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HELIUM HEAT TRANSFER AND REFRIGERATION IN SUPPORT OF MAGNETIC FUSION ENERGY SYSTEMS

Cryogenics Division
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Boulder, Colorado 80303

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NATIONAL BUREAU OF STANDARDS, Ernest Ambler, Director

ABSTRACT

This is the first year-end report on a program of studies on cryogenic engineering data in support of magnetic fusion energy projects. The report is divided into four parts: (1) an assessment of the cryogenic engineering data base used in the MFE community, and recommendations for needed work for that base, (2) experimental progress on measurement of transient helium heat transfer; the data are of importance for magnet stability analysis, (3) presentation of a newly developed general technique for analyzing the efficiency of helium refrigerators of any configuration and thereby identifying sources of inefficiency, and (4) progress towards setting up a data bank on refrigeration system reliability The technology assessment, item (1), is a revision of a preliminary version dated May 13, 1977, with inclusion of feedback received from both within and outside of the MFE community.

Both the technology assessment, Part One, and the efficiency study, Part Three have been formatted for individual publication subsequent to this report. The technology assessment will be distributed to those who received copies of the preliminary edition. The efficiency study will be submitted for publication in Cryogenics. Because of our fairly recent publication on transient heat transfer (Part Two, appendix A) , it is appropriate to do further research in this field before writing it up for a follow-up publication in an outside journal.

Key words: Efficiency; heat transfer; helium; pressure measurement; refrigeration; stability; superconducting magnets; transient heat transfer.

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NBSIR 78-877, Part One

MAGNETIC FUSION ENERGY
CRYOGENIC ENGINEERING PROGRAM
A SURVEY

October 1, 1977

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I. INTRODUCTION AND EXECUTIVE SUMMARY

Cryogenic Engineering plays an important role in Magnetic Fusion Energy because the program requires superconducting magnets and large refrigeration systems. Further, the role of cryogenic engineering extends from conception through execution, approximately as follows:

basic design	requires	materials and systems data
operational reliability	requires	proven components and reliable controls
performance evaluation	requires	accurate instrumentation
industrial safety	requires	documentation of the above for standard practices.

In this report we consider the cryogenic engineering field which includes helium heat transfer, flow, refrigeration, and instrumentation, as it relates to the above requirements. It is important to note that this report complements the recent report of Reed, Fickett, Kasen, and McHenry (March 1, 1977) of our laboratory outlining research on cryogenic materials. Both reports share in a common data base on the MFE program which has been collected within our laboratory.

The report identifies areas where the above cryogenic data or technology will be inadequate to meet the goals of the MFE program on a timely basis and recommends specific research programs to fill that need. Emphasis is given to the TNS and/or EPR programs targeted for the mid-1980's, with some consideration to the DEMO stage programs in, nominally, the 1990's. It is important to recognize that a significant portion of our input on this technological summary comes from other advanced technological programs as well as the MFE program, and our specific recommendations are moderated by on-going work supported by these other programs.

Section II discusses the MFE program components whose design, fabrication, and/or purchase heavily involves cryogenic engineering. Section III is an assessment of current cryogenic technology supporting these components. Section IV contains our recommendations for cryogenic research in support of these components. Section V lists references and acronyms.

II. CRYOGENIC ENGINEERING IN MFE SYSTEMS

The three major MFE components based in cryogenic technology are (A) superconducting magnets, (B) cryopumps, and (C) refrigeration systems. In this section we discuss these components and the input which has been received concerning them from the MFE community and related programs.

Table II-1 gives an outline of the schedule proposed by ERDA for the development of a number of the fusion devices considered here.

Table II-1

ERDA PROPOSED MAGNETIC FUSION SCHEDULE (1976)*

Project	Start Conceptual Design	Begin Title 1 Funding	Begin Operation	Number Planned	Total 21 yr Cost 76-98** ($M)
Doublet III	Done	Done	1978	1	100
TFTR	Done	1976	1981	1	230
MX MFTF	1976	1979	1982	1	100
TNS/ITR+	1977	1980	1986	3	400
EPR	1983	1986	1992	2	800
Demo	1983	1986	1999	1	1200

*The schedule actually has many branch points depending on results of experiments completed.

+This is the tokamak version; the mirror version is on the same schedule offset 4 years into the future.

Other Facilities:					
RTNS	Done	Done	1978	1	5
LCP	Done	Done	1980	3-6 coils	37
INS	Done	Done	1981	1	25
Superconducting Magnet Test Facility	1980	1982	1985	1	50
FERF/ETR	1980	1983	1989	1	500

Source: ERDA-76/110/1. **These data are difficult to extract from the available literature.

II.A. Superconducting Magnets

Consideration of present designs for toroidal confinement fields for the experimental power reactor (EPR) tokamak, currently thought the most promising confinement system, suggests that approximately 20 magnets, each of about 10 meter radius and weighing about 1,000 tons, must be constructed. Superconducting tokomak toroidal field coils are being planned for peak fields in the 7 to 15 Tesla range. Mirror magnet coils include about the same range, though perhaps extended to 20 Tesla. Ohmic heating coils will have a peak of 4 to 10 Tesla, with rise times of 1 to 20 seconds. For comparison, the largest present-day magnet, at Argonne National Laboratory, weighs an order of magnitude less, 100 tons, and reaches a peak field of less than 2 Tesla. Table II-2 gives some general parameters of the toroidal system designs. Both design and operating factors contribute to the complexity of furnishing experimental fusion reactors with superconducting magnets. [Henning, 1976]

The problems of scaling up the present technology by this order of magnitude are formidable, and it is not surprising to find that a great majority of the research effort within the scope of this study is associated with superconducting magnets.

A wide variety of design concepts have been proposed for superconducting MFE coils, but for our purposes we divide them into two classes. The first consists of the bath cooling designs and the second consists of the force-cooled designs. The difference between these two types of designs can be traced, at least in part, to the assumed size and duration of the perturbing heat inputs which the cooling system must absorb to achieve magnet stability. A massive perturbation requires a fully cryostable design, where the magnet can recover without interruption of current even if the initial perturbation causes all the current to flow in the copper over a significant portion of the winding. Pool boiling systems can be designed to recover from substantial perturbations of this type, the limit being set by vapor-blanketing of the conductor. Forced convection systems, on the other hand, would react to a substantial perturbation with a considerable rise in local coolant pressure and temperature accompanied by strong flow perturbations, and full recovery may not be attainable for a time at least as long as the transit time of the helium through the cooling channel; however, if the initial perturbation is small in total energy content, force-cooled systems can be highly stable. Inherent in the force-cooled approach is the assumption that research and analysis will reach the point that massive perturbations can be prevented by good design.

Table II-

TOROIDAL SYSTEMS SUMMARY

Project	Power output (MWe)	Torus diameter (m)	Toroidal Field Coils: Size (m)	#	Normal(n) superconductor(s)	Field on axis/max. (T)	Some poloidal field coils inside TR? n or s?	Pulse Times: Initiation (s)	Burn (s)	Toroidal β %	First Wall Loading (MW/m^2)
ANL EPRI		12	9x14 D	16	s	3.34/7.5	no, s (n)	1	37		
ANL EPRII	30	12	9.5x14 D	16	s	4.47/10(3 K)	no, s	2	55	4.8	0.86
GA D III	-										
GA EPRI	180	16	∿7x12 D	24	s	4/8	yes, n	∿7.5	60		1.03
GA EPRII	124	24	8.1x12.4 D	16	s	3.9/7.9	yes, n	2	105	10	0.85
GA/ANL TNS											
GA DEMO	600	30	11x17 D	16	s	3.9/7.5	yes, n		800		0.97
MIT/HDTR		11.2	7.5x12.5 D	24	s	8/14.7	yes, n	∿2	400		
ORNL EPRI	300	13.5	8.6x11.4 oval	20	s	4.8/11	yes, n	2	100		0.168
ORNL EPRII		13.5	8.8x11 oval	20	s	5/11	yes, n		100	3	1
ORNL/W TNS											
ORNL EBTR48	∿1600	120	7.2 circ.	48	s	2.5-4.5/7.3	--	--	--	25	1.13
ORNL DEMO											
ORNL LCP	--	4.6	2.5x3.5 noncir.	6	s	-/8.0	--	1	--	--	--
PPPL TFTR	--	5.4	3.8 circ.	20	n	5.2/9.5	no, n	4	--		0.15
PPPL Plant	2030	21	14.5x21 D	48	s	6/16	yes, s	∿13	6000	4	2
UWMAK I	1500	26	17x24 D	12	s	3.83/8.66	no, s	100	5400	5.2	1.25
UWMAK II	1700	26	21x30 D	24	s	3.67/8.30	yes, s	10	5400	6.5	1.16
UWMAK III	2000	16	15x26 D	18	s	4.05/8.75	yes, s	15	1800	6-8	1.91
TETR	--	6.5	9.5x10.5 D	16	s	4.19/8.5	yes, n	1	60	3.8	1.0

It is important to recognize, however, that the distinction between force-cooled and bath-cooled magnets is not always clear. For example, in some designs the background heat load on a bath cooled magnet can generate substantial convection flow which may be analyzed, from a heat-transfer viewpoint, as forced flow.

The conservative approach to magnet design, which has served well in the past, has been expressed by Hilal and Boom [1976]:

"Large fusion reactor magnets such as the D magnet TF (toroidal field) coils seem to require the reliability inherently obtained with cryogenically stable conductors. There is little need to employ high current density cryogenically unstable conductors since the major space requirement is for massive structure. In addition, the amount of copper used is not a major cost item; eliminating copper to obtain a smaller unstable conductor would be a minor achievement. Unexpected super-to-normal transitions typical of cryogenically unstable coils are unacceptable in terms of potential magnet damage and lost time for a utility company."

However, provided that the problems of reliable design and operation of higher current density coils can be understood and controlled, there are many potential advantages which deserve consideration. The force-cooled magnets, e.g., under study at MIT and supported by ORNL, lie in this category. This approach is not unique to the ORNL group, but can be found, for example in non-MFE programs as well: a forced circulation cooling system (FCCS) has also been selected by the ISABELLE project at BNL.

In this document we must give due consideration to both bath-cooled and forced-convection cooled systems. Both are advocated by responsible groups within the MFE community, as well as the broader technical community, and both are reflected in our technical assessment (section III) and recommendations (section IV).

Cryostatic stability analysis as used by Cornish, Purcell, and Boom is based essentially upon the early work of Stekly [1965] and the modification for end-cooling effects suggested by Maddock, James and Norris [1969]. Recently Wipf and Sikora at LASL have been improving and generalizing this analytical approach, but a publication is not yet available.

Stability considerations for smaller perturbations, such as flux jumps or small frictional conductor movements, have been summarized by Wilson [1976]. In particular, he develops the concept of transient stability, or recovery on the millisecond time scale.

An important trend which was mentioned both by Purcell (GA) and Boom (U. Wisc.) is towards operation at less than 4 Kelvin. Purcell believes that required fields in the 10 Tesla range can be obtained more assuredly by NbTi coils at (nominally) 3 Kelvin than by the more difficult engineering of Nb_3Sn coils at 4 K. Boom feels that as magnets become very tall, pool boiling may have to be abandoned because of excessive vapor collection at the top of the dewar during periods of high heat loads. In this case, superfluid cooling may look attractive. His comments apply more to huge Energy Storage coils than to MFE systems of the 1980's, but they may be applicable to DEMO's in the 1990's.

The problems, then, which the magnet designers face may be listed as follows:

1. What is the maximum magnitude and duration of the initial perturbation which the helium cooling system must be designed to handle?

2. What are the correct numerical values of the heat transfer parameters which appear in the Stekly, Maddock-James-Norris, and Wilson types of stability theories? When is it necessary to use transient rather than steady state heat transfer correlations in these theories? How much difference is involved?

3. For pool-boiling systems, will natural convection provide enough fluid motion to avoid vapor-blanketing during the course of an instability? Can the geometry and orientation effects be predicted? If operation at less than 4 K is desired, how does the attainable heat transfer depend upon whether pool boiling at reduced pressure, or pressurized subcooled liquid is used?

4. For forced-convection cooling systems, how large are the pressure, temperature, and flow perturbations following an initial instability of a given size? Will these require that the coil be de-energized in order to avoid burnout?

The first question (above) requires performance analysis on real magnet systems, which is beyond the scope of this report, though it does influence the answer to the succeeding questions. In section III we assess the current technology related to these questions, leading to specific recommendations in section IV.

II.B. Cryopumps

Multiampere neutral beam injectors will require vacuum systems capable of pumping large quantities of hydrogen and deuterium (greater than 10 Torr-liters per second) while maintaining a vacuum in the 10^{-4} Torr range during neutral beam pulses. In addition, large quantities of these gases have to be pumped out without interrupting the operation of the system. Cryopumps offer some outstanding advantages for this application [Halama and Bamberger, 1975].

This subject was discussed by telephone both with H. Halama at BNL and T. Batzer at LLL. The conclusion is that design data for such pumps is largely available, the exceptions being:

1. If one chooses to use a gas which melts at a relatively high temperature (e.g., CO_2) to smother and trap a low temperature gas at higher-than-usual cryopump temperature, performance data is lacking.

2. If the trapped gases contain tritium, its radioactive decay will release heat, which may degrade the trap performance.

Though these were viewed as relevant and timely problems, they were under study, and it was felt that additional work under this program was not necessary for the MFE program. Accordingly, we have neglected further consideration of this topic, and will not discuss it further in this report. However, we will be glad to receive further input from interested reviewers, especially if other actions are recommended.

II.C. Refrigerator Systems

Essentially all of the existing work on refrigeration for MFE is connected with the Large Coil Project Test Facility (LCPTF) at ORNL. Basic to this facility will be a large refrigeration system with a complex of outputs to up to six different test coils which can be mixed pool-boiling and/or force cooled. Considerable planning is going into the installation, and it appears that the system will utilize the best of present cryogenic technology in an effective way. However, it is important to realize that, at least at this time, it is not being designed with the redundancy of critical components, advanced computer control, etc., which will be demanded of an EPR for which unscheduled refrigeration down-time may be disastrous.

The refrigeration problems listed by the LCPTF project review (April 14-15, 1977) include the following:

1) Impurities from the magnets and support structure are a problem. Adequate impurity analysis and purification procedures need to be found.

2) Liquid helium pumps may be needed for the forced circulation systems. However, long and expensive development programs may be required to obtain pumps of documented reliability.

3) All magnets and leads will require accurate measurement of helium coolant flow. Flow measurement systems must be calibrated since there is no experimental data which will allow accurate prediction of flowmeter performance at the anticipated helium states and flow rates.

4) Thermal oscillations can cause excessive heat leaks into the cryogenic system. This phenomenon is not well understood, requiring careful design and construction of liquid helium transfer systems to try to avoid this type of difficulty during operation.

Important input has been also obtained from the well-researched reports of Kadi and Longsworth, and Manatt, et al., prepared for ERDA-Power Transmission. Much of the information they contain is applicable as well to MFE refrigerator plans. The interested reader is encouraged to review these two reports very carefully, and we shall not attempt to discuss them here. Suffice it just to list the conclusions and recommendations which are applicable to MFE as well as SPTL systems.

1. Both reports recognize the need for on-line instrumentation to monitor impurities in the gas stream, and recommend that research be done in this field. See section III.C for further discussion of the topic, and IV.A. for our recommended work.

2. Both reports recommend that a Refrigerator Reliability data bank be established and operated. This is further discussed later in this section, and recommended work is outlined in IV.C.

3. Both reports give considerable discussion to the room temperature compressors which are used for helium refrigerators. The screw compressor is the first choice at this time because it is readily available and probably very reliable.

However, its efficiency is low (38 to 42%) and it presents severe oil removal problems for which long time answers are unknown. Manatt, et al. go on to present a strong case for the development of centrifugal compressors. This is discussed in more detail in III.C.

It is important to note that the helium refrigerator capacities required for the EPR installations are estimated to be 10^2 to 10^3 larger than the largest existing helium refrigerator (about 1500 watts at 4.2 K), or roughly the size of large industrial air separation plants.

III. ASSESSMENT OF THE CRYOGENIC ENGINEERING DATA BASE

In this section we assess various cryogenic engineering technologies which are available to the MFE community. The material has been collected from a number of sources and personal contacts. The assessment leads to the identification of several areas where we recommend that DMFE support research towards a clearly identified goal of the MFE program. We encourage direct discussion with any readers who wish to raise questions concerning the completeness or accuracy of the material presented.

III.A. Helium Heat Transfer and Flow in Superconducting Magnets

During the week of July 25-29, 1977 a most productive and useful workshop on stability in superconducting magnets was held at Los Alamos (not funded by this project). The agenda for the technical program is listed in Appendix A, and the attendees are listed in Appendix B. During this workshop the problems of heat transfer and flow were thoroughly discussed, both in a separate subgroup, and then before the whole workshop. Typed notes from this workshop are expected in the near future, but only handwritten notes of the participants are available now. This section of our technology assessment strongly reflects the conclusions of that workshop.

Heat Transfer

Accurate heat transfer data are necessary in order to analyze, understand, and predict the stability of superconducting coils. For conservatively designed, low current density coils, e.g., the ANL bubble chamber magnet and the Stanford LASS magnet, existing steady state heat transfer data suffice, though tests on representative sections may always be necessary when special geometries are used. But as current densities, operating fields, stresses, etc., are pushed upwards, evidence from the work of Dressmer [1976] and Wilson [1976, and recent unpublished work] indicates that one must invoke some estimate of time-dependent heat transfer in order to understand the recovery process and the border between stable and unstable magnet performance. Toward this end, the conclusions and recommendations of the workshop may be summarized as follows:

(1) There is experimental evidence (Dressner, 1977] that steady state heat transfer data are inapplicable during the typical 10^{-1} to 10^{-2} second recovery time of a stable, or quasi-stable, magnet. To understand the recovery process, transient heat transfer data, such as being developed by Dr. Steward at NBS and Dr. Iwasa at MIT, are urgently needed. The experimental measurements should provide accurate heat transfer data on a millisecond time scale; some special applications will require 1 to 10 microsecond response. The NBS measurements are the most comprehensive, with response time of 20 microseconds, but the measurements to date have utilized a carbon film surface which was found to inadequately represent the heat transfer of practical superconductors at short times (a Kapitza conductance effect). The MIT measurements complement the NBS work by utilizating practical superconductors, but with some slower response times (1 millisecond), and more limited fluid conditions. Proposed work for continuing the NBS studies with representative surfaces is outlined in IV.A.

