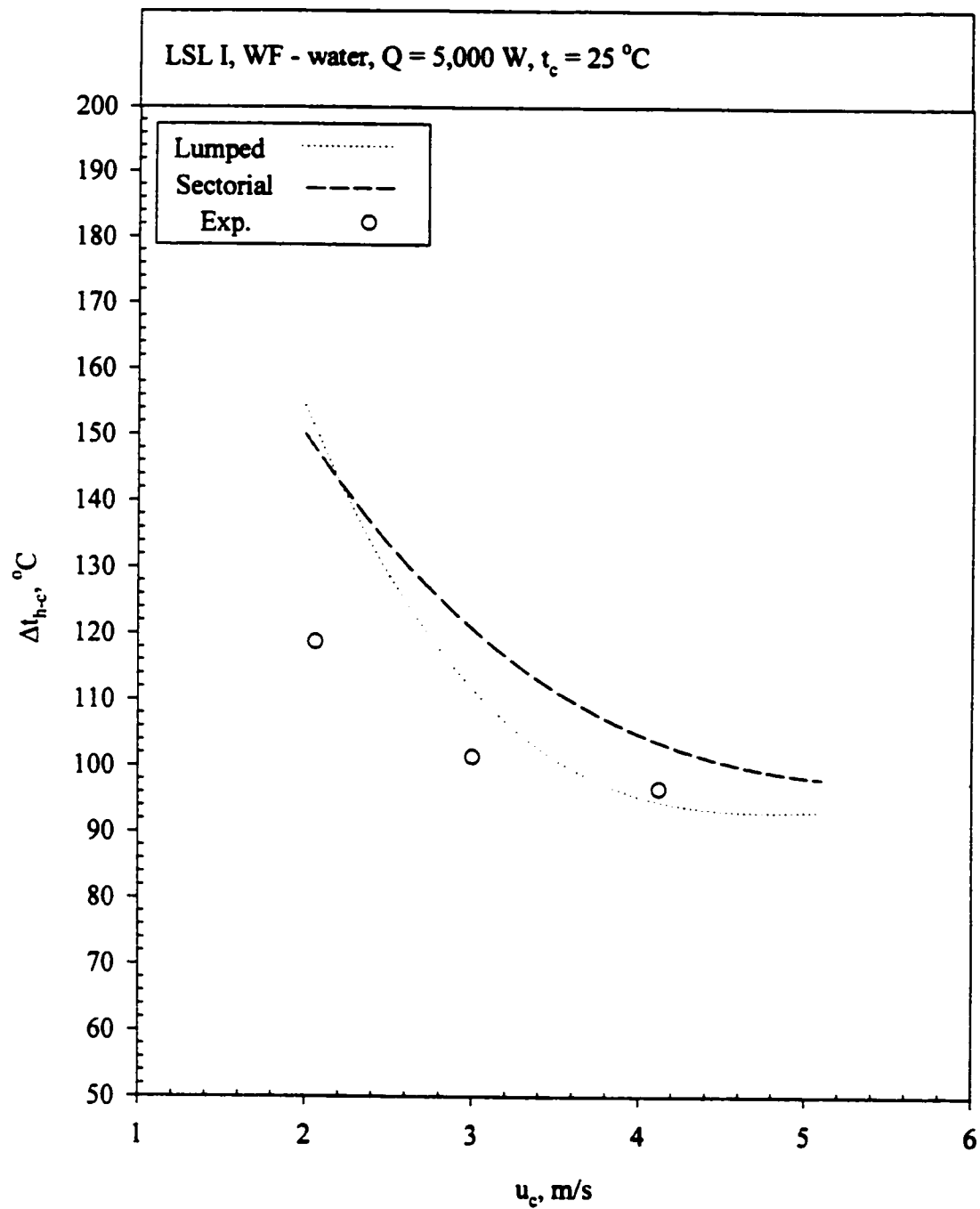


Figure 5.35 (d). Comparison between Experiment and Simulation;
Effect of u_c on Δt_{h-c} , LSL I