(2) The "cold end recovery" process, first discussed by Maddock, James and Norris [1969], requires knowledge of the complete q (heat flux) vs ΔT curve, including any regions where $\partial q/\partial(\Delta T)$ is negative. This "negative resistance" region is often undocumented -- i.e., it has never been documented for pool-boiling helium -- because q rather than ΔT is usually the controlled parameter in heat transfer experiments. In a q-controlled experiment, this negative resistance region is masked by hysteretic effects. This documentation is needed, both for steady state and, more generally, for controlled values of dT/d(time), both positive and negative. This is a most important area of transient heat transfer research.

(3) The effects of confining geometries in bath cooling are not easily predicted, but some correlations developed for other fluids should be investigated for applicability to helium systems. As part of any large magnet design, it is probable that specific model tests will be required. However, the question of generalized analysis and guidelines was seriously discussed. For encapsulated force-cooled conductors, e.g., those under study by Hoenig and Montgomery at MIT, such analysis rightfully accompanies the experimental work and will be a valuable guide to future design and development. For short-channel flow perpendicular to a conductor, e.g., parallel to the plane of a pancake winding, less systematic work is in progress. First, such flow can be induced by background heat leaks in bath-cooled systems and the possible enhancement of such flows by "chimneys" or other thermosyphon effects have apparently not been considered for helium-cooled coils as they have for other fluids and systems. Secondly, heat transfer in low L/D channels (e.g., pancake coil flow passages) have perhaps not been adequately explored. For supercritical helium this is essentially a transient heat transfer problem, related to boundary layer formation. For bath-cooled magnets, this is related to item (4) below. Finally, limitations, if any, to such analyses in very tall systems where substantial gravitational pressure differences exist have not been considered.

(4) Fluid flow has relatively little effect on nucleate boiling, but there is some evidence that it has relatively substantial effect on the transition to film boiling. This, in turn, would have a substantial effect on the cold-end-recovery stability analysis, and in fact may make forced two-phase flow an especially attractive cooling mode from a stability viewpoint. Study and documentation is needed.

(5) There is considerable interest in using varnished or coated conductors in order to secure favorable manipulation of the nucleate boiling characteristic and the transition to film boiling. This approach, first reported by Butler et al. [1970], has been recently given more study by Iwasa at MIT. It was felt that the perturbation in heat transfer due to the insulating film probably could be estimated knowing the conductivity and thickness of the film, at least for steady state. This needs to be tested for both steady state and transient heat transfer conditions.

(6) Interest is reviving in the possible use of superfluid helium for rapid removal of large heat pulses; maxima in the range 4 to 20 W/cm^2, as limited by helium properties, motion, and past history, were mentioned. Very little data on this high-flux regime exist in the literature.

It is our considered opinion that, of all the problems outlined, the one which requires the most lead time (because it is a difficult experiment) and which is most useful for stability studies without having to specify a conductor geometry, is the measurement of transient heat transfer. Our recommendations for this work are in section IV.A.

System Behavior

System behavior in bath cooling would be related to single phase natural convection or thermosyphon effects at low heat flux, and displacement of liquid by vapor formation at high heat flux. Natural convection effects can be treated analytically in simple geometries, but the complex of passage sizes, shapes, and orientations which one can expect in a real magnet vitiates the usefulness of generalized analysis not tied to that particular magnet geometry. The same is true, approximately, of vapor blanketing or two-phase fluid motion in a coil under high heat flux. Thus we do not recommend any separately funded, generalized research in this area for pool boiling systems.

In forced flow systems, on the other hand, time-dependent coolant flows, pressures, and temperatures from a general analytical method can be related quite unambiguously to the predicted performance of a specified coolant system. A number of published stability or recovery calculations belong in this category: Green and Saibel, 1968; Bald, 1970; Mori and Inai, 1974; Todoriki and Agatsuma, 1974; Tsukamoto, 1974; Hoenig, et al., 1975. However, all of these have been performed in the approximation that the coolant flow and pressure remain unchanged as its temperature does change in response to a heat input. In this sense they are applicable to small, fast transients, e.g., Wilson's transient stability analysis, but they cannot begin to explore the degradation in performance which would accompany larger or longer-lasting perturbations. The reason that coolant flow and pressure perturbations have previously not been included is probably because of the mathematical complexity of the problem.

At the present time it appears that the MIT group, supported by ERDA-DMFE and OCR (Office of Coal Research)-MHD, is making good progress on the problem, and we do not see the need for additional funding through our program. It is strongly recommended that their existing funding for this work be continued. This point is outlined in IV.G.3.

The Possibility of System Oscillations

The tendency of liquid helium cooling systems to develop flow, pressure and temperature oscillations has been referred to in the LCPTF study, and it has become of sufficient concern to superconducting power transmission line programs that our laboratory has been funded for review and study of the phenomenon.

Three distinct modes of oscillation in forced flow systems can be identified from a large body of literature on other cryogenic and high temperature fluids. These are:

(a) density wave oscillations, which have a period approximately equal to the residence time of the fluid in the channel (Friedly, et al., 1967; Jones and Peterson, 1975).

(b) acoustic or "organ pipe" oscillations with a standing wave set up between flow discontinuities (e.g., a valve) (Thurston, Rogers and Skoglund 1967; Krishnan and Friedly, 1974).

(c) pressure drop or Helmholz oscillations, analogous to a mass on a spring (Maulbetsch and Griffith, 1969).

Provided the time constant for radial thermal diffusion is small compared to the period of oscillation, these can all be studied by the application of classical linear stability theory to the one dimensional conservation equations of mass, energy and momentum describing flow in the heated channel. But of equal importance to the dynamics of the channel itself are the external flow impedances or acoustic impedances at inlet and outlet of the channel since they determine the manner in which propagated disturbances are reflected or transformed.

We have made calculations for type (a) for supercritical helium systems. We find that under certain circumstances (e.g., low pressures and proximity to the transposed critical temperature) these oscillations are a real possibility in supercritical helium (Jones and Peterson, 1974). Stability computations for acoustic oscillations, type (b), are considerably more difficult and those that have been reported (Krishnan and Friedly, 1974) have not permitted generalization of the results. We, therefore, still lack criteria for this type of oscillation that can be applied to helium systems.

We have studied the possibility of negative differential flow resistance in helium cooled channels and find that this is only a possibility at inlet conditions of below about 3.5 K and below about 3 atm for short channels with high heat flux. Thus, oscillations of type (c) may be of very limited possibility in applications of superconductivity, but other possibilities for sustaining this mode of oscillation need investigation.

We recommend that the possibility of system oscillations should be investigated as part of the development of any specific system design, but we do not recommend a generalized study at this time.

III.B. Refrigeration

Over the past fifteen years, there have been many improvements and advances in low temperature refrigeration technology. First of all, the analytical helium equation of state [Mann, 1962; McCarty, 1972,1973] has permitted fast, convenient thermodynamic cycle studies yielding accurate results. Material and fabrication developments have improved the quality of the machinery while better instruments and controls have made refrigerator operation less labor intensive. All of these events have been the result of the growth in the use of superconductors for one application or another. The early superconducting devices must have seemed large to the scientists at that time but the MFE superconducting magnets, power transmission cables and particle accelerators under development now will cost millions of dollars and are part of progress important to the nation. Therefore, in our opinion, it is imperative that the supporting refrigeration equipment be the best that modern technology can produce.

Two important problems exist today; first, we must strive to make the refrigerators as reliable as any other part of a system and second, we must continually emphasize that for maximum efficiency the refrigerator and the device to be cooled must, insofar as possible, be designed to complement one another. We divide this technology element into several subcategories.

Refrigeration Cycles.

The optimum design of a cryogenic refrigerator or liquefier is a complex thermodynamic problem, primarily due to the non-ideal properties of the working fluid. The variable specific heats of the real gas can yield designs unknowingly involving negative heat exchanger temperature differences which are not uncovered by a simple first-law analysis. Variable properties also require optimization of pressure ratios and inlet temperatures for the various components used in the system. The usual approach to cycle studies is to make a parametric study of the cycle using a computer and then choosing one of the calculated cycles that is compatible with available components and the operational requirements of the particular installation. Although this approach gives exact results, one is never quite sure that a different (and unconsidered) cycle might not yield a superior efficiency.

The NBS Cryogenics Division has developed a method for calculating a maximum possible efficiency for liquefiers as a function of component efficiency. Reductions in efficiency due to system pressure drop and heat exchanger temperature differences were also included in the analysis. Recently, under this contract we have expanded this work to include refrigerator as well as liquefier efficiencies, and to include temperatures down to 4 K. This work is presented in Part Three of this report, and will be submitted to Cryogenics for outside publication. This analysis can serve as a guide to the real gas analysis by providing an ultimate refrigerator or liquefier efficiency based on the available component efficiency. It also shows how the efficiency of each component in the cycle effects the overall efficiency so the economics of additional research to improve the efficiency of a particular component can be evaluated.

We do not believe that further work needs to be done in this field for MFE systems at this time.

Refrigeration Reliability.

In 1973, NBS was commissioned by the Fermi Laboratory to visit the five large industrial helium plants in Kansas and Oklahoma to gather data and assess the operating history of helium expansion machines. Two of the facilities separate helium from natural gas only, while there are four helium liquefiers at the other three locations. These, the world's largest helium liquefiers, are roughly equivalent to a refrigerator of up to 3 kW capacity at 4.2 K. The unpublished result was one of the first known surveys of operating and failure history for helium liquefier expansion engines, heat exchangers, compressors, purification equipment and auxiliary equipment of this size.

Recently, ERDA-Power Transmission let two identical contracts to private industry to assess and study existing concepts and methods of cryogenic refrigeration for superconducting transmission cables. As part of the work, many of the major laboratories in this country and Europe were visited or contacted to gather operating and failure histories of helium refrigerators and liquefiers [Kadi and Longsworth], [Manatt, et al.]. Thus, the failure rate data has been extended and much other useful information was presented. In addition, both reports concluded that a refrigerator reliability data bank should be established to collect as much information as possible so that the maximum benefit may be realized from the reliability engineering sciences. NBS has also informally suggested on past occasions that this data bank should be established.

NBS began work on this data bank during the present fiscal year. The project is specified in more detail in Section IV.C. of this survey, and a year end report is given in Part Four of this report.

Compressors

Most helium compressors are converted from air or refrigeration service. Presently those in use are oil lubricated reciprocating or screw and non-lubricated reciprocating compressors. The lubricated machines always present the potential for contaminating the refrigerator cold box internals even though oil removal systems are installed. The non-lubricated reciprocating compressors typically have piston ring life of 90 days.

The screw compressor is a rotating, positive displacement machine capable of respectable pressure ratios in a single stage. However, for such machines there is no long term history in helium refrigeration service and, at present, screw compressors appear to be less efficient than reciprocating compressors. The high pressure ratio per stage without excessive heating is accomplished by pumping relatively large amounts of cooled oil through the machine along with the gas being compressed. Therein lies a potential problem. For continuous operation without fouling heat exchangers, piping, turbine blading and valves, the helium working fluid must be very pure. Various estimates of the allowable oil impurity fall in 10 to 100 ppb range. While tests show that current oil removal apparatus can achieve those low levels, oil carryover into the cold box would have the following effects: The oil will foul or poison the adsorbers intended to remove water and CO_2 so those constituents may pass on and poison the lower temperature adsorbers intended to remove N_2 and O_2 and so on until the entire purification stream is ineffective. The potential for oil contamination along with all the steps taken to eliminate that possibility simply must be weighed against the advantages of smaller size, lower cost, turndown capability and the probable long life that has already been demonstrated by screw compressors in fluorinated hydrocarbon applications. Screw compressors have only recently been used for helium compression and should be attractive for refrigerators requiring a few hundred kilowatt input power (the largest screw air compressor that we are aware of is about 800 kW). Screw compressor and the required oil removal development for helium service should be carefully watched. However, the much larger capacities which will be demanded of EPR refrigerators forces consideration of much larger compressors, i.e., turbine compressors, as used in industrial plants.

At room temperature and up to the moderate pressures used for helium refrigerators, helium behaves very nearly as an ideal, monatomic gas. The ratio of specific heat at constant pressure to that at constant volume is 1.67 for helium a compared to 1.40 for the diatomic gases such as oxygen, nitrogen and hydrogen. The high specific heat ratio and low molecular weight of helium present two compression problems.

1) The specific heat ratio determines the temperature rise of the gas upon adiabatic compression. The larger the specific heat ratio, the higher the temperature for a given compression ratio. In all compressors, the temperatures must be kept below the oil cracking temperature if lubricated or below material limits in any case. Therefore, the permissible compression ratio per stage for intercooled helium compressors is limited relative to other gases and more stages of compression are required for the same final pressure.

2) Rotating centrifugal compressors for high flow rates are attractive because of demonstrated long service life. To the first order, the pressure ratio obtainable per centrifugal stage is dependent on the molecular weight of the gas. For helium, with molecular weight 4, from 8 to 14 intercooled stages would be required for centrifugal compression from one to 15 atmospheres.

At this time, both centrifugal and regenerative compressors have been proposed for helium service in the larger capacities. A regenerative compressor is a rotating machine that is capable of higher compression ratio per stage than a centrifugal machine and like the centrifugal can be oil free in the helium spaces. Improvements in the aerodynamics and a preliminary design for 1270 kw helium compressor modules has recently been reported by Sixsmith.[1976]

In many ways the helium centrifugal or regenerative compressors will be very similar to existing machinery. The bearings, seals, lubrication and control systems should be state-of-the-art; however, the aerodynamic design and performance with helium have not been proven for machines of the size that will be required for MFE systems.

Our recommendations for compressor development are given in IV.C.

Liquid Helium Pumps

The use of low temperature helium pumps in force-cooled systems is planned in the LCPTF project and has been carefully analyzed by Bejan [1976]. The general conclusion from his work is that economic cooling systems must employ efficient cold pump recirculators in which the large flow demanded by the force-cooled superconducting magnet is confined to the cold end of the refrigerating column; further, if the liquid helium pump efficiency is less than 40%, systems employing elevated temperatures are more economic. Centrifugal helium pump efficiencies in the 40% range have been reported by Ludtke [1975].

Several reports of low temperature helium pumps are available [Sixsmith and Giarratano, 1970; McConnell, 1973; Ludtke, 1975; Morpurgo, 1977]. In addition, Fermi Lab has obtained both centrifugal and reciprocating helium pumps on a commercial basis, and ORNL will be using a commercial reciprocating pump in the LCPTF. Aside from the NBS work (McConnell and Ludtke), very little documentation of pump performance exists, and cumulative operating experience is limited. The NBS work did show that centrifugal pumps behave quite predictably in liquid helium, and can hardly be made to cavitate. They should be quite reliable, though several stages may be required to obtain a useable pressure ratio for MFE systems.

Complete documentation and records of pump performance, reliability, etc. will be invaluable to the MFE goals. We do not recommend any separate pump testing program under this contract, but we do recommend that, when a particular project requires a pump, sufficient funding and effort be planned to faithfully document both its initial performance and long-term service. This should go into the refrigerator reliability data bank, and would be an important input to subsequent program planning.

Ejectors

Most helium refrigerators operate at 4.3 K or above. However, if the trend indicated by Purcell toward lower temperature operation is followed, future MFE systems may be required to operate at 3 K (nominally) in order to achieve higher field. The approach taken in the few refrigerators operating below 4.2 K has been to install several 'vacuum' pumps in series upstream of the normal compressor suction. Since the pressure is low, these 'vacuum' pumps must have high volumetric capacity -- typically Roots blowers are used. Because of this added equipment, both capital and operating costs are greater than for a conventional system. More importantly, the potential for contaminating the helium stream is much greater since air will enter the reduced pressure system at any leak in the equipment (e.g., at rotating seals). Freezing of contaminant species is a leading cause of cryogenic refrigerator failure. Reliability is also compromised with the use of mechanical blowers.

An expansion ejector is a small device which functions as a low temperature pump driven by the potential energy of the high pressure stream, which is normally wasted in a Joule-Thomson valve. The ejector, which has no moving parts, is in the vacuum space of the cold box as is all the subatmospheric piping. Therefore the risk of contamination is reduced to the ordinary problems associated with low temperature refrigerators operating at pressures greater than one atmosphere. Use of an ejector would also eliminate the need for the large room-temperature vacuum pumps mentioned above and therefore improve system reliability.

The ejector appears to offer substantial practical advantages for refrigeration systems operating below about 4.5 K. However, very little has been published on the performance of helium ejectors -- certainly not enough to reliably calculate their expected performance on large refrigeration systems. The most extensive low temperature performance data published reports work done at NBS with nitrogen as the working fluid [Daney, McConnell and Strobridge, 1973]. Recently Quack [1977] has suggested that ejectors may advantageously replace pumps in many applications.

We have recently designed and installed two small ejectors on one of the refrigerators for testing ISABELLE magnets at Brookhaven National Laboratory. With it, this particular refrigeration unit reached 3.6 K (c.f. a predicted value of 3.5 K) whereas before the ejector was installed, the minimum attainable temperature in the same test had been 6 K [Strobridge, 1977].

The important milestone in this work is to determine whether the straightforward ejector design now being pursued produces satisfactory performance and design data for refrigeration below 4.2 K. Several more tests will be required to reach the milestone, a convenient stopping point if all goes well. It is important that this work be completed, but we do not recommend additional work over that already planned for ISABELLE. This project is listed in section IV.G.5.