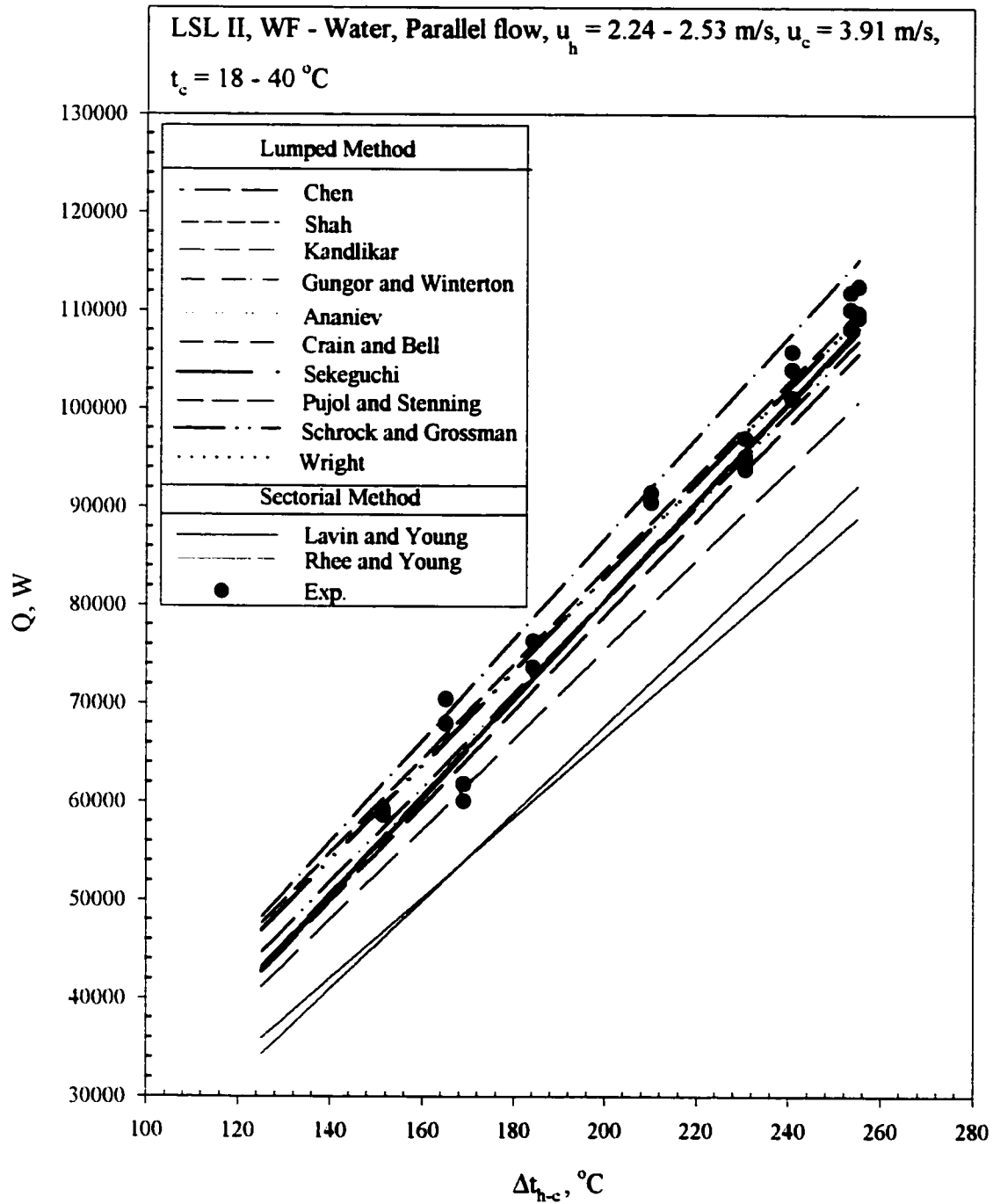


Figure 5.36 (a). Comparison between Experiment and Simulation;
 LSL II with Parallel flow