<u>Controls</u>

The refrigeration control system for a complex, multi-magnet MFE system will have to be more reliable than for, e.g., a bulk liquefaction plant, and it seems quite clear that advanced process-control technology will be required. There are two parts to this problem. The first relates to the availability of accurate pressure, temperature, liquid level, flow, etc. sensors, as is discussed in section III.C. The second part involves the rapidly developing technology of minicomputer or microprocessor controls. This latter topic has been discussed in the refrigeration reports of Kadi and Longsworth, and Manatt, et al. The conclusion from these reports is that the problem must be recognized and appropriately funded when refrigeration systems are designed, but that, in the absence of a specific application, a separate study of control systems would not be appropriate at this time.

III.C. Instrumentation

The development of MFE systems will require extensive stage by stage testing in order to evaluate the design, performance and safety of superconducting magnet systems, and it is quite apparent that accurate and reliable instrumentation will be required to monitor and evaluate these tests.

The instrumentation considered in this survey is essentially that associated with the refrigeration. Thus it does not include the important topic of quench detection (local normal region detection) in the superconducting magnets. This latter topic deserves separate consideration, as it was given, for example, at the Los Alamos workshop on magnet stability.

The environmental conditions to which the measurement devices will be subject are high magnetic fields (8-10 T), if located on or near the magnet, and low temperatures 2.2-20 K) as a result of the liquid or supercritical helium coolant. Walstrom [1976] has discussed typical variables to be measured and has classified them as: (1) controlled or input variables which may be set by the test engineer for normal operating conditions, (2) diagnostic variables which indicate the response of the system to the input variables and (3) alarm variables which are a "subset of the input and diagnostic variables and which provide input for initiating a protective shutdown of the coil system in the event of a quench or failure of a critical facility or test coil component."

Table III.1 is a summary of the above variables to be measured with the appropriate instrumentation. In Table III.2 we present the anticipated ranges of operation and accuracies desired for all the instrumentation which has been considered in this MFE study.

The assessment of instrumentation for pressure, flow, liquid level, and density measurements at helium temperatures has been based on our experience with these types of measurements here in the Cryogenics Division of NBS-Boulder plus extensive evaluations and surveys of instrumentation already done for other cryogenic fluids. The publications which contain these previous evaluations are:

(1) Arvidson, J. M. and Brennan, J. A., ASRDI Oxygen Technology Survey, Vol. III: Pressure Measurement, NASA SP-3092, 1975.

(2) Sparks, L. L., ASRDI Oxygen Technology Survey, Vol. IV: Low Temperature Measurement, NASA SP-3073, 1974.

(3) Roder, H. M., ASRDI Oxygen Technology Survey, Vol. V: Density and Liquid Level Measurement Instrumentation for the Cryogenic Fluids Oxygen, Hydrogen, and Nitrogen, NASA SP-3083, 1974.

(4) Mann, D. B., ASRDI Oxygen Technology Survey Volume VI: Flow Measurement Instrumentation, NASA SP-3084, 1974.

Table III.1. Typical variables to be measured in MFE magnet system test or operation.

Controlled (or input) variables to be measured	Remarks	Diagnostic variables to be measured	Remarks
1. Test coil current		1. Deflections and strains in coil support structure and windings	For indication of mechanical behavior
2. Background coil current		2. Sudden local conductor movement	
3. Heater current		3. Conductor resistance	
4. Pulsed winding current		4. Conductor temperature	To detect normal zones
5. Magnetic field		5. Helium cryostat pressure	
6. LHe cryostat temperature 7. LHe cryostat pressure	Pool boiling system	6. Pressure and temperature downstream of inlet for forced cooled systems	To measure response to inlet condition changes
8. LHe flowrate 9. LHe inlet temperature	Forced convection cooling systems	7. Cooldown measurements a. temperatures in the coil windings and coil support structures	
10. LHe inlet pressure		b. Coil ground current detectors	To monitor system for electrical insulation failure
11. LHe cryostat liquid level	Pool boiling system	8. Facility related measurements a. electrical lead resistance b. vacuum level c. coolant supply line temperature	

Table III.2. Instrumentation operating range, accuracy and time response.

Parameter	Range	Accuracy	Time Response	Basis
Temperature	2-10 K	.01 K	1 sec.	detect slow drifts under normal operation
	2-300 K	1 K	1 sec.	monitor cooldown
Pressure	0.5 to 20 x 10^5 Pa	2%	.05 sec.	detect incipient oscillations or instability over the normal operating range
Flow Rate	1-300 g/sec. (with different instruments)	3%	few seconds	min. is determined by probable flow rates through individual channels max. is determined by largest refrigerator compressor.

(5) Brennan, J. A., Stokes, R. W., Kneebone, C. H. and Mann, D. B., An Evaluation of Selected Angular Momentum, Vortex Shedding and Orifice Cryogenic Flowmeters, NBS Tech. Note 650, March 1974.

The above instrumentation assessments are comprehensive in scope and contain e.g., descriptions and operating principles of all the traditional and more recent devices known to provide the previously mentioned measurements in cryogenic applications. Performance capabilities, encompassing probable precisions to be expected, ranges of operation, potential problem areas in application of the devices and other information of interest to potential users are also included.

Careful consideration of these documents and our experience with measurement systems operating at cryogenic temperatures lead to the following:

(1) The state of the art in thermometry and liquid level or phase detection is sufficiently advanced for applications in MFE systems. Arp, et al. [1975] have investigated the sensitivity of thermometers to electromagnetic fields and Sample and Rubin [1976] have provided a later summary of errors in thermometry induced by magnetic effects. These papers, along with the evaluations of Sparks [1974] and Roder [1974] provide an adequate basis for selection of all but the newest sensors, e.g., carbon-in-glass or capacitance types. It is recommended that some of these new types be tried by ongoing projects, but a separately funded study of them is unnecessary.

(2) The state of the art for density determination is somewhat limited both in documented cryogenic (particularly liquid or supercritical helium) experience with densimeters and in commercial availability of these instruments. However, simultaneous temperature and pressure measurements along with an adequate equation of state, e.g. McCarty [1972], provide the density. Therefore, the lack of significant development for densimeters poses no particular problem for successful MFE system instrumentation (provided, of course, adequate thermometry and pressure transducers are available).

(3) Pressure and flow are the measurements that are potentially the most difficult to achieve accurately and reliably in MFE systems. Although the difficulties, primarily in calibration and shifts in calibration due to temperature effects, are not unique to helium temperatures, they nevertheless are unresolved.

In view of the above we have concentrated our attention in this report on pressure and flow measurements and have provided recommendations to alleviate the problems areas.

III.C.1. Pressure measurement. Pressure measurements in helium cooling systems may be accomplished in either of two ways, (1) by bringing pressure taps out from the low temperature region to room temperature, where a wide variety of conventional instrumentation may be used, or (2) by placing a transducer at low temperature, in the helium system, and bringing an electrical signal out to room temperature. It is common to experience difficulties with method (1), e.g., the temperature gradients in the pressure taps promote thermoacoustic oscillations, variable thermal gradients show up as a drift in the output, and lengthy pressure lines may easily reduce the frequency response to below acceptable limits. Coolant supply line pressure, vapor bath pressure, magnet system coolant inlet and downstream pressures may also be monitored. In addition, an important and practical method for measuring helium flow is to use a venturi or orifice plate in conjunction with an appropriate pressure transducer.

Temperature effects.

Changing temperature conditions are the origin of many of the difficulties associated with cryogenic pressure measurement. Dean and Flynn [1966] have analyzed the temperature dependence of the performance of non-self generating pressure transducers (any that require external power to carry the information, e.g., strain gauge and capacitance bridges, linear variable differential transformers, and potentiometers). The essence of the analysis is that:

(1) The temperature dependence of the sensitivity shift (change of slope of output vs pressure curve) is due primarily to the change in Young's modulus upon which the spring constant of the force summing element of the transducer depends. Secondarily, the expansivity of materials contributes to the sensitivity shift.

(2) The temperature dependence of the zero shift (bias of the output vs pressure cruve) is a function of the size and expansivity of the transducer components.

(3) Both sensitivity and zero shifts are worse under the condition of a temperature gradient across the transducer body.

Our experience with cryogenic pressure measurement and some commercially available transducers lead to the following suggestions for avoiding temperature effects on pressure transducer performance.

(1) First, we note that little can be done about Young's modulus temperature dependence for a given material but --

(2) Proper selection of materials for transducer components can minimize this effect as well as expansivity effects.

(3) Proper design can minimize the expansivity effects even though expansivity changes are unavoidable (e.g., in a potentiometric device described by Dean and Flynn the wiper arm acts as a cantilever spring to absorb the differential contraction (or expansion) instead of being moved along the variable resistor which would thus cause the sensitivity and or zero shift.

(4) Avoid extreme environmental conditions (but with a corresponding loss in response) by remote location of the sensor. For single phase fluid in the sensing lines the frequency response may be estimated from the following considerations:

a) tubing and tranducer may act as:

i) organ pipe (volume of pipe large compared to the transducer cavity) with the resonant frequency given by

$$f \doteq \frac{v}{4\ell}$$

where v is the velocity of sound in the measurand and ℓ is the tube length.

ii) Helmholz resonator (volume of transducer cavity large compared to the pipe, e.g. close-coupled installation) with the resonant frequency given by

$$f = \frac{v}{2\pi} \frac{\pi r^2}{(\ell + 1.7\ r)V}$$

where

r = pipe radius
V = volume of transducer cavity.

(5) Locate one flush mounted transducer in the extreme environment and another transducer at the end of a tube in a more temperate environment. The flush mounted transducer will provide the high frequency response (dynamic levels, e.g., with small piezoelectric transducer) and remote mounted transducer will provide static levels. The anticipation is that the temperature shock contribution to dynamic response will be small or of low enough frequency to separate easily.

(6) Employ a close coupled temperature regulated pressure transducer installation. A heater and thermostat would be required to regulate the transducer temperature, and pressure tap lines must be optimized so that sufficient response is obtained but with heat leak to cryogen minimized. Further, the transducer must be oriented such that the cryogen (if in liquid state) does not flow into the transducer cavity and insulation around the transducer should be employed. It is possible to accomplish the same result (i.e., minimize temperature gradients) without the heater and regulator but using the insulation. The effect is to increase the time constant of the transducer (it will eventually reach the temperature of the cryogen) and sensitivity and zero shifts will still be experienced but thermal gradient effects will be reduced.

Survey of commercial availability of pressure transducers for low temperature application.

In order to assess the instrumentation industry capability to provide pressure transducers operable at the temperatures of interest (< 20 K) we have sent requests for technical literature on cryogenic pressure transducers to more than

70 different suppliers. The source of potential suppliers was: The Thomas Register "Cryogenics and Industrial Gases Buyers Guide," and the listing of exhibitors at the Instrument Society of America Exhibit and Conference, Houston, Texas, 1976. Response has been sporadic (far from complete) and not always relevant to the inquiry. However, a report completed in 1972 assessing pressure measurement system for LOX and LH_2 systems [Hayakawa, 1972] has provided an indication of commercial availability of these devices.

Our impressions, as determined by our own experience with commercial pressure transducers and as determined by the responses to our request for technical literature, are that since 1972 (time of Hayakawa report) industry has not increased their capability in the area of cryogenic pressure measurement. Even more important, the test data and conditions of test for substantiating advertised performance is not readily available for assessment.

As mentioned previously, temperature effects on transducer performance are difficult to predict but are one of the largest sources of pressure measurement errors in cryogenic systems. There are several reports of tests in independent laboratories of commercially available pressure transducers (e.g.: Smelser, 1962; Kinzie and Murphy, 1967; and Horn, 1965). All these reports point out and substantiate the susceptibility of the devices to temperature effects. Furthermore, Kinzie and Murphy noted a substantial difference in performance of transducers tested when sensing the pressure in a flowing cryogen compared to pressure in the same transducers sensing the pressure of a static cryogen.

Typical thermal sensitivity shifts (TSS) and thermal zero shifts (TZS) for commercial pressure transducers range from 0.009% F.S./K (0.005% F.S./°F) to 0.05% F.S./K (0.03% F.S./°F) [Hayakawa, 1972]. Under the assumption of a temerature change of 296 K (i.e., from 300 K to 4 K) then combined zero shift and sensitivity shift errors could range from aout 5% to 31% if no effort was made to recalibrate the transducer at the expected temperature of operation. Transient and thermal grandient effects would most likely increase the error.

Resolution capability and other performance parameters of commercial pressure transducers are generally adequate for purposes of MFE systems but space limitations may restrict the use of some of the instruments.

<u>Survey of literature for pressure measuring devices designed for/or operated at liquid helium temperatures</u>.

We have reviewed the technical literature for "laboratory" pressure measuring devices which have been designed for or operated at liquid helium temperatures but which are not commercially available. A summary of the review is given in tables III-3 and III-4. It was not possible to determine from the papers all the parameters which generally represent performance capability of pressure transducers; thus, the tables are not complete for all parameters.

All of these devices appear to fall into two general groupings. The first group (table III-3) are instruments employing pressure ports conducive to engineering application. This group also all use capacitance as the general operating principle (either in direct measurement of capacitance change with pressure, or frequency change of LC circuit as capacitance changes with pressure).

Table III.3. Pressure measuring devices operable at liquid helium temperatures (group 1).

Group 1	1. [Arp & Frederick, 1977]	2. [Chester, et al., 1968]	3. [Gonano, 1970]	4. [Landau, et al., 1970]	5. [Straty & Adams, 1969]
Operating Principle	Microwave oscillator	Capacitance	Capacitance	Capacitance	Capacitance
Measurand Range (Pa)	0-5.065 x 10^4	10^{-1} to 10^6	1.33 x 10^2-10.7 x 10^2	-2.66 x 10^4 to + 2.66 x 10^4 differential pressure at 2.026 x 10^5 base pressure	up to about 10^7
Power Dissipation (microwatts)	25	0.01	--	less than 0.01	negligible
Non-Linearity % full scale output	≈5%	≈6% (estimated from graph)	0.1% at and below 1.333 x 10^2 Pa. Linear between 1.33 x 10^2 Pa and 10.7 x 10^2 Pa	≈13% (estimated from graph)	--
Hysteresis	negligible	--	--	5 x 10^{-3} % FSO	not observable
Resolution	Limited by electronics to about 1:10^5 full scale	1.3 x 10^{-1} Pa	9.33 x 10^{-4} Pa	1.333 x 10^{-1} Pa	2.026 x 10^{-1} Pa
Full Scale Output	0-10 V	--	--	± 13 pF @ ± 2.66 x 10^4 Pa	50 pF
Temperature Range	0-300 K Best below 20 K	2-300 K	3-300 K	1-300 K	0.013-300 K
Thermal Sensitivity Shift (TSS) and Thermal Zero Shift (TZS) (percent full scale output per K)	TSS: Approx. 0.1%/K -- between 77-20 K TZS: Approx. 0.03%/K between 300-20 K, much less below 20 K		TSS: = 0 between 77-3 K, = 2.5% between 300-77 K	TZS: 0.002%/K	TSS: ≈0, below 4 K, quite large on cooling from room temp. to 4 K
Weight and Size	Wt.: Approx. 60 g Size: 1.95 cm dia. x 3.81 cm long	Wt.: -- Size: 3 cm dia. x 5 cm long	Wt.: } Large compared Size: } with other instruments in this table.	Wt.: -- Size: 2.1 cm x 2.5 cm long	--
Sensitivity	0.0002 V/Pa	38-77 Hz/Pa @ 10 MHz and an ambient pressure of about 1.3 Pa	1.05 x 10^{-4} ΔR/Pa where ΔR is change in capacitance ratio	1.2 x 10^{-4} pF/Pa	9.87 x 10^{-6} pF/Pa
Materials of Construction	Cu	Monel base, 304 SS diaphragm	Used commercial Barocel unit	Body: Silica doped epoxy resin Diaphragm: Mylar with one side coated with gold	Be-Cu

Table III.4. Pressure measuring devices operable at liquid helium temperatures (group 2)

Group 2	6. [DeSorbo, 1971]	7. [Genshiro, et al., 1971]	8. [Itskevich, et al., 1967]	9. [Lutes, 1967]	10. [Stankiewicz & White, 1971]	11. [Wright & Frank, 1975]
Operating Principle	Faraday magneto-optic effect	Superconducting transition temperature	Superconducting transition temperature	Superconducting transition temperature	Resistance (carbon resistor)	Resistance (tunnel junction)

The second group (table III-4) are "devices" whose physical properties change with strain (or pressure), e.g., carbon resistors and tunnel junction resistance, or devices which rely on pressure dependence of phenomena such as superconducting transition temperature or Faraday magneto optical effect.

Generally, these phenomena are weakly dependent on pressure and/or require sophisticated equipment for their operation (e.g., the change in superconducting transition temperature of tin with pressure, $dT_C/dP = -4.5 \times 10^{-5}$ K/atm and the need for optical system for Faraday magneto-optic device). For these reasons Group 2 pressure measuring devices are not considered pratical for the application at hand and only the operating principles are given in table III-3.

Returning to a discussion of table III-3, we shall note first that they all have been designed and built specifically for liquid helium temperature operation. A careful selection of materials and carefully controlled fabrication techniques have been employed in their construction. The matter of fabrication is not trivial since leaks that could develop if care is not exercised in construction would render the transducer useless for pressure measurement.

This group is also characterized by low power dissipation (attractive for use in helium systems) and good resolution for high accuracy measurements.

While all these instruments have been designed for and have been demonstrated to be successful in operation at helium temperatures, we note that temperature effects on sensitivity and zero bias are still a potential source of error in pressure measurement if room temperature or liquid nitrogen calibrations are used for the helium temperature operation. (For changes of temperature between 300 K and 4 K, the sensitivity variation ranges between 2.5% to "quite large" for these instruments.)

Finally, the small size of instruments (except for #3) is compatible with the anticipated space restrictions imposed when used in MFE systems.

Magnetic field effects.

Although magnetic effects were not to be considered in great detail in this preliminary instrumentation assessment it is apparent that since some of the pressure transducers may be required to operate in a high magnetic field region (8-10 T) this effect will have to be considered in the selection process.

Briefly, the magnetic effects may be considered as intrinsic or extrinsic. There is an intrinsic effect if the operating principle of the sensing device is inherently dependent on magnetic field (e.g., superconducting transition temperature, magnetic reluctance, and resistance). There may be an extrinsic effect if proper material selection is not exercised in the construction of the device. For example, even though the operating principle of the sensing device is insensitive to magnetic field (such as capacitance devices) the improper use of magnetic materials in the construction may distort the field as well as introducing errors due to unanticipated body forces (and strains) induced by the high magnetic field on the transducer components. Thus possible magnetic effects are:

Intrinsic effects of magnetic field on:

Superconducting transition temperature
Magnetic reluctance
Resistance

Extrinsic effects of magnetic field:

May induce unanticipated body forces if improper materials used
Eddy current heating may introduce error if device is temperature sensitive
(A.C.) Electromagnetic noise may introduce signal processing difficulties (e.g., trying to record low-level output signal in electromagnetically noisy environment.

Either of the above effects will introduce error in pressure measurement and therefore instruments susceptible to them must be avoided.

Summary of Pressure Measurement Problems

The primary obstacles to selection and/or application of pressure transducers for operation in MFE systems are:

(1) Important detailed calibration data are not available on the few instruments which may be suitable, e.g.,

   a) Temperature change effects on sensitivity and zero.
   b) Temperature gradient effects.

(2) Insufficient information on optimization of installation configurations to reduce temperature effects and maintain required time response.

(3) Lack of standardization of cryogenic test procedures for uniform evaluation of existing or proposed cryogenic pressure transducers.

(4) No readily available source of potential commercial suppliers of cryogenic pressure transducers with accompanying test data to substantiate advertised performance; no readily identifiable commercial pressure transducers with proven performance.

(5) Lack of information on extent of magnetic field effects on transducer operation. (For applications where the transducer would experience high magnetic fields.)

Our recommendations for work in this area are given in IV.D.

III.C.2. Flow measurement. Helium flow through MFE systems may often be split into parallel channels so as to keep the overall hydraulic resistance or temperature rise within acceptable bounds. In such cases, it is necessary to monitor the individual flows since the development of a plugged flow channel may not otherwise be evident until a portion of the magnet unexpectedly goes normal.

Measurement of helium flow in the laboratory is not new, but the various measurement instruments and techniques have never been systematically documented and evaluated as they have, for example, for oxygen or LNG [Mann, 1974; Brennan, et al., 1974].

Very little data are available which are directly applicable to flow measurement of liquid or low temperature gaseous helium. Almond, et al., (1972) and Niinikoski, (1972) have reported flow measurement of helium gas (less than 1.5 mol/s in He^3-He^4 dilution refrigerator systems) using thermal flowmeter techniques, but the information presented is generally too brief to allow an overall evaluation of the flowmeter and the accuracy claims. For applications where long time constants and low accuracy requirements are sufficient, these flowmeters are probably quite adequate but at the present time there is no acceptable flow facility available for providing base line information on helium flow measurements.

For liquid helium flow measurements, there is a considerable amount of indirect data available from the space program to indicate some flowmeters that might be appropriate for liquid helium service. Predominate among the flowmeter types are the turbine and the head type. Turbine flowmeters have been used for nearly all the cryogenic fluids and enjoy wide acceptability in the space program for measurements of liquids hydrogen, oxygen and nitrogen. They are delicate instruments, however, and can be severely damaged during cooldown if allowed to spin at high speeds in vaporized gas. Turbine flowmeters nominally have a flow range of 10 to 1 and are available in a wide variety of sizes.

To prevent large errors, the calibration of a turbine flowmeter must be done in the fluid in which it will be used. Little success has been reported in efforts to calculate or otherwise predict the calibration of a turbine flowmeter using surrogate fluid calibrations. The same is true to some degree with all flowmeter types but more effort has been placed on this capability for turbine flowmeters than on most other types and with little success.

Vortex shedding and ultrasonic flowmeters might also be appropriate for helium service but no data is presently available for these flowmeters at temperatures lower than that of liquid nitrogen.

Head type flowmeters (venturis and orifices) are the most promising candidates for use in large refrigeration systems. However, for all but possibly the largest liquid helium flows, the meter diameter will probably be less than the 7.5 cm which is the smallest size for which ASME codes exist. ASME experience indicates that the calibration of smaller sizes cannot be reliably calculated from their physical dimensions, presumably because of normal machining inaccuracies.

Thus, absolute measurement errors of approximately 5% would not be unexpected in the use of this type of helium flowmeter, though measurement precision would be much better, nominally $\approx 2\%$ over its design range. When desired, the accuracy could be brought within usual limits by calibration, which normally for this type of meter is expected to be independent of fluid properties. However, helium has a high compressibility and extremely low viscosity compared to water, which is used for most industrial calibrations, and no experimental work has shown that an industrial meter calibration will apply to a head meter used in liquid helium. A small bit of theoretical work is being done under this contract on the expected sensitivity of venturi flow measurement to helium properties, especially near the transposed critical line, where neither incompressible fluid nor ideal gas properties are even remotely valid.

Head type flowmeters would benefit greatly if the required pressure measurements could be made at the flowmeter instead of at the room temperature end of the impulse lines. Pressure oscillations in the lines are one of the most serious problems with this type of flowmeter when used in cryogenic systems. Normally this type flowmeter is limited to about a 3 to 1 flow range by the range of the pressure measurement instrumentation.

The lack of a helium flow facility seriously impairs the state-of-the-art determination of helium flow measurements. As requirements for these measurements become more critical, the more serious the need for a facility will become. It would be well advised to develop a suitable facility sufficiently far in advance of the need to allow flowmeter manufacturers the necessary lead time to make any required modifications to their flowmeters. However, the expected cost of such a facility would be 0.5 to 1.0 M$, as judged by the cost of the NBS liquid nitrogen flow loop, and even with cost-sharing this may be difficult to justify from the DMFE budget.

Furthermore, as with pressure instrumentation, a readily available source of potential commercial suppliers of helium flow measurement devices does not exist.

Our recommended program to determine the accuracy of helium head-type flow calibrations using industrial water calibration facilities is outlined in IV.E.

III.C.3. Helium impurity measurement. A major cause of unscheduled downtime in helium refrigerators is the plugging by impurities. The LCPTF report and both refrigeration studies (Kadi and Longsworth, Manatt, et al.) point out the need for reliable impurity monitors. Accurate impurity level measurement requires a system which collects a representative sample and a detector which measures the impurity level. The sampling system must be designed to eliminate any change in the sample composition by addition of impurities or by precipitation of impurities. Ideally, the monitoring device should be in situ to avoid sampling problems. This device should be a continuous, on-line detector which can be set to sound an alarm if the impurity level exceeds a certain preset value; simple calculations show that this preset value must be in the range of 1 ppm or less if impurity plugging is to be eliminated.

Selection of the optimum type of impurity detector depends upon the impurity and whether it is in the gas phase (i.e., homogeneous mixture) or in a liquid or solid phase (i.e., heterogeneous mixture). Jensen and Harrison at BNL are attacking the problem of detecting oil droplets in the helium discharge from a screw compressor used for SPTL studies. They have a private company developing a light scattering device to detect oil droplets (or any other liquid or solid impurity) in the 0.1 to 10 micron size range. The system is promising and should be an important input to the MFE community; however, the project is moving along and requires no additional funding through this contract at this time.

To our knowledge, no one is actively trying to develop an on-line monitor for detecting impurities present as a gas. Duval at LLL uses a gas chromatograph containing a helium ionization detector to detect impurities in the 10 to 20 ppb range. However, the instrument requires a trained technician and frequent maintenance to keep it operating properly. Similar instruments are in use at BNL (SPTL project) and at SLAC.

There is an obvious need to evaluate the existing techniques for detecting trace gas impurities in helium to see if they can be adapted for continuous on-line service. Based on the need for this type of work we recommend initiating the research program described in section IV.A.

IV. PROGRAM PLAN

The following program plan is based on our assessment of the cryogenic engineering data base for MFE systems, as discussed in section III. It assumes a substantial DMFE committment totaling about $300,000 per year.

IV.A. Transient Helium Heat Transfer

The need for transient heat transfer with temperature controlled (rather than heat flux controlled) surfaces in order to understand and predict superconducting magnet stability was a dominant theme at the LASL workshop, July 25-29, 1977.

When heat is suddenly applied to a surface cooled by helium, a finite time is required to build up the fluid convection or vapor content to its steady state value. This time interval ranges from 0.01 to several seconds during which the temperature difference and heat transfer rate will differ significantly from the values finally achieved in steady state. The problem is that flux jumps, training steps, etc. typically occur on a much shorter time scale, so that regime is somewhat independent of fluid motion, i.e., pool boiling vs forced convection, so that this type of data are readily applied to both cryostability analysis [Dressner, 1976] and transient stability analysis [Wilson, 1976].

Relevant heat transfer measurements for these analyses have been reported by Iwasa and Apgar [1976] at MIT, and Steward [1977] at NBS. The MIT measurement used both bare and insulated copper for their heat transfer surface, pool boiling helium at 1 atmosphere, and relatively slow response, about 1 millisecond. Recently, at the LASL conference, they indicated that some of their measurements were of uncertain accuracy because of a thin film of helium leaking into the insulation between their heater and their heat transfer surface. The reported NBS measurements covered a much wider range of helium conditions, geometry effects, and times (down to 20 microseconds), but used a carbon film heat transfer surface. The current NBS activity is devoted to producing a temperature-controlled copper surface in contact with the helium.

The temperature-controlled heat-transfer surface, as recommended by the LASL workshop, is required so that the negative slope portions of the q (heat flux) vs. ΔT (temperature rise) curves can be traced out as a function of time. This "negative resistance" portion of the heat transfer curve is of particular importance in understanding the recovery process of superconductivity following a localized energy release. It can show up as hysteresis if q rather than ΔT is controlled in the experiments, and there is some question as to its magnitude during transient conditions. Its magnitude during steady state conditions possible can be inferred from existing measurements, but it has never been well documented for liquid helium.

The AFOSR has indicated that they will support this project with $40K in FY 78, which is one-half of the full time cost for the project. Their interests and objectives are coincident with those of the MFE stability analysis, with one exception: they are interested in faster response times (20 microseconds or less, simulating fast magnet pulses) than are necessary for MFE work (approximately 1 millisecond). We have already achieved this fast response.

We propose that ERDA-DMFE support this work with an additional $40K, so that full progress can be made in this important and timely project.

IV.B. Helium Impurity Detectors

The objective of this work is to develop an on-line system which continuously monitors trace gaseous impurities in helium refrigeration systems.

As a guide to selecting the proper detector, those developed for gas chromatography are the prime candidates, but other detectors also will be evaluated. Each detector will be judged on its sensitivity to composition, temperature, flow and pressure fluctuations. Based on the results of this phase, the most promising detectors will be evaluated experimentally.

Methods for obtaining a representative sample from a flowing helium stream and for controlled concentration of impurities will be considered. If concentration techniques are feasible, it will offer the possibility of using more reliable and rugged detectors. These sampling problems will be cost-shared where feasible with an on-going project supported by a consortium of LNG (liquefied natural gas) manufacturers, who face the same type of problems in custody-transfer of LNG.

Cost (3 year project)

Preliminary screening and evaluation	$ 5K
Laboratory evaluation of most promising detectors, development as needed	80K
Study of performance on a real system: the problem of obtaining a representative sample	95K
	$170K

IV.C. Refrigerator Reliability Data

This project is intended to gather information on the reliability of helium refrigeration systems as they relate to needs and problems of the MFE program plan. Its output will be used to further define and delineate the critical elements in the operation of such systems, and thus to recommend procedures and/or necessary research.

The program is proceeding as follows:

1) Determine the extent to which NBS can legally be involved in such a program. (We are precluded, for example, from recommending a specific product.) Presuming that our proposed actions are judged appropriate, we would:

2) Determine exactly the data needed by reliability experts. At this time we intend that the reliability analysis be done by established experts in the field;

3) Determine how much interaction there can or should be with the Government-Industry Data Exchange Program (GIDEP) or other similar programs. We will take as much guidance as possible from the existing failure rate program.

These three steps have been completed in FY 77 (see Part Four of this report for a detailed discussion). The next steps are as follows:

4) Solicit the cooperation of managers of all possible helium refrigeration/liquefaction facilities;

5) Design the data collection system, taking into account suggestions from both GIDEP and facility managers;

6) Start the program with heavy NBS involvement which will be phased out if it is found that one of the other established failure rate progrms can successfully assimilate this project.

Progress in this program will have to be carefully assessed at intermediate stages in its development. We will also try to obtain the financial involvement of other agencies which would use the data when the system is operational.

Cost: $10K/yr

IV.D. Turbo Compressor Development

Based upon the potential for significantly increased efficiency and reliability of refrigeration systems, as compared with those using screw compressors, we recommend that a substantial program be initiated for room temperature helium turbo compressor development. This could take two directions:

1) Develop machines specifically for helium, a low density, high gamma gas. These could either be centrifugal, or of the regenerative type (which produce a higher compression ratio and might be easier to fabricate, though they are not so well known). The developed unit should be at least two stages with a throughput equivalent to an 800 kW compressor operating between 1 and 15 atm. The units should be fully instrumented to obtain all pertinent data and should be supplied with a test stand suitable for continuous operation.

Output from this program would be the design data necessary for private industry to build a practical, full size helium turbo compressor when needed for refrigeration service.

2) The Union Carbide-Linde Division presently holds a patent for a combined helium-freon refrigeration cycle which uses conventional centrifugal compressors. This has never been used to obtain 4 K refrigeration, to our knowledge, and one naturally wonders whether the helium-freon separation (which must occur at some intermediate temperature in the cycle) will be complete enough to avoid plugging the 4 K section with solid freon during extended operation. It is conceivable that phase equilibria data may be required to answer these questions.

Our inclination is to opt for the compressor development work, based upon the seemingly higher probability of reliable long-term operation. However, before making such a decision, the freon-helium approach needs much more careful consideration than we have given it to date.

We propose, therefore, first to enter into discussions with Linde concerning their degree of assurance that the freon-helium compressors will remain impurity-free for extended 4 K refrigeration service. Then, because we in the Cryogenics Division lack special expertise in turbo compressors, we will convene a panel of appropriate specialists from government laboratories and industrial experts to assess the complete picture and recommend a course of action which will best fit with the program goals. This panel will provide steering and guidance for the program throughout its life.

Cost:

Preliminary discussions, panel recommendations $ 10K

Subsequent development:

Quite uncertain, perhaps $200K/yr for 4 years.

This project would be by far the largest single item in the proposed program plan. Because it would be of considerable value to other large superconductive programs as well, every effort should be made to obtain cost sharing with them. Further, it is reasonably possible that industry, sensing the potential value, may be willing to engage in cost-sharing with this program.

IV.E Cryogenic Pressure Transducer Tests

This project has two parts (1) development of a calibration facility, and (2) screening and evaluation of transducers for liquid helium service.

The development of the calibration facility would be done at NBS, where design has already begun using NASA funding. This ERDA project would be responsible only for the incremental cost of necessary modifications to make the facility usable down to 4 K, rather than just down to 76 K, and to modify the external control system to achieve calibrations at up to 20 Hz at low base pressure appropriate to helium cooling systems.

In the screening and evaluation work, candidate commercial transducers would be tested. If it appears that small production changes, e.g., use of an alternate material for a given component, would result in superior cryogenic performance, these changes sould be tried and evaluated. Because of the relatively small market for cryogenic transducers, this stage of work probably would involve small "development" contracts as well as outright purchases.

The output from this project would be a quantitative assessment on cryogenic pressure transducers for use in MFE systems.

Cost:

(Prior NASA support	$ 55K)
Assembly, system tests, procedure development (ERDA share)	27K
Transducer purchases; small contracts; experimental evaluations; accuracy 2%	50K
(Additional work to prove the ultimate system accuracy -- not requested from DMFE	60K)

IV.F. Cryogenic Flow Measurement

The absence of any facility within the U.S. (and probably the world) for evaluating liquid helium flowmeters is due, at least in part, to the probable expense in setting up such a facility. The throughput of an 800 kW (1000 H.P.) helium refrigerator is about 2 liters/s at liquid density, or 300 g/s. A facility for calibrating liquid flowmeters at this flow rate would cost 0.5 to 1.0 M$, as judged by the cost of the NBS liquid nitrogen flow loop, and even with cost-sharing could hardly be justified out of the DMFE budget.

We propose a much more modest approach based upon head-type flowmeters with in-situ pressure measurement. The objective will be to show that an industrial water calibration can be used to predict a liquid helium calibration (hopefully) to within the nominal precision of about 2%. We do not anticipate that, for this accuracy appropriate to the MFE program, it will be necessary to test at more than one helium state, since the relevant properties (viscosity and conductivity) do not vary greatly except near the critical point and transposed critical line; in practice, the meters generally would not be used within this region. For convenience, the helium state can be chosen as boiling (or near boiling) liquid. The flow rate can be smaller than quoted above, provided that the meter size does not get too small for standard fabrication techniques to be used in building it; a design flow rate of 30 g/s represents a reasonable compromise.

A non-trivial problem with head-type flowmeters for liquid helium is that of obtaining an accurate ΔP reading. Long pressure taps running to an external (warm) transducer tend to oscillate (as discussed in III.C.). The cure for this often involves very high fluid impedances with consequent long run times and large helium use. Greater testing efficiency would be obtained if these tests were made subsequent to those in IV.E., so that a calibrated cold pressure transducer could be used. Likewise, this combination would be very practical for a working installation.

The procedure will be to build a modest helium flow loop capable of accepting the meter under test. The helium calibration can be obtained by liquid level measurement, motion of a piston or bellows, or possibly by warming to room temperature. A nominal accuracy of 2% would be a design goal. The installation would not be intended as a permanent standard.

Only a venturi meter should be tested, since these are to be recommended over orifice meters for their generally higher accuracy. These would be patterned after ASME standards for larger sizes. The meters would be calibrated in both an industrial water calibration facility and the helium flow loop.