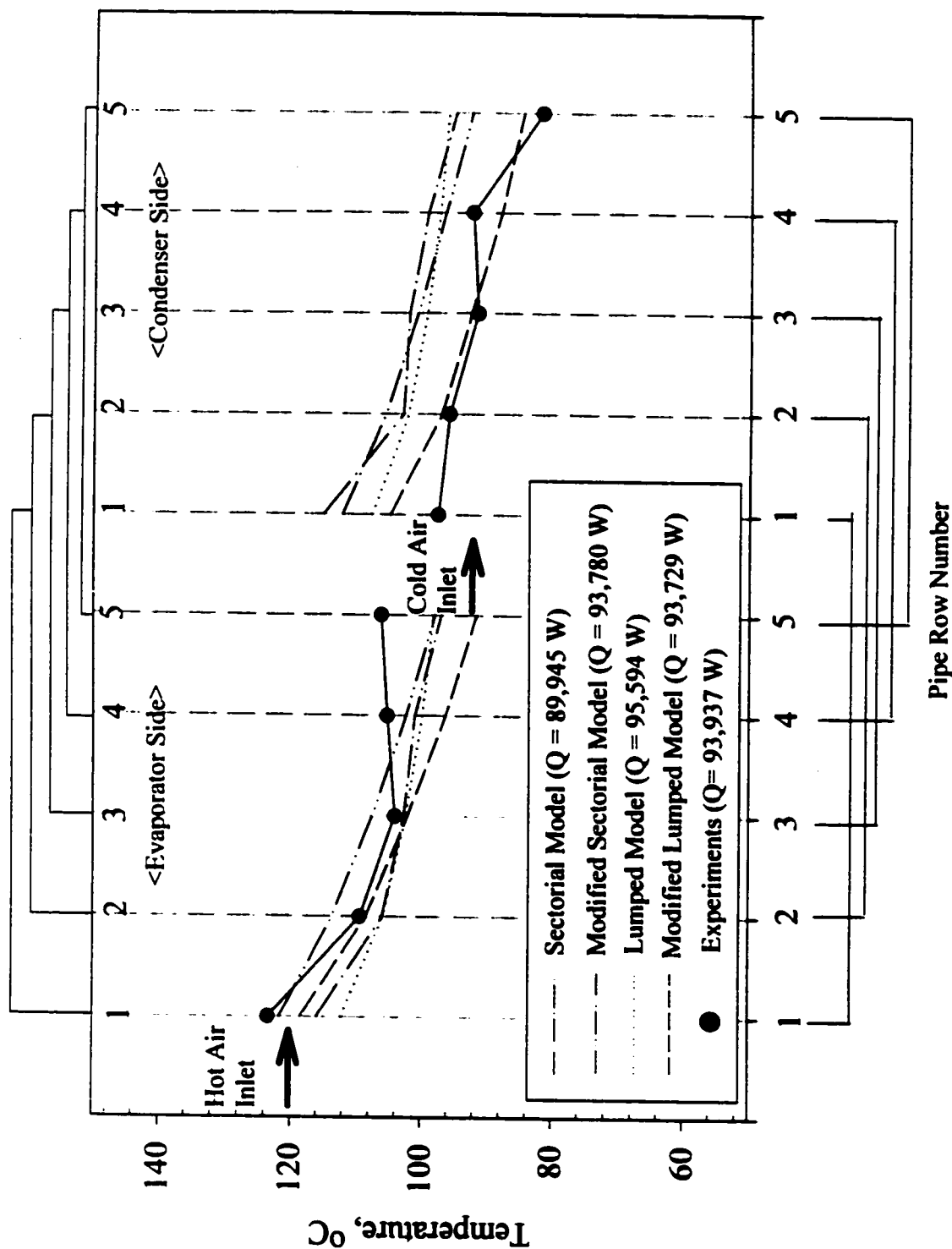


Figure 5.36 (b). Comparison of Temperature Distribution
 Experiment and Simulation (Parallel Flow); WF = Water, LSL II
 ($Q = 93,937 \text{ W}$, $t_h = 250^\circ\text{C}$, $t_c = 17^\circ\text{C}$, $V_{WFR} = 75\%$, $u_h = 2.33 \text{ m/s}$, $u_c = 3.13 \text{ m/s}$)

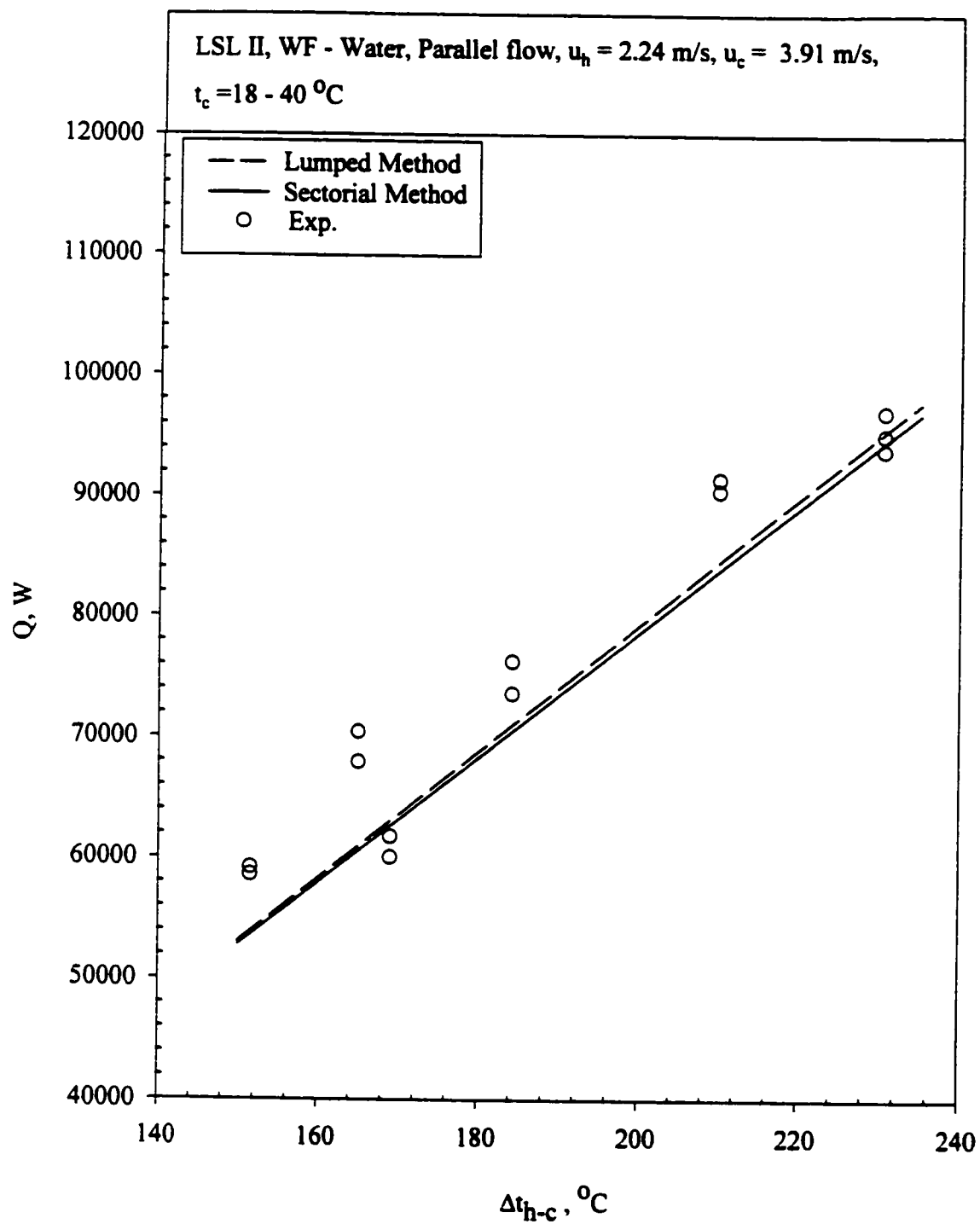


Figure 5.36 (c). Comparison between Experiment and Simulation;
 Parallel Flow, LSL II