The estimated cost of this project is as follows:

Design, build, and check out liquid helium flow loop for flowmeter tests for 2-3% accuracy.	\$80K
Build one venturi with cold pressure transducers, evaluate performance in helium flow and water calibration loops.	\$40K

IV.G. Other Programs

It is important to explicitly recognize and encourage certain on-going programs which are presently funded by other agencies, or already supported by DMFE. Their output and goals are of considerable importance to the MFE program, within the framework of this cryogenic engineering assessment. The previous recommendations IV.A to IV.F are made assuming these projects will continue. These include:

1) The Low Temperature Materials Program, also funded through our laboratory. Our program does not overlap with it, but shares many common interests and resources.

2) The helium impurity monitoring instrumentation being evaluated at BNL for the Superconducting Power Line project. This instrument is designed to detect entrained liquids or solids, rather than impurities in solution as recommended for study in IV.A. Obviously, these are complementary projects and close communication between the projects must be maintained.

3) The MIT studies on forced-cooling system response to a pulse heat input. These studies are most valuable in completing the more general analysis of cooling system requirements for magnet stability. Complementary activities within this area are funded by DMFE (through ORNL) and by OCR-MHD.

4) Low temperature pump usage and studies at ORNL for MFE, and at Fermi Lab for HEP. Both centrifugal and reciprocating pumps are in use at Fermi Lab. Pumps may be an important element in the LCPTF and other cooling systems in the future. Input to the GIDEP program is recommended.

5) The ejector studies now underway within the BNL ISABELLE program. These simple devices have the potential of obtaining increased system performance by lowered operating temperature, without the complications and expenses of subatmospheric refrigerator suction pressure.

The regular project review which will be held to assess the performance and direction of this cryogenic engineering program should explicitly include consideration of the above items.

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AFOSR	Air Force Office of Scientific Research
ANL	Argonne National Laboratory
BNL	Brookhaven National Laboratory
DEMO	a successor to EPR, nominally planned for 1995
DMFE	Division of Magnetic Fustion Energy
EPR	Experimental Power Reactor, nominally planned for 1985
FCCS	Forced circulation cooling system
GA	General Atomics Company
HEP	High Energy Physics (program of ERDA)
ISABELLE	Intersecting Storage Accelerator at BNL
ITR	Ignition Test Reactor
LASL	Los Alamos Scientific Laboratory
LCP	Large Coil Program at ORNL
LCPTE	Large Coil Program Test Facility
LLL	Lawrence Livermore Labotatory
LNG	Liquefied Natural Gas
MFE	Magnetic Fusion Energy
MHD	Magnetic hydrodynamic (program or power generation)
NBS	National Bureau of Standards
OCR	Office of Coal Research
ORNL	Oak Ridge National Laboratory
PB	Pool boiling
PPPL	Princton Plasma Physics Laboratory
SLAC	Stanford Linear Accelerator Center
SPTL	Superconducting Power Transmission Line
TFTR	Tokomak Fusion Test Reactor
TNS	The Next Step, a reactor prior to EPR
TSS	thermal sensitivity shift (of transducers)
TZS	thermal zero shift (of transducers)
U. Wisc.	University of Wisconsin
UWMAK	University of Wisconsin Tokomak

Part One, Appendix A

Agenda from the

WORKSHOP ON STABILITY IN SUPERCONDUCTING MAGNETS

JULY 25-29, 1977

NATIONAL SECURITY AND RESOURCES STUDY CENTER
LOS ALAMOS SCIENTIFIC LABORATORY

TECHNICAL PROGRAM

MONDAY

Chairman:	H. Laquer
8:30	Welcome to Los Alamos........................H. Agnew
	Introductory Remarks................W. V. Hassenzahl
9:00	Review of Cryostability..............Z. J. J. Stekly
9:45	Stability in Terms of Minimum Propagating Zones and Disturbance Spectrum..........S. L. Wipf
10:40	Coffee
11:00	Magnetic Instabilities and Quenching Currents in Superconducting Composites.....B. Turck
11:30	Theoretical Research on Stability at ORNL................................L. Dresner
12:30	Lunch
Chairman:	S. L. Wipf
1:30	Modus Operandi............................S. L. Wipf
2:00	What More is There to Know?.............M. N. Wilson
2:40	Tea
3:00	Working Session (Contributed Papers) - The Disturbance Spectrum
	Role of Acoustic Emission in Monitoring Stability in Superconducting Magnets..........O. K. Mawardi
	Mechanical Perturbation Studies at ANL...............................S. T. Wang
	Stability and Mechanisms Which Give Rise to Instabilities in Superconducting Magnets......................J. H. Murphy
	Measurement of Frictional Heating.........Y. Iwasa
4:00	Working Session (Seminar) - Can We Predict a Quench?
5:15	Bus Leaves (Bus returns in time for Cocktail Party)
6:00 - 7:30	Cocktail Party and Buffet, National Security and Resources Study Center

TUESDAY

Chairman:	John D. Rogers
8:30	Facing the Stability Problem in Magnet Construction........................J. Purcell
9:15	Development and Operating Experience on NbTi and Multifilamentary Nb_3Sn Magnets................................D. N. Cornish
10:00	Coffee
10:20	Working Session (Panel Discussion) - Stability in Large Coils: Engineering Implications Panel: R. D. Bradshaw, J. H. Murphy, D. N. Cornish, J. Purcell, H. Desportes, Z. J. J. Stekly, S. T. Wang
12:00	Stability Considerations and Safety Factors in Large Systems with Thermal Runaway Problems (Example: Fission Reactor)..........To Be Announced
12:30	Lunch
Chairman:	R. I. Schermer
1:30	Working Session (Contributed Papers) - Structure Versus Stability Pulsed Dipole Performance and Coil Structural Constraints.................W. S. Gilbert The Fermilab Tevatron Magnets.............M. Kuchnir The Isabelle Coils..................A. D. MacInturff Design Tradeoffs between Stability and Structural Requirements..........C. L. Linkinhoker Stability of the 300-kJ METS Magnets....J. J. Wollan
3:30	Tea
3:50	Working Groups (Parallel): Coil Monitoring (Wilson) Heat Transfer (Arp) Other groups to be established based on the interests of the attendees.
5:30	Bus Leaves

WEDNESDAY

Chairman:	J. J. Wollan
8:30	Review of Heat Transfer Including Transient Effects....................V. Arp
9:30	Working Session (Contributed Papers) - Response of Coil to Disturbances: Heat Transfer
	Transient Heat Transfer Measurements........Y. Iwasa
	Superconductor Stability in Narrow Cooling Channels........................J. W. Lue
10:30	Coffee
10:50	Working Session (continued)
	Stability as a Problem of Heat Transfer...G. Claudet
	Recovery of a Cryostable Magnet Following a Mechanical Perturbation.............L. R. Turner
	Boiling Heat Transfer Induced Instabilities in Large Superconducting Magnets.........J. W. H. Chi
	Cryogenic Stability Using Supercritical Helium as a Coolant....................M. A. Hilal
12:30	Lunch
Chairman:	M. N. Wilson
1:30	Working Session (Contributed Papers)
	- Hollow Conductor Systems
	Analysis on the Stability of Hollow Force-Cooled Superconductors................O. Tsukamoto
	A Computer Simulation of Normal Zone Propagation......................J. K. Hoffer
	Forced Flow Conductor Development at ORNL..............................J. R. Miller
	Cryostability in Forced Flow Cooled Cabled Superconductors................M. O. Hoenig
	Performance of SIN Muon Channel...........G. Vecsey
3:20	Tea
3:40	Working Session (Seminar)
	- Can We Improve Stability?
	Higher Temperatures for Better Stability..............................D. Atherton
5:15	Bus Leaves

THURSDAY

Chairman:	K. D. Williamson
8:30	Working Session (Seminar)
	- Can We Learn From Coil Failures?
	Operational Experience of Existing Superconducting Magnet Systems......S. Y. (David) Hsieh
10:00	Coffee
10:20	Working Groups (Parallel)
12:30	Lunch
Chairman:	S. L. Wipf
1:30	Working Session on Terminology and Definitions
	Introduction.............................A. F. Clark
	Small Discussion Groups
	(Group Leaders: Clark, Dresner, Laquer, Oberly, Strauss, Wilson, Wipf)
3:10	Tea
3:30	Synthesis of Definitions (All Discussion Groups)
4:30	Buses leave for *al fresco* supper at Bandelier National Monument

FRIDAY

Chairman:	W. V. Hassenzahl
8:30	Review of Working Group Results
10:10	Coffee
10:30	Consensus (?) on Terminology and Definitions
12:30	Lunch
1:30	Summaries of Conference
	Monday: S. L. Wipf
	Tuesday: J. D. Rogers
	Wednesday: H. Laquer
3:10	Tea
3:30	General Conclusions: Wilson
4:45	Bus Leaves

NOTES CONCERNING THE PROGRAM

The working sessions provide a forum to present the most up to date reports of current work relating to stability. They range from sessions for prepared contributed papers to the less formal seminars for plenary discussions, with panel sessions somewhere in between. All participants are encouraged to contribute in the working sessions.

Suggested topics to be dealt with in seminars include, but are not limited to, the following.

1. Can we predict a quench:

   Operational prediction versus design prediction.
   How can coils be monitored?
   Can something be done after a would-be-quench has already started?
   What is the use of distinguishing between quench current below and above short sample critical current?
   What information can be obtained from test coils?
   Can we use the propagation velocity?

2. Can we improve stability?

   High or low operating current versus stability. The role of electrical insulation.
   Advantages and disadvantages of hollow conductors and forced flow.
   Structural material versus copper.
   Fiddling around with parameters (spec. heat; heat transfer; thermal conduction; mechanical parameters; superconducting parameters.)
   Is training useful? Can it be achieved without quenching?

3. Can we learn from coil failures?

   Failure (quenches) of large coils: are they related to stability?
   Can we gain any criteria on disturbances related to coil size?
   Can we define a safety factor? If so, how does it depend on size?

Small working groups will be formed on Tuesday afternoon to discuss special topics under the leadership of a group chairman. These groups will work informally during the week and on Friday morning the group chairmen will summarize the results. Suggested topics for the working groups include the following:

Coil monitoring
Heat transfer
1.8 K operation of coil systems
Low field-high frequency or pulsed operation
Propagation velocity
Sources of disturbances in coils

The work on Terminology and Definitions will begin by dividing existing and new terms into three classes:

Class 1. Terms which are useful and well defined.
Class 2. Terms which are useful but not well defined.
Class 3. Terms which are either not useful, confusing, or to be avoided.

We want to establish a consensus on the definitions of the terms in Class 1.

Please bring relevant material.

Part One,Appendix B

Attendees at the

Workshop on Stability in Superconducting Magnets
July 25-29, 1977
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NBSIR 78-877, Part Two

HELIUM TRANSIENT HEAT TRANSFER

October 1, 1977

W. Gene Steward

1. INTRODUCTION

Transient helium heat transfer experiments were carried out at the NBS Cryogenics Division for two years under sponsorship of the Air Force Office of Scientific Research. In these tests thin carbon films deposited on flat quartz substrates served both as fast time response heaters and thermometers. The first phase of these tests, transient heat transfer to a static helium bath, is covered in detail in Appendix A which is a preprint of a paper to be published in the International Journal of Heat and Mass Transfer. A summary of these tests is as follows:

SUMMARY OF TESTS

Temperature Rise and Heat Transfer Coefficient as a Function of Time

	Static Helium	Forced Flow
Heat flux range .05 to 20 W/cm^2	X	X
Ambient pressure	X	X
Elevated pressure to 0.3 MPa	X	to 1 MPa (in progress)
Variable orientation	X	X
Repeated pulses	X	X
Transient decay	X	X

Forced convection transient heat transfer tests, which began under the AF funding, were continued under ERDA funding, as was the development of computer plotting of the data. Currently, work has started on a new type of heater-thermometer which employs heat transfer surfaces other than carbon. Looking toward a method of obtaining transient data in the form desired for stability analyses we have worked out a preliminary design for an electronic system to vary heater power with time in such a way as to produce scheduled rate of surface temperature change (either rising or falling).

The final section of this report discusses plans for future work.

2. FORCED CONVECTION EXPERIMENTS

The forced convection experimental apparatus is shown schematically in figure 1. The objectives of the apparatus design are to provide transient heat transfer data under conditions of flow and heating similar to those of forced flow cooled superconductors. Flat plate heaters of 0.5 cm wide x 1.0 cm long (in the flow direction) are inserted into a rectangular flow channel 0.15 cm deep. A length of uniform channel approximately 18 times the channel depth precedes the test heater to ensure a fully developed momentum boundary layer. The pre-heater shown is intended to establish the thermal boundary layer and thus avoid entrance effects. Power is supplied to the pre-heater at approximately the same power density as in the test heater. Further specifications are:

1) Liquid helium maximum flow rate with test loop installed -- 18 g/s. (maximum Reynolds number, $N_{Re,L}$ in the test section = $1.7(10)^6$)

2) Heater area - 0.1637 cm^2, pre-heater area = 0.322 cm^2.

3) Heater surface material -- vapor deposited carbon.

All of the instrumentation and data acquisition equipment are the same as in the static helium tests with the addition of the pump, variable frequency power supply, and flowmeter frequency counter. Data were recorded on magnetic tape and plotted by computer.

Test results shown in figures 2 through 9 are for heat fluxes of 0.1, 0.5, 5, and 10 W/cm^2, usually with nominal mass velocities of 0, 40, 120, and 240 $g/s{\cdot}cm^2$ which correspond to liquid Reynolds numbers of 0, $2.8(10)^5$, $8.3(10)^5$, and $1.7(10)^6$. Reynolds number is shown as the parameter here because some form of Reynolds number is the principle flow related correlating factor for forced convection heat transfer in steady state.

At the lowest heat flux of 0.1 W/cm^2 the positive slope portion of the temperature rise curve in figure 2 is where fluid conduction and Kapitza resistance are the most important factors in heat transfer (see Appendix A). It is apparent, however, that convection also plays an important role in the conduction and nucleate boiling region, since there is a significant temperature reduction (and heat transfer coefficient increase in figure 3) as flow increases. It is also apparent that the increasing turbulence due to increasing flow leads to nucleate boiling at lower surface temperature (superheat) and correspondingly higher heat transfer coefficients. Figures 4 and 5 show the temperature rise and heat transfer coefficients at 0.5 W/cm^2 (peak nucleate boiling heat flux in steady state occurs between 0.5 and 1 W/cm^2). The steady state nucleate boiling temperature and heat transfer coefficient are reached by 10^{-4} s, the conduction period (Appendix A) being nearly finished at the earliest measurement time of $2(10)^{-5}$ s. Flow has little effect at any time. Apparently the bubble nucleation effect is strong enough at this heat flux to overshadow the convection effect on heat transfer. The ineffectiveness of forced convection at the high nucleate boiling heat flux was noted previously by Giarratano [1] in steady state helium heat transfer experiments.

In figures 6 through 9 the heat flux is above the steady state nucleate boiling limit, but the final steady state film boiling temperatures and heat transfer coefficients are reached only after one second or more of heating. The level region extending from $2(10)^{-5}$ to $2(10)^{-4}$ s in figures 6 and 7 is the "transient nucleation" period in which nucleate boiling exists briefly. As in nucleate boiling of figures 4 and 5 forced flow has little effect in the transient nucleation period; however, the curves of constant flow rates begin to diverge and the difference becomes significant by 10^{-3} s. Finally, in steady state, the heat transfer coefficients at mass velocities of 235 to 238 $g/s{\cdot}cm^2$ are factors of four to five times that of a static fluid. The steady state variation of heat transfer coefficient at 5 W/cm^2 with Reynolds number is shown in figure 10 along with a correlation of Hendricks, et al. [2] developed from liquid hydrogen data.

Heat transfer coefficients predicted by the Hendricks correlation have been reduced by 20 percent since that reduction has been observed previously for rectangular channels heated on one side only [3]. This correlation is not entirely satisfactory since it predicts h = 0 at $N_{Re,L} = 0$, so that its applicability at best is limited to high flow rates. Other steady state correlations are being investigated. Further flow tests will be carried out at pressures up to 1 MPa.

3. PLANNED CONTINUATION OF TRANSIENT HEAT TRANSFER STUDIES

Thin carbon films vapor deposited on quartz substrates act as both heater and thermometer for all the experiments to date. These films have the advantage of extremely small mass relative to the surface area, hence, low heat capacity and fast time response, a thin cross section ($5(10)^{-5}$ cm) which makes thermal gradients through the film very small, and absence of any possible temperature difference between the heater and thermometer since one film performs both functions. The films are sensitive as thermometers and have proven to be stable in spite of severe thermal shocks.

While the carbon film heaters have many advantages, the results obtained in the first phases of this study indicate that some materials on the surfaces of superconductors may have heat transfer characteristics substantially different from the carbon film. Figure 11 shows that the kind of material in contact with the helium is particularly important in the early stages of transients when kapitza resistance is the predominant thermal resistance (see Section 3 in Appendix A). This figure compares measured heat transfer coefficients with those estimated for a hypothetical material with negligible Kapitza resistance. Since this resistance is believed to vary inversely as the third power of temperature, the two curves of figure 11 essentially merge after approximately 10^{-3} s when the temperature begins to rise rapidly. It should be emphasized that other surface material and surface condition effects exist even in steady state boiling and convection heat transfer. However, these other surface effects which have been observed in steady state heat transfer, are so unpredictable that no attempt has been made to illustrate them in figure 11. Data of reference 7 in Appendix A would place heat transfer coefficient curves for metals somewhere between the two curves of figure 11; however, the Kapitza resistance is not well enough established to allow a good approximation for a particular metal.

To establish helium transient heat transfer coefficients for metals and other surface materials of interest, experiments would be conducted in a manner similar to the present experiments, but with metallic and other surfaces used instead of carbon. The considerations for the design of test heaters are as follows:

1) Extremely thin vapor or vacuum deposited films do not necessarily have the same heat transfer characteristics as the bulk material even though the chemical composition may be the same. Therefore, the heat transfer surface should be of significant thickness capable of being prepared in the same way as bulk materials.

2) Metals or other materials of interest generally are not suitable as resistance thermometers, particularly near liquid helium temperature. For this reason it will be necessary to separate the thermometer from the heater. Carbon films seem to be the best choice as thermometers because of their fast response, and because their small cross sections relative to their surface area minimizes error due to heat conducted from the electrical leads, and maximizes the thermal coupling to the heater surface.

3) With the thermometer electrically insulated from the heat transfer surface it is not suitable to use the same film as thermometer and heater because a large heat flux through the electrical insulation would cause an unacceptable temperature difference across the insulation.

4) Because of their low electrical resistance, materials such as copper would not function well for resistance heating. This means that the heater must also be separate and electrically insulated from the heat transfer surface. Other materials with higher resistivity, such as stainless steel, could double as heater and heat transfer surface; however, the thickness would have to be very small to provide sufficient electrical resistance and prevent excessive temperature differences across the thickness.

The configuration of the proposed test heaters which takes these factors into consideration is shown in figure 12. The configuration shown is for a material of low electrical resistance and high thermal conductivity such as copper or aluminum. The carbon film below the surface would be divided into zones of which the larger area would be devoted to resistance heating and the smaller area would act as a thermometer. Lateral direct conduction of heat between the surface and the thermometer is small, but thermal contact between the surface and the thermometer is large. Since the heat flux to the thermometer is small, the measurement error should be small. The size of the space between heater zones which is devoted to thermometry should be made as small as possible to reduce the temperature non-uniformity at the surface at high heat flux. The surface material must be thick enough to distribute the heat uniformly, but not so thick as to produce large temperature differences across the thickness. For example, a copper surface material 0.05 cm thick has an approximate thermal time constant of

$$\tau = 5.2(10^{-7} \text{ s at 4 K, and}$$

$$\tau = 2.4(10)^{-6} \text{ s at 10 K.}$$

For a 0.1 cm thermometer gap in the heater, and a surface-to-helium heat transfer coefficient of 3 W/cm^2K an estimate of the maximum deviation in the surface temperature at the gap is,

$$\frac{T_{min} - T_b}{T - T_b} = 0.96.$$

The temperature difference due to conduction of heat through a 0.05 cm thickness of copper may be as high as 10 percent of the surface-to-helium temperature difference being measured. Though this is a significant difference it is possible to calculate the correction with reasonable accuracy.

Surface materials suggested for testing are copper which may be cleaned, oxidized, tinned, or coated with a typical insulating material such as enamel. Other materials may be tested as consistent with time and funding.

Future tests planned include the following:

a. Forced flow near critical and supercritical pressures with the present carbon film heaters.
b. Use of metal and other heat transfer surfaces.
c. Transient tests in which the heater power is controlled in such a way as to produce a uniform rate of temperature rise or fall (rather than constant heat flux).
d. Further tests in which temperature recovery following a pulse is measured.
e. Shorter power supply rise time so that steady power is reached and constant heat flux measurements may be started at earlier times ($\sim 1 \mu s$). This is of particular interest in the design of fast response superconducting switches, and for better understanding of the conduction heat transfer period and Kapitza resistance.

4. NOMENCLATURE

c_p	specific heat at constant pressure, j/(g.K)
G	mass flow rate per unit cross sectional area (or mass velocity), $g/(s \cdot cm^2)$
h	channel height, cm, heat transfer coefficient $W/(cm^2 \cdot K)$
k	thermal conductivity, $W/(cm \cdot K)$
$N_{Re,L}$	Reynolds number based on liquid properties - Gh/μ
$\overline{T}$	surface temperature if heating were uniform, K
T_b	bulk liquid temperature, K
T_{min}	temperature at greatest deviation from $\overline{T}$, K
μ	dynamic viscosity, $g/cm \cdot s)$
ρ	density, g/cm^3
τ	thermal time constant, s

5. REFERENCES

1. Giarratano, P. J., Hess, R. C. and Jones, M. C., Forced convection heat transfer to subcritical helium I, Advances in Cryogenic Engineering, Vol. 19, Plenum Press, New York, (1974).

2. Hendricks, R. C., Graham, R. W., Hsu, Y. Y., and Friedman, Experimental heat transfer and pressure drop of liquid hydrogen flowing through a heated tube, NASA TND-765 (1961).

3. Rohsenow, W. M., and Hartnett, J. P. (ed.) Handbook of Heat Transfer, McGraw Hill, Inc. (1973), Ch. 7.

Figure Captions

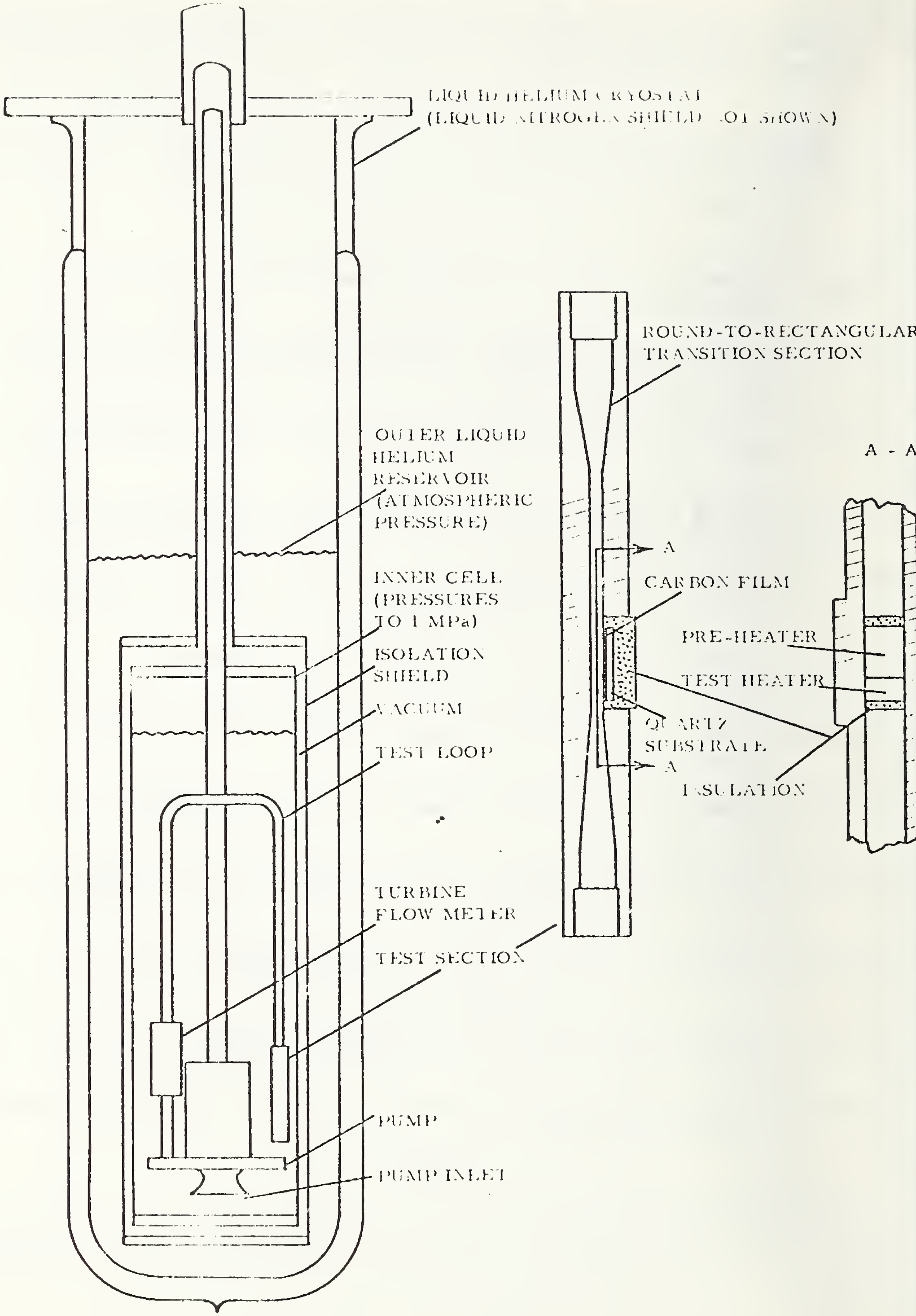

Figure 1. Schematic of the forced convection transient helium heat transfer apparatus.

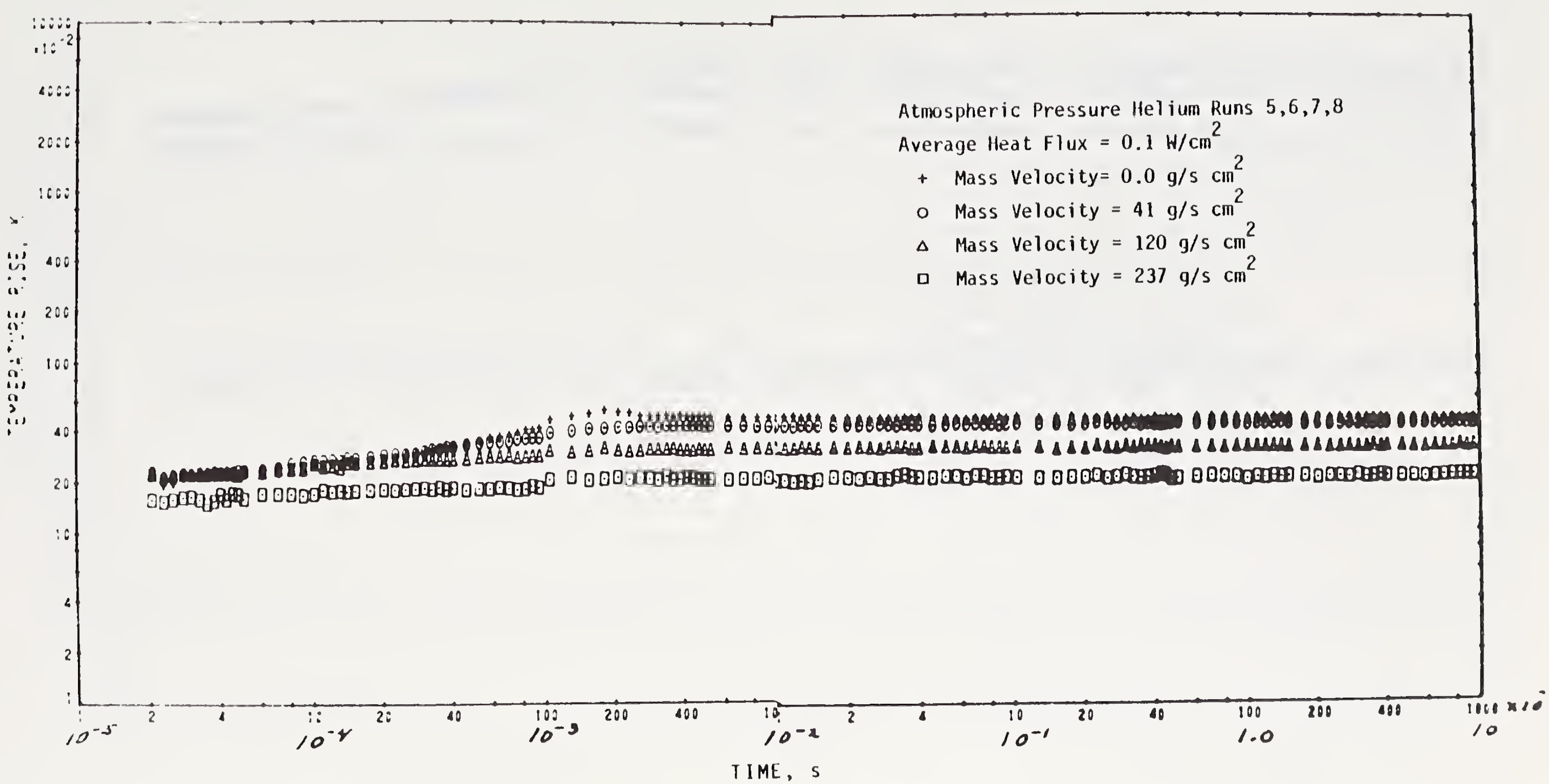

Figure 2. Effect of helium flow on heater temperature transient rise for a heat flux of 0.1 W/cm^2.

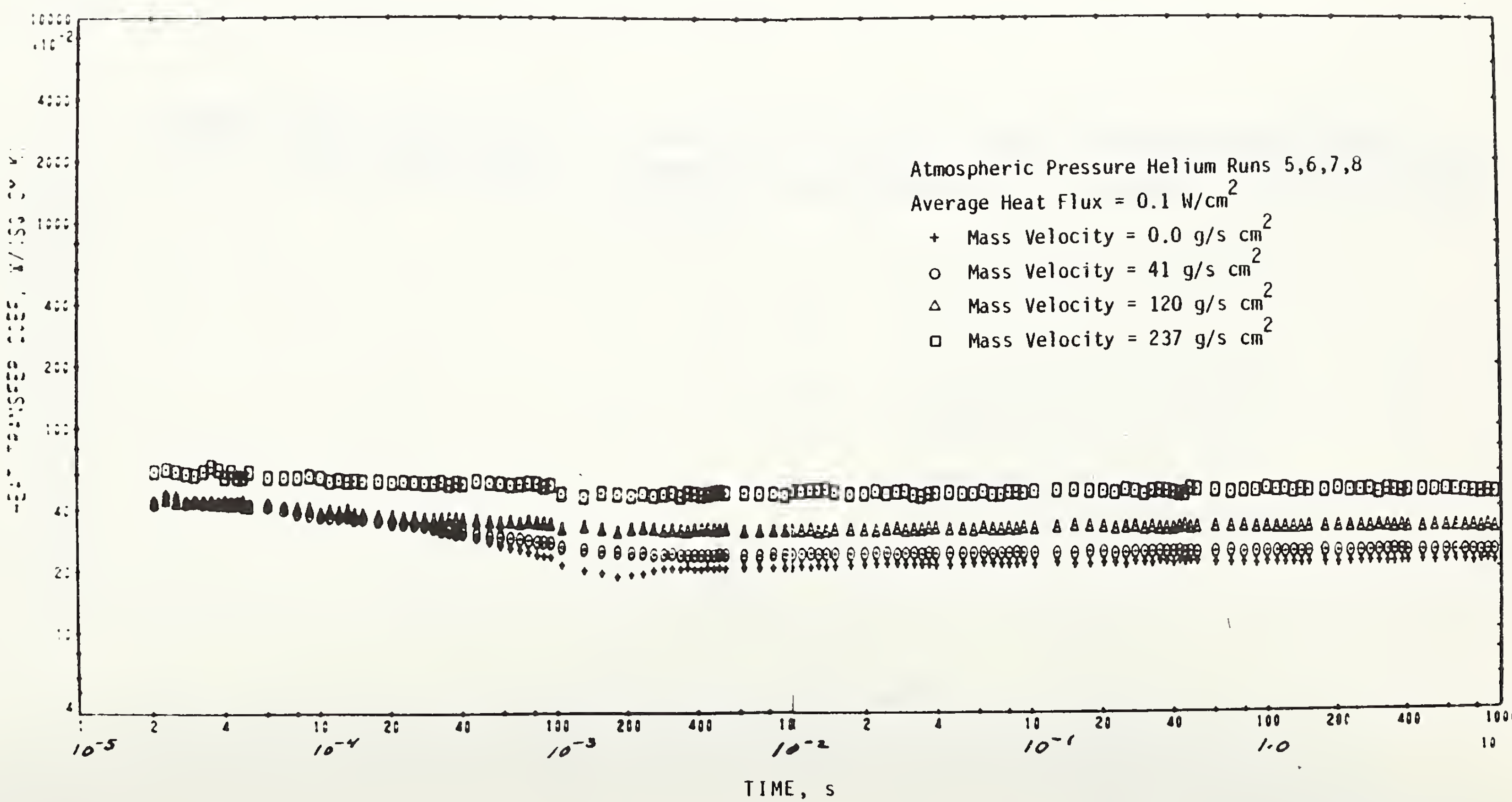

Figure 3. Effect of helium flow on transient heat transfer coefficient for a heat flux of 0.1 W/cm^2.