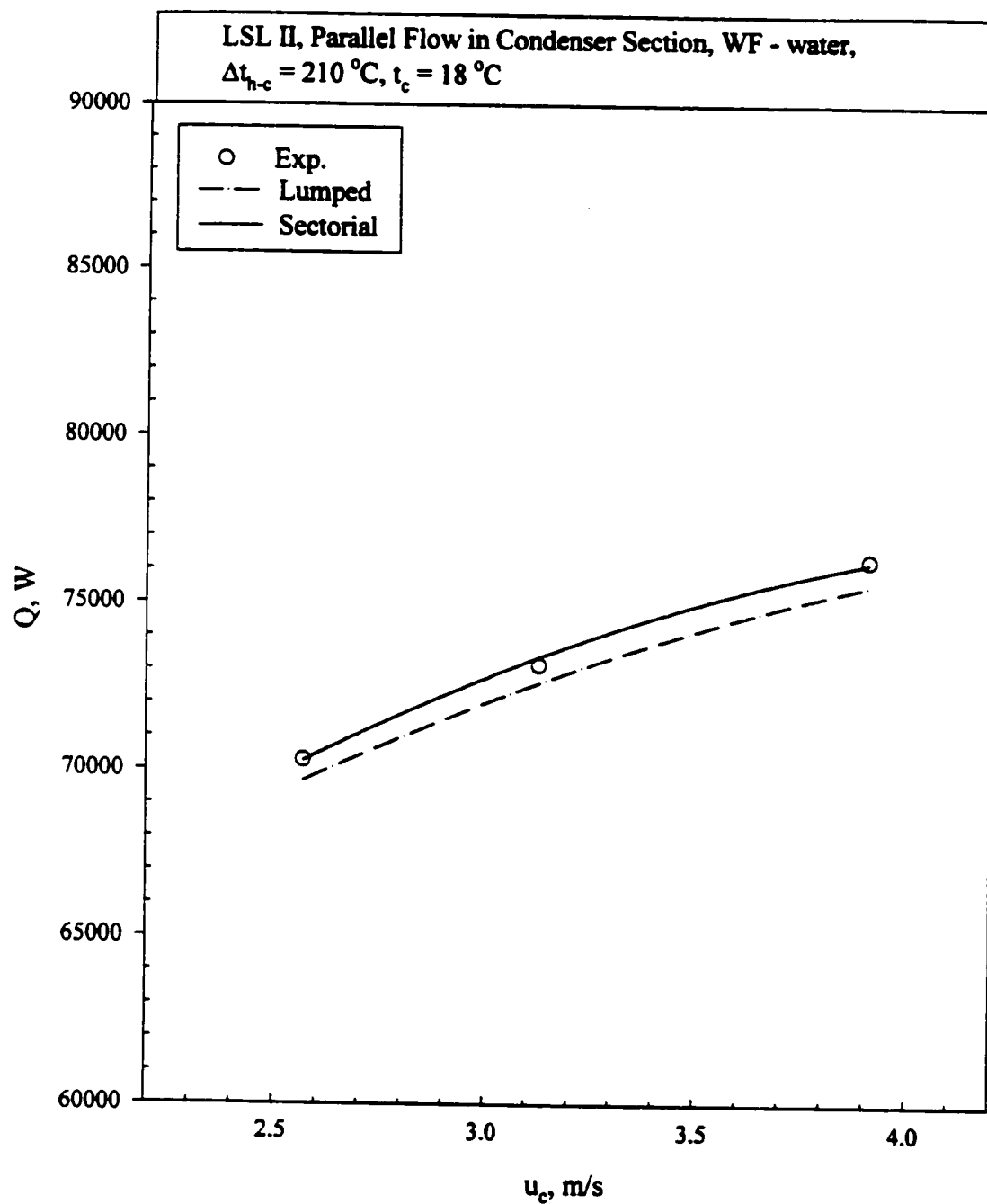


Figure 5.36 (d). Comparison between Experiment and Simulation;
LSL II with Parallel Flow