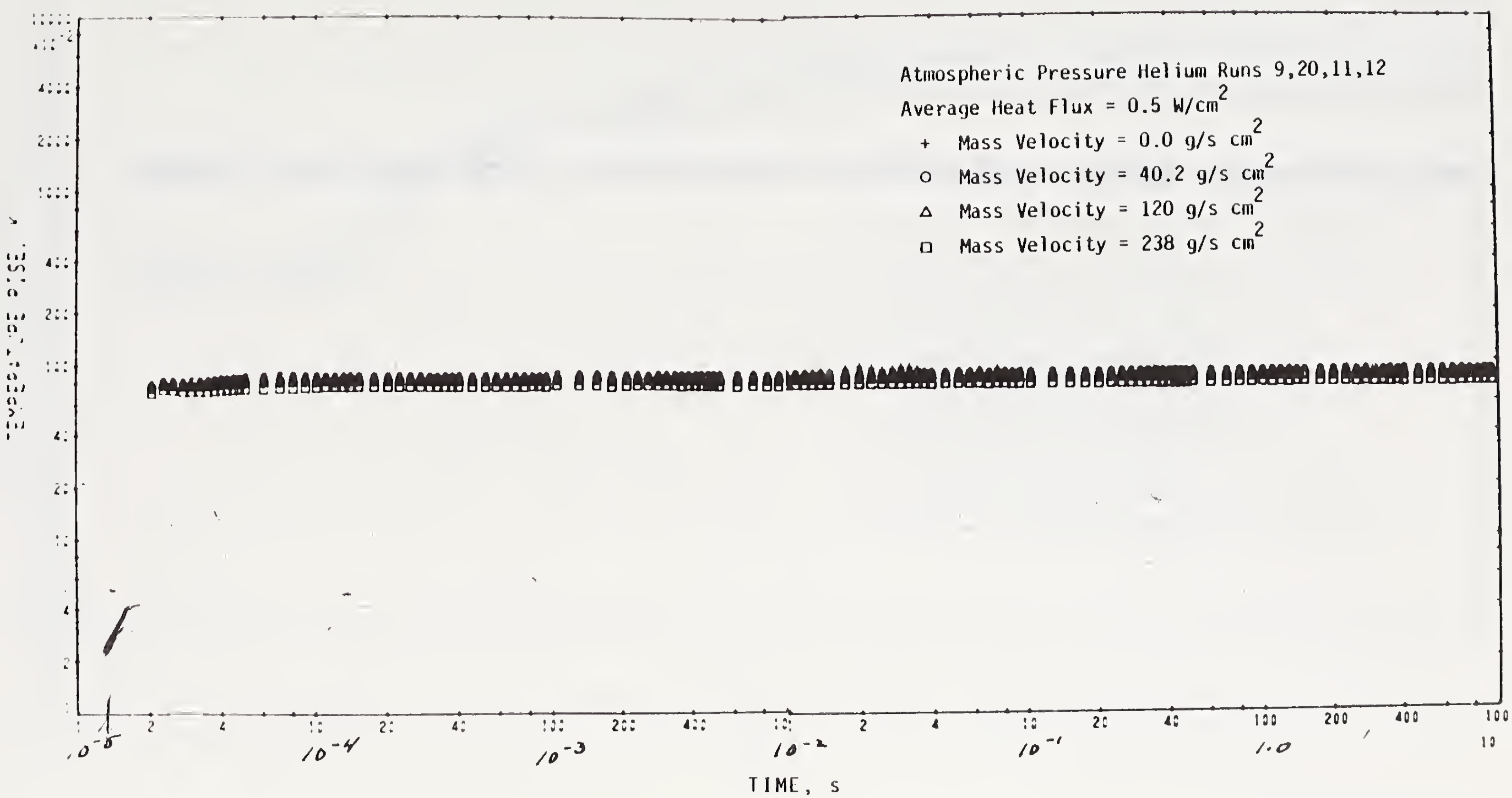

Figure 4. Effect of helium flow on heater transient temperature rise for a heat flux of 0.5 W/cm^2.

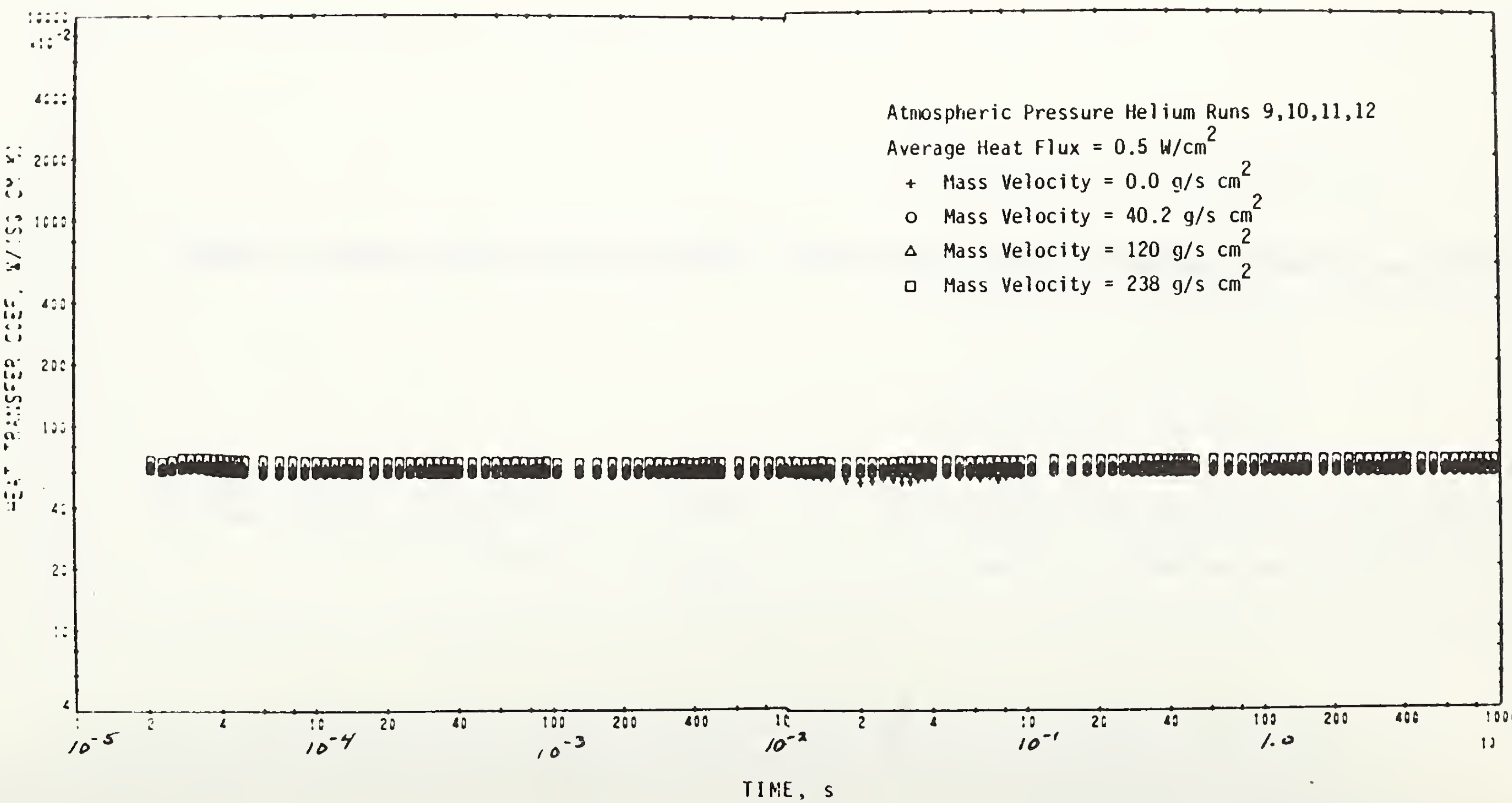

Figure 5. Effect of helium flow on transient heat transfer coefficient for a heat flux of 0.5 W/cm^2.

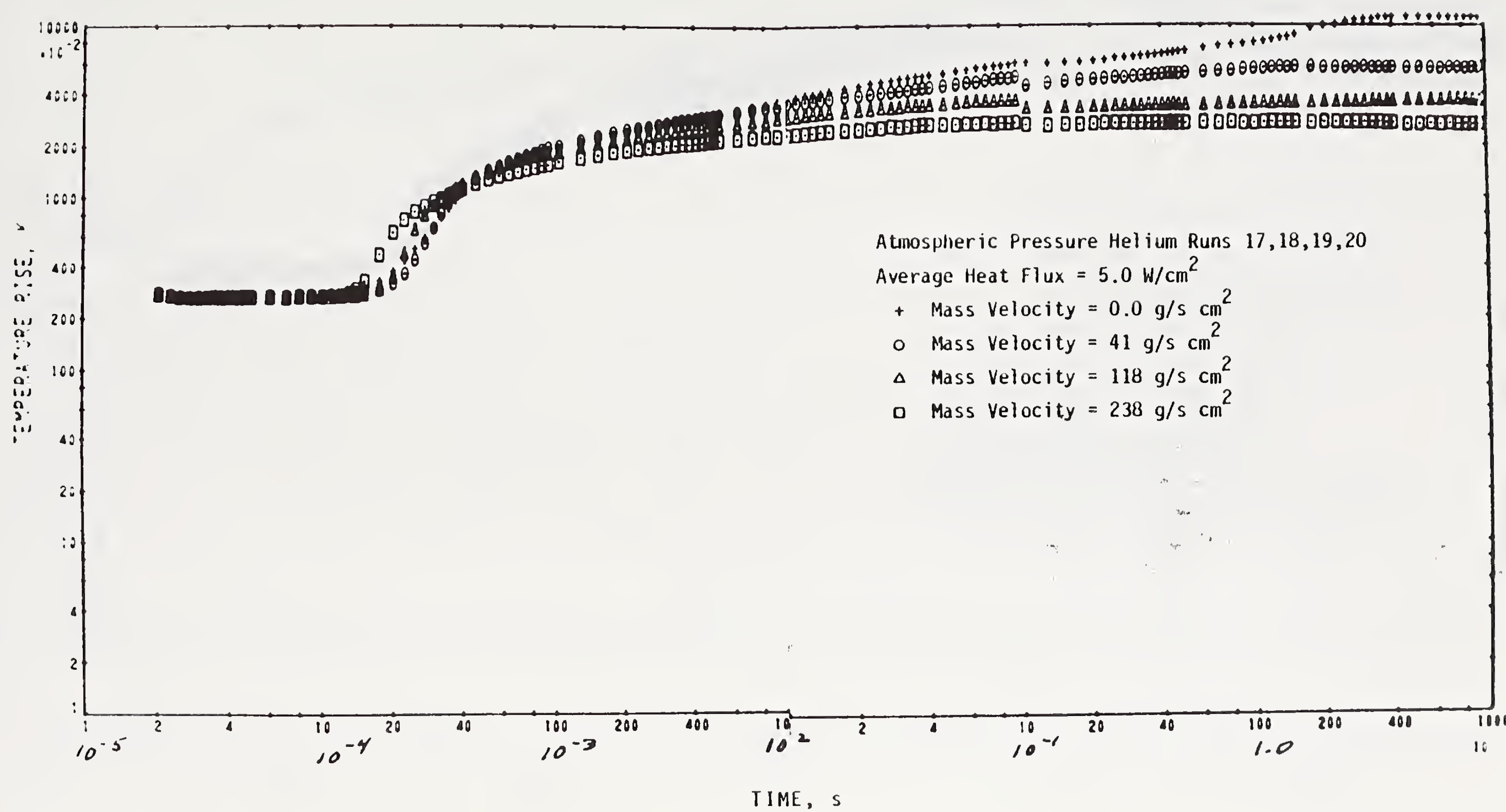

Figure 6. Effect of helium flow on heater transient temperature rise for a heat flux of 5 W/cm^2.

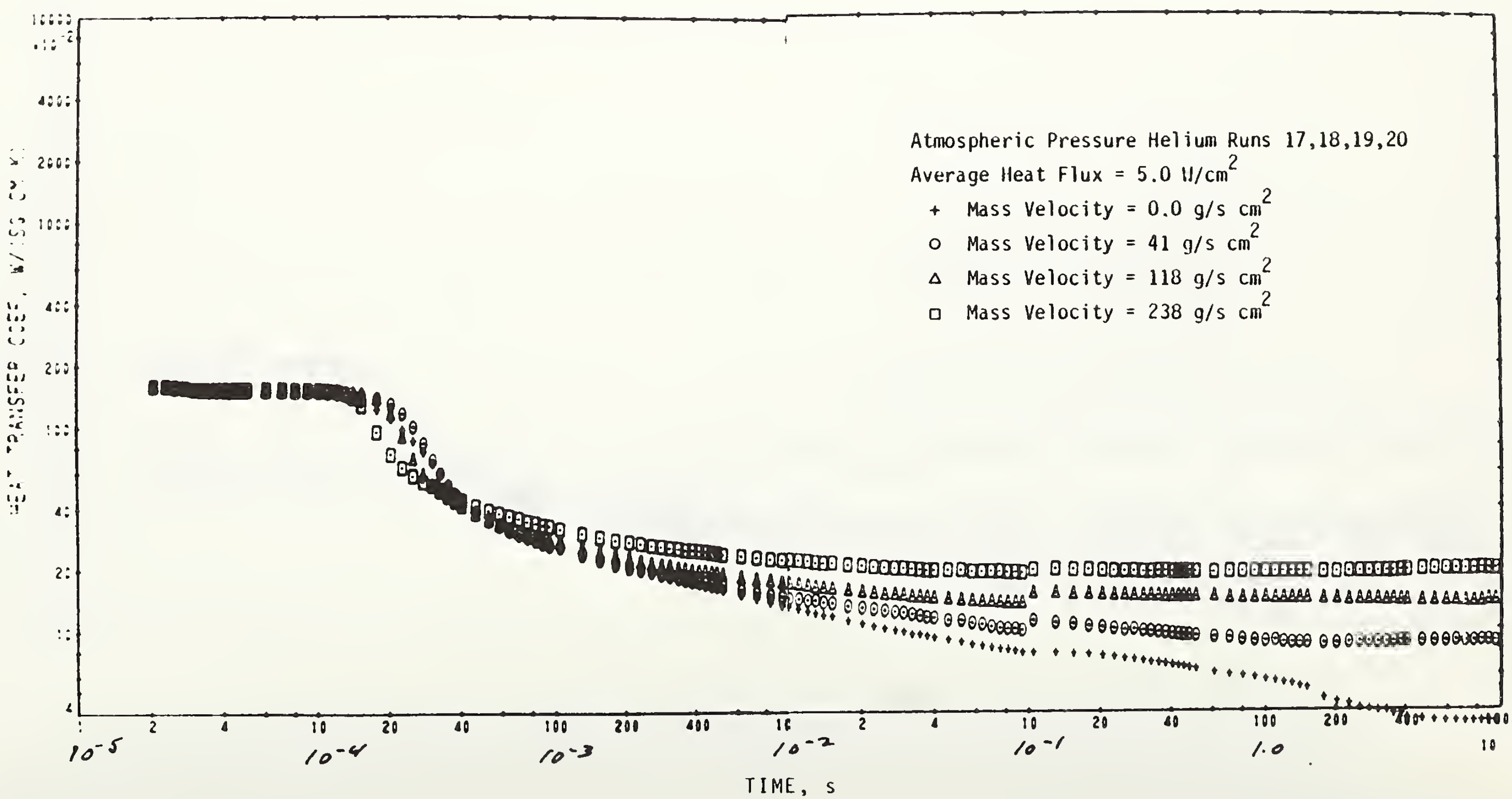

Figure 7. Effect of helium flow on transient heat transfer coefficient for a heat flux of 5 W/cm^2.

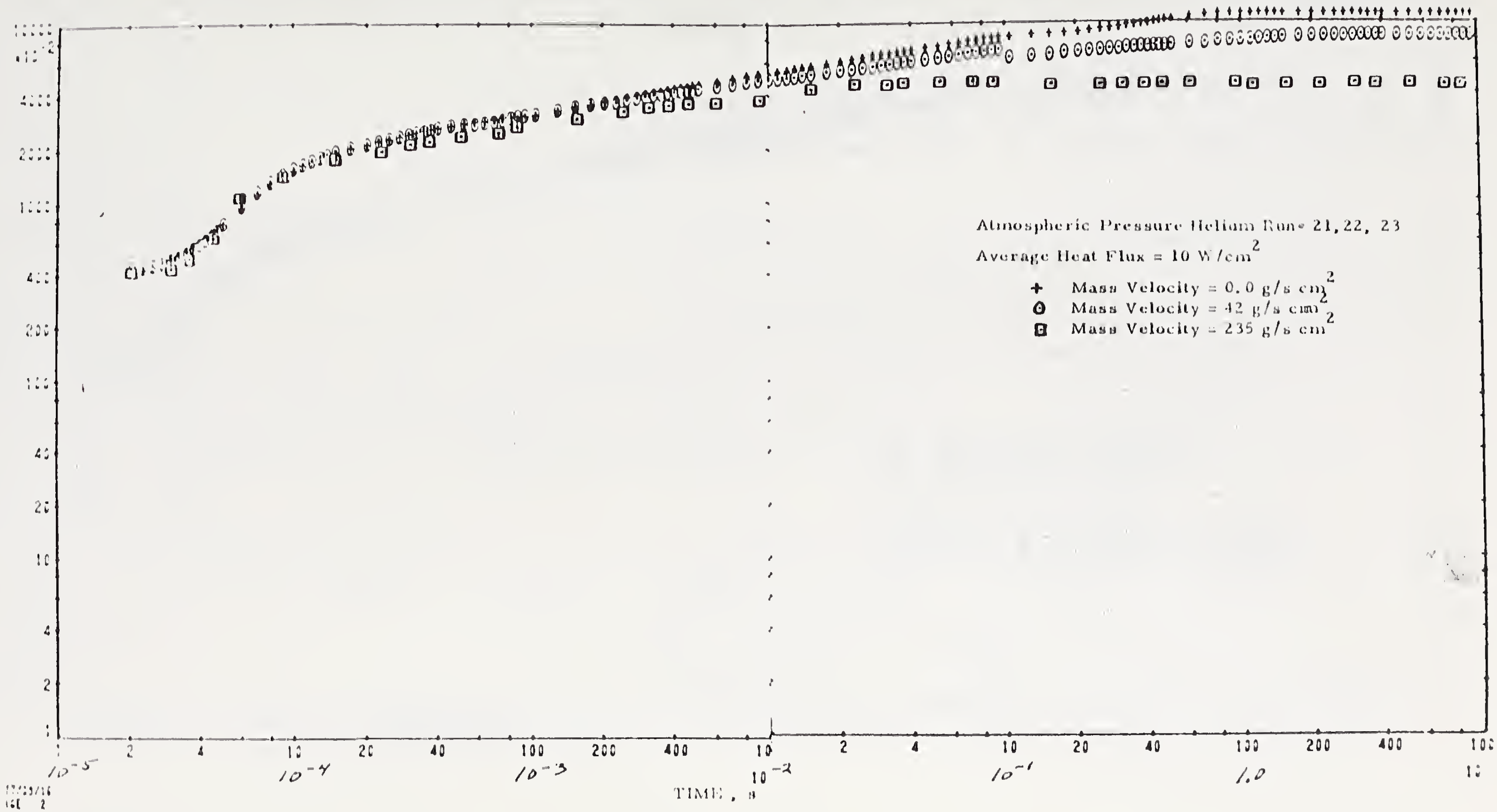

Figure 8. Effect of helium flow on heater transient temperature rise for a heat flux of 10 W/cm^2.