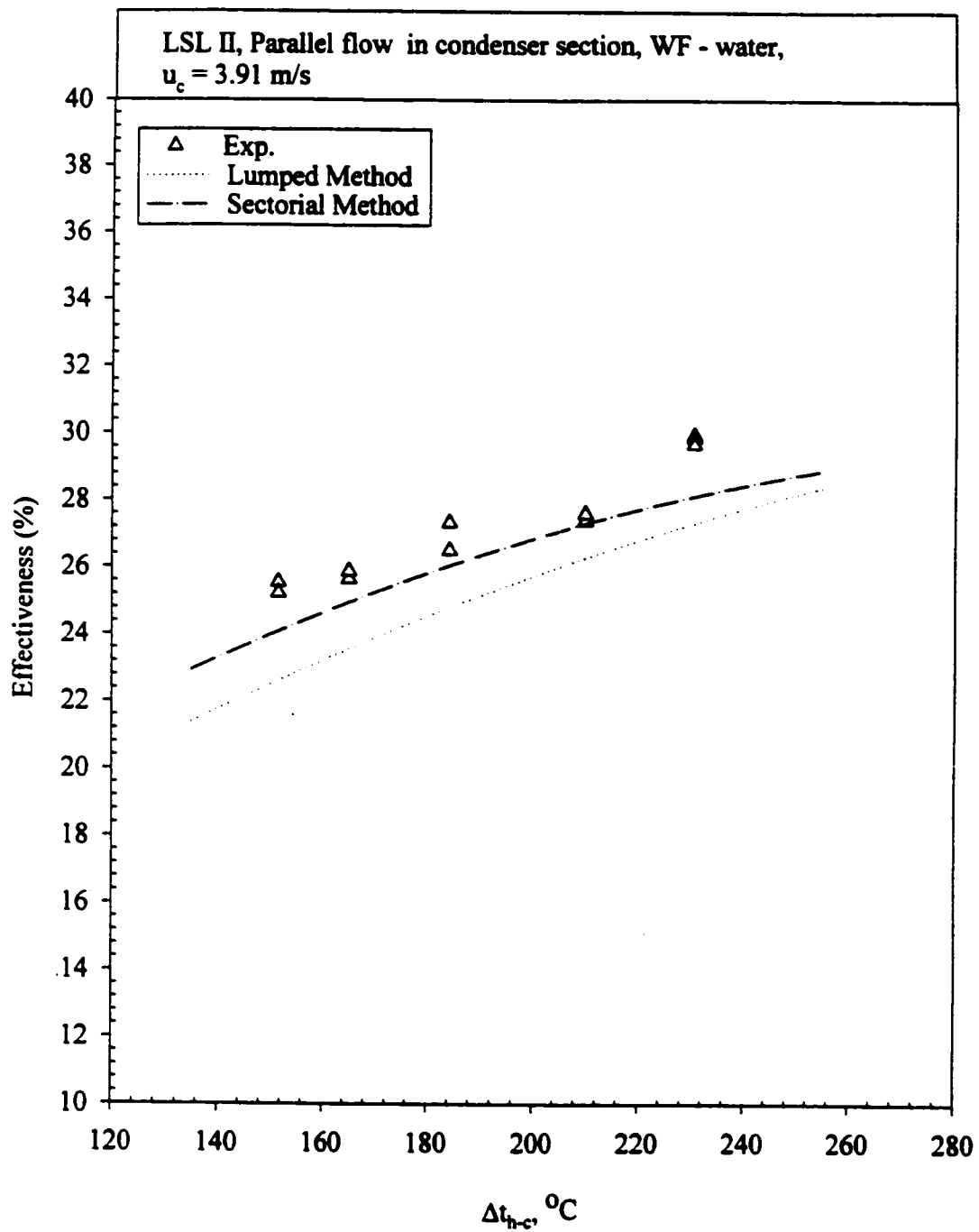


Figure 5.36 (e). Comparison between Experiment and Simulation;
Effect of Δt_{h-c} on Effectiveness (%),
LSL II with Parallel Flow

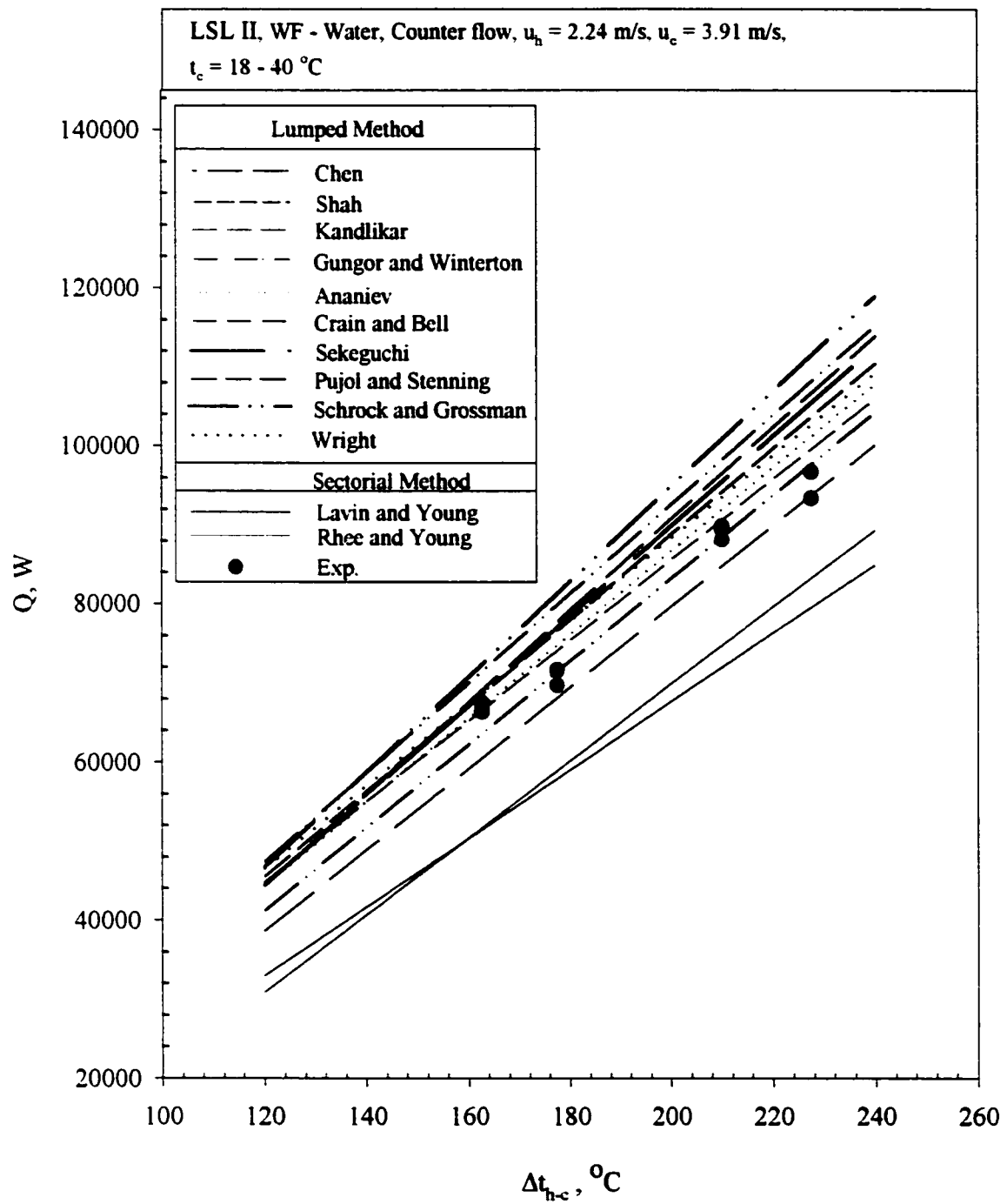


Figure 5.36 (f). Comparison between Experiment and Simulation;
 LSL II with Counter flow

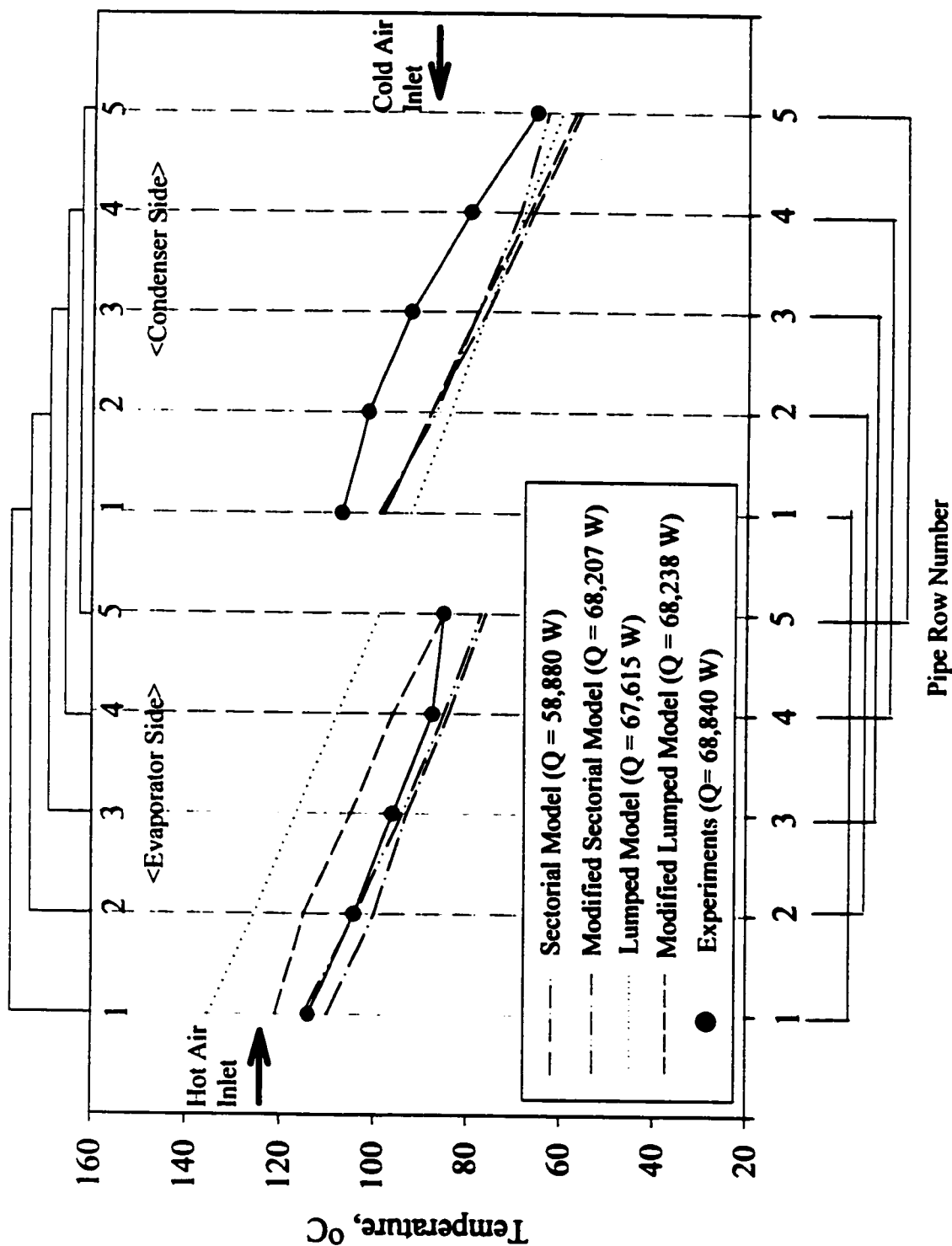


Figure 5.36 (g). Comparison of Temperature Distribution
Experiment and Simulation; WF = Water, LSL II with Counter flow
($Q = 68,840 \text{ W}$, $t_h = 200 \text{ }^\circ\text{C}$, $t_c = 20.3 \text{ }^\circ\text{C}$, $V_{WFR} = 50 \%$, $u_h = 2.24 \text{ m/s}$, $u_c = 2.57 \text{ m/s}$)

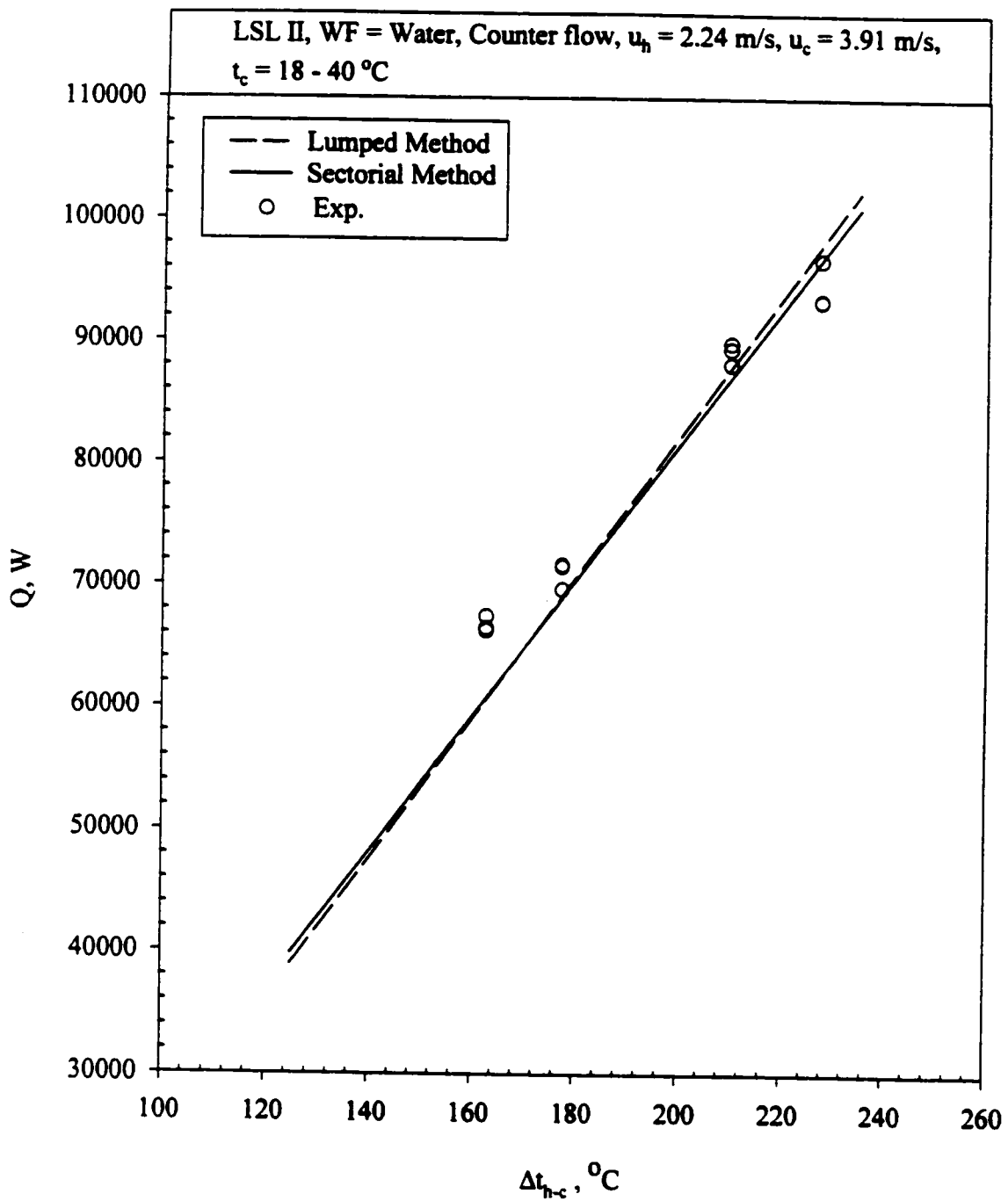


Figure 5.36 (h). Comparison between Experiment and Simulation;
 LSL II with Counter flow

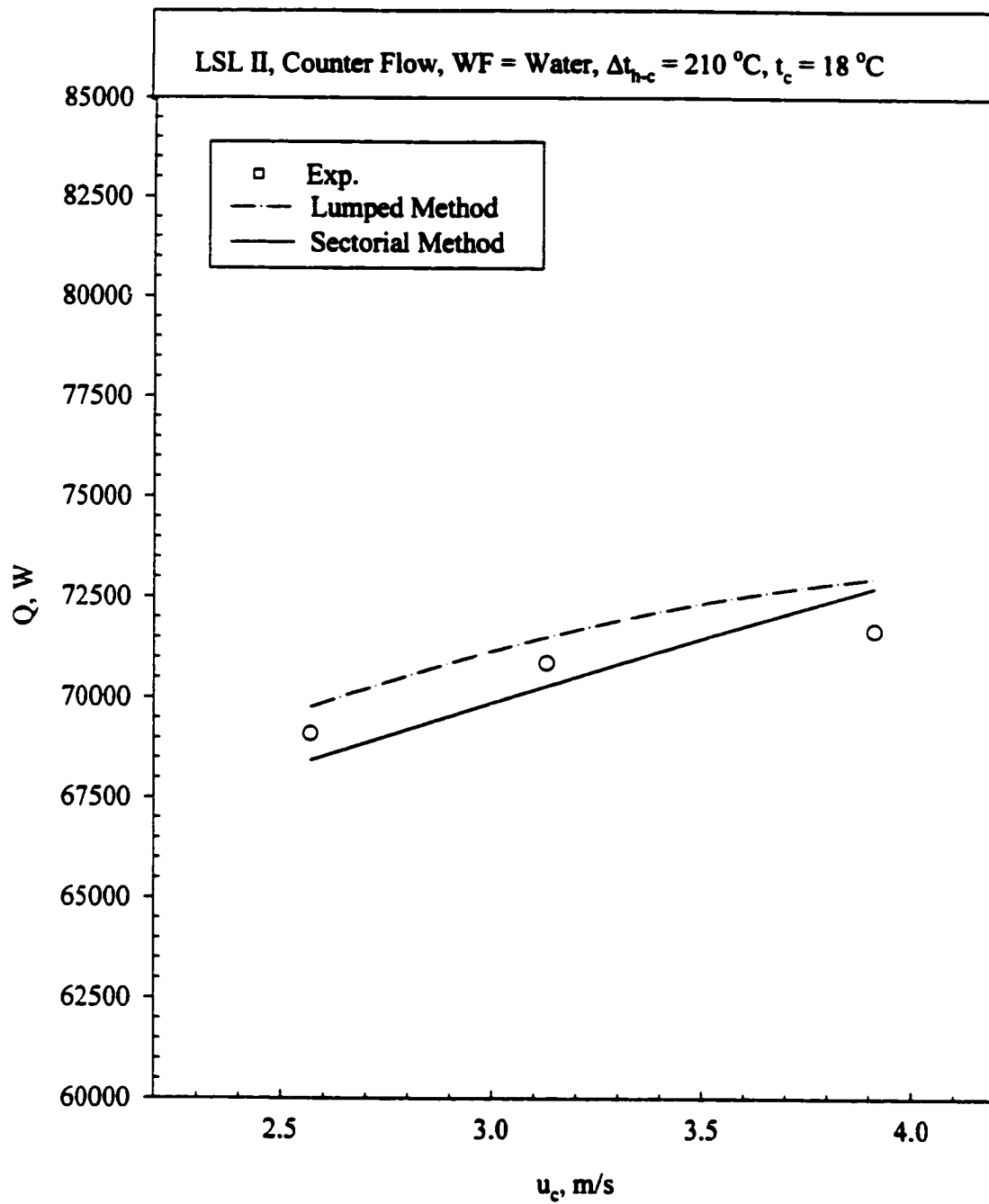


Figure 5.36 (i). Comparison of Experiment and Simulation
Effect of u_c on Q , LSL II with Counter Flow

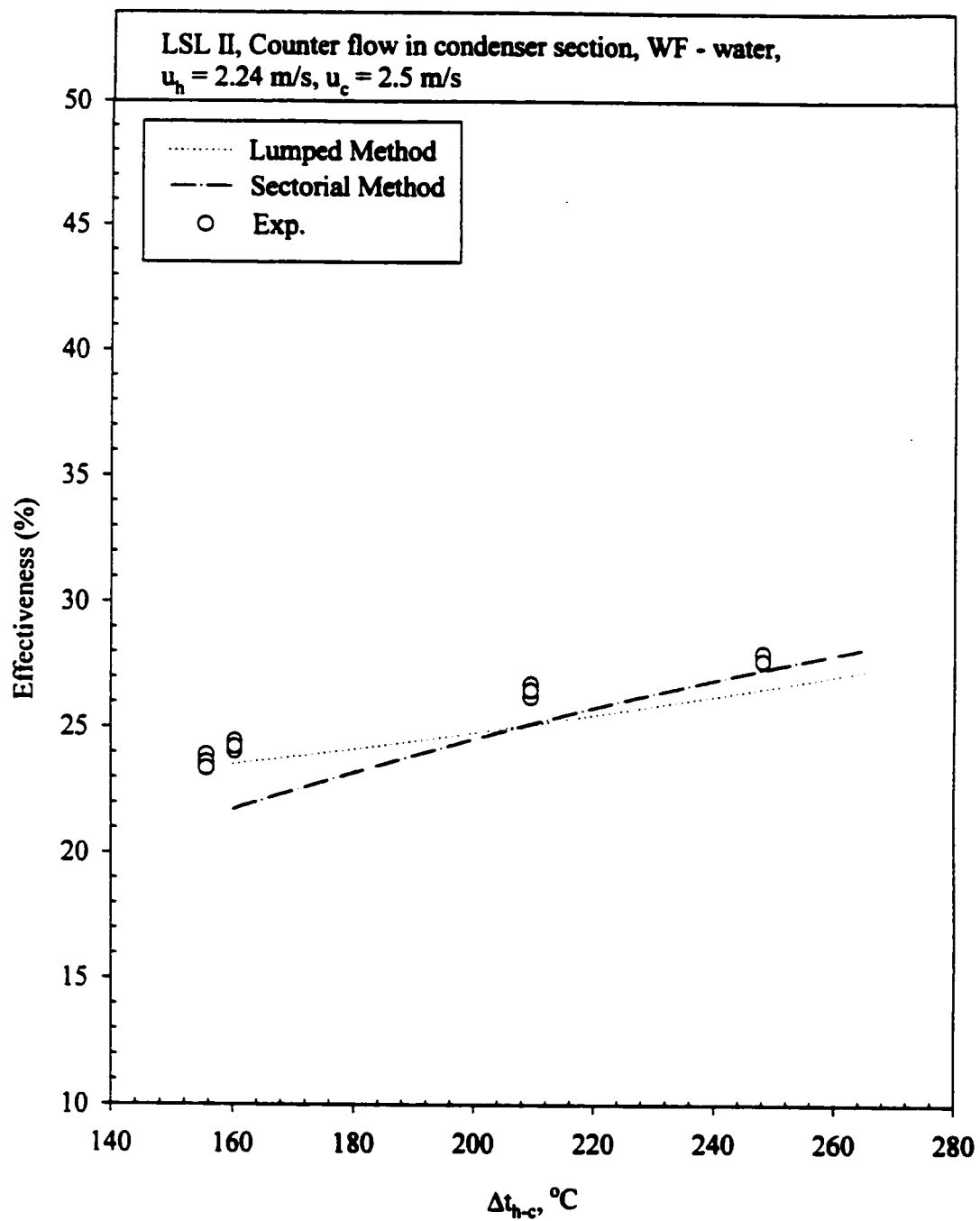


Figure 5.36 (j). Comparison between Experiment and Simulation, Effect of Δt_{h-c} on Effectiveness (%) ; LSL II with Counter Flow