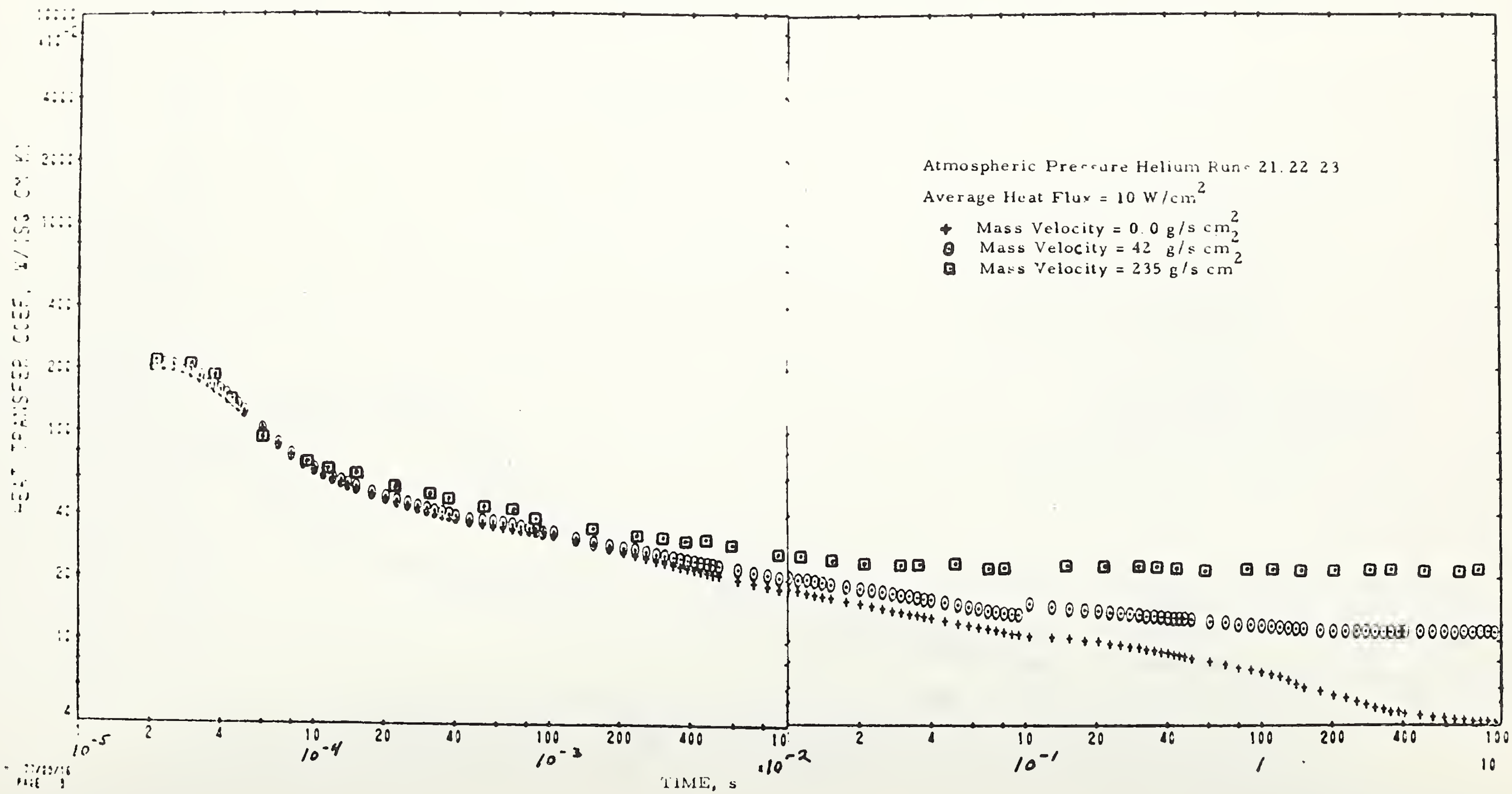

Figure 9. Effect of helium flow on transient heat transfer coefficient for a heat flux of 10 W/cm^2.

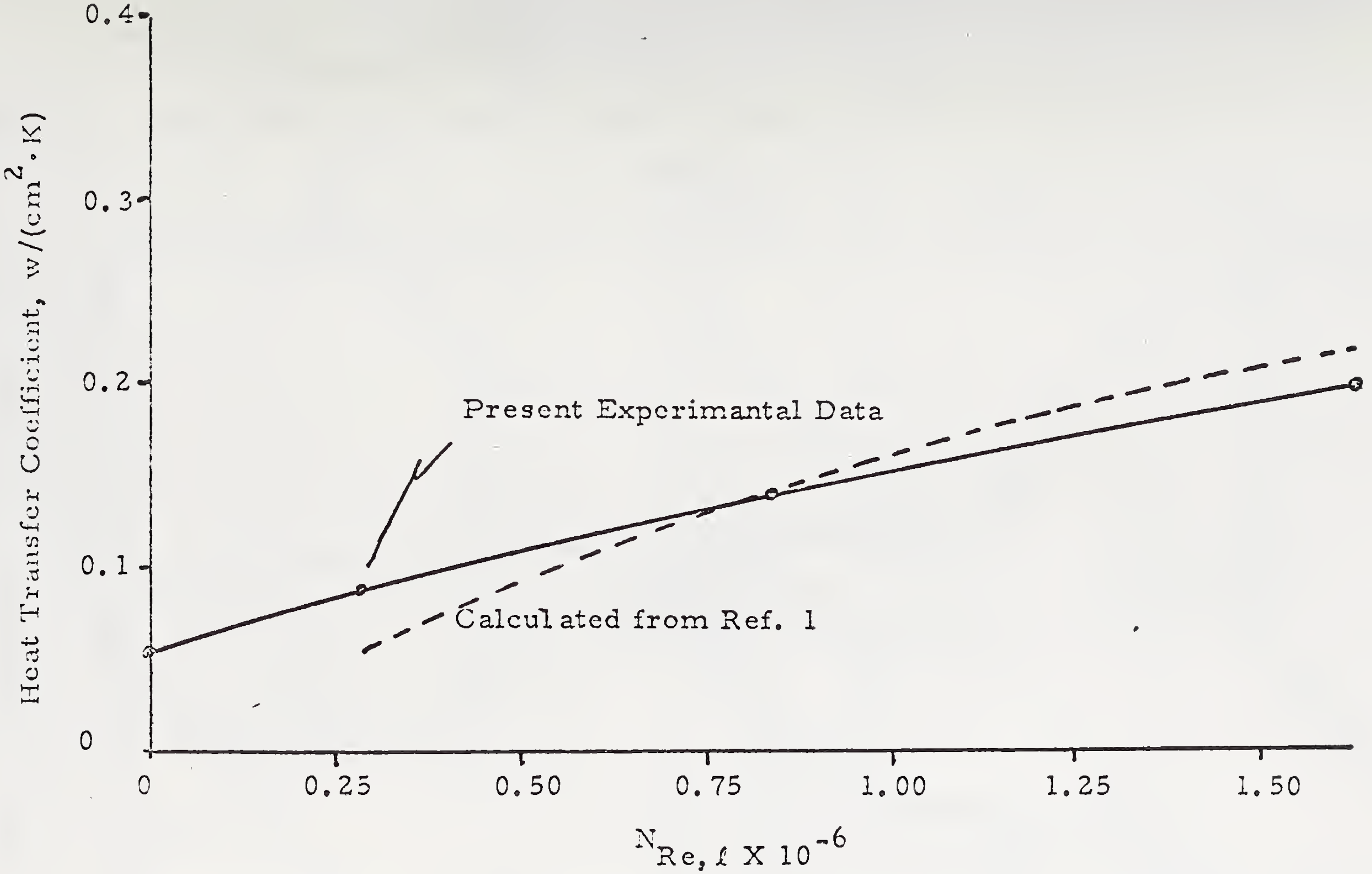

Figure 10. Effect of Reynolds number on steady state heat transfer to liquid helium at 4K and a heat flux of 5 w/cm^2.

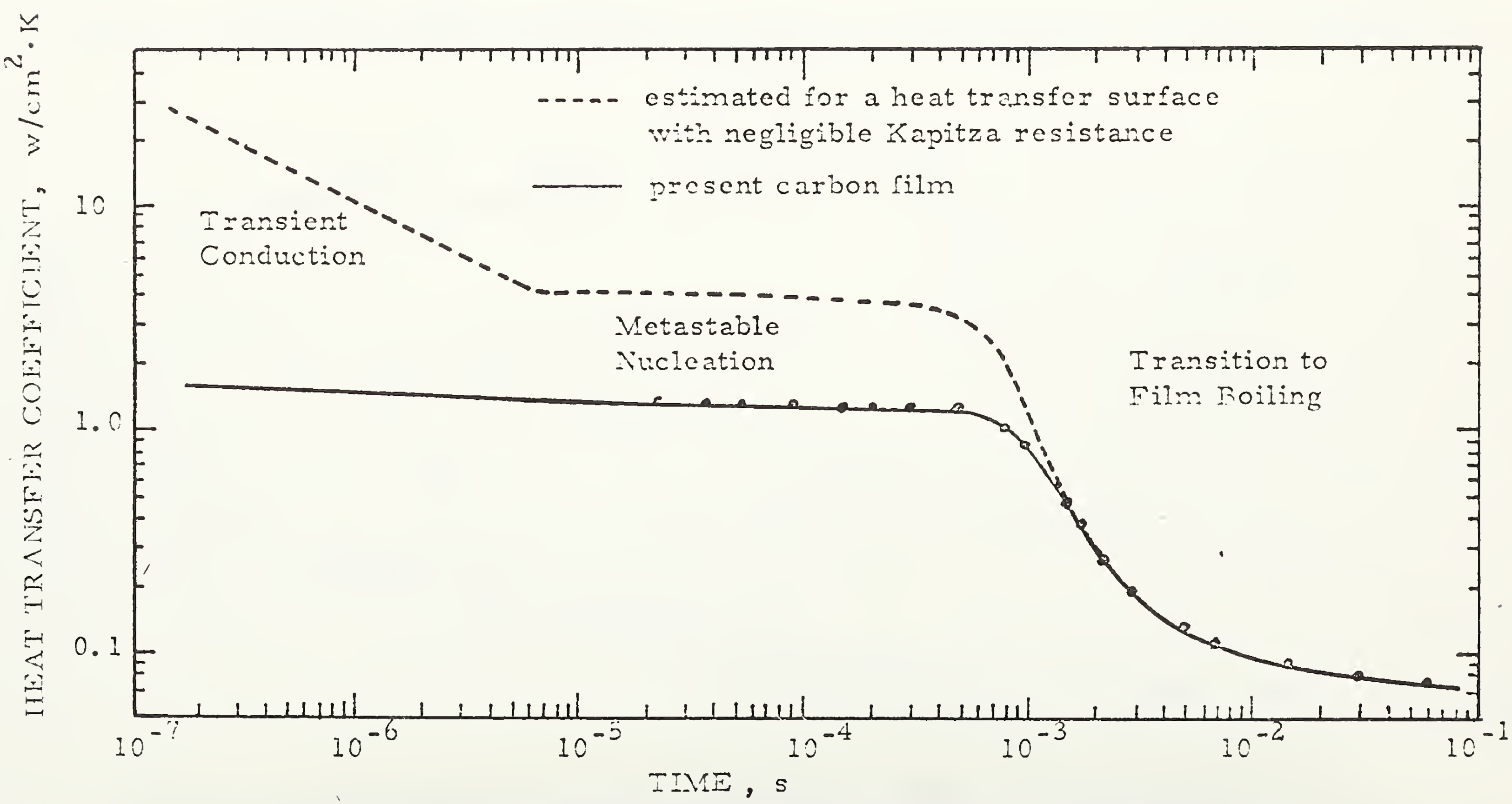

Figure 11. The effect of heat transfer surface Kapitza resistance on transient helium heat transfer at a heat flux of 2 w/cm^2.

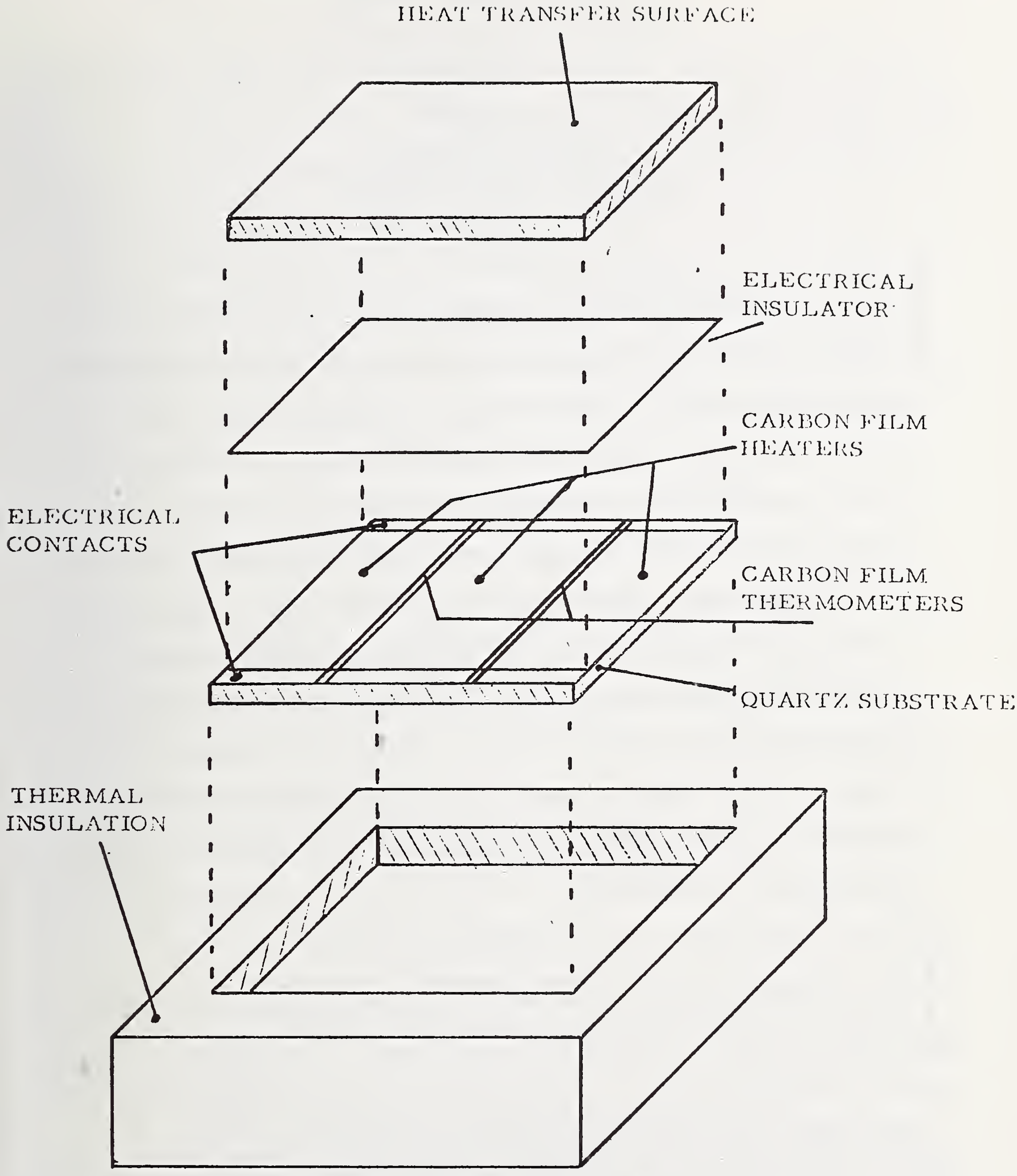

Figure 12. Experimental arrangement for transient heat transfer tests on surfaces other than carbon film.

(preprint of paper to appear in the International Journal of Heat and Mass Transfer)

TRANSIENT HELIUM HEAT TRANSFER
PHASE I - STATIC COOLANT*

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Transient heat transfer data have been obtained for flat heating surfaces in static liquid and supercritical helium. Measurements start $2(10)^{-5}$ s after step power inputs, and cover a heat flux range of 0.05 to 20 W/cm^2, pressures from 0.09 to 0.3 MPa, and four different heater orientations. Initial heat transfer coefficients, being limited primarily by the Kapitza resistance, are 10 to 100 times greater than steady state, and the time to reach steady state varies from 10^{-5} s to 1 s. For heat flux below the steady state peak nucleate boiling limit the temperature follows calculations based on pure conduction to the steady state nucleate boiling level. Above that limit the transient conduction period leads to an apparent metastable nucleation period followed by a transition to film boiling.

1. INTRODUCTION

Many superconducting devices employ large power pulses or are subject to power transients due to overloads or other causes. It is usually desirable, and often imperative, that the superconductor not quench during the pulse. In other cases one is concerned about the temperature recovery following heat release from flux jumps or micromechanical movements. Fortunately, heat transfer from the superconductor surface to the helium coolant can be many times

* This work was carried out at the NBS Cryogenics Division, Boulder, CO under the sponsorship of the Air Force Office of Scientific Research, Washington, DC. Continuation of the work is sponsored by ERDA-DMFE.

higher during a short pulse than in steady state for a given surface temperature rise. Thus, even though rapid pulses or transients cause relatively high losses, the ability of the helium coolant to dissipate pulsed heat rapidly is a compensating factor.

At the beginning of a heating transient, as the power rises rapidly, the temperature gradient in the fluid adjacent to the solid surface may be extremely steep since a finite time is required for an equilibrium temperature field with less steep gradients to be established. During this period the thermal resistance is so low that Kapitza solid-liquid interface resistance, which is usually ignored at temperatures above the lambda point, may be the dominant resistance. Since nucleation, growth of bubbles, and formation of vapor films also require finite time, liquid exists at least briefly in non-equilibrium states above the saturation temperatures and boiling can be delayed significantly. As the thermal boundary layer thickens through prolonged application of power or repetition of pulses, convection effects become important and eventually steady state convection or boiling heat transfer are established. In a chain of pulses the surface temperature rises while the power is being applied and decays asymptotically toward the starting value while the power is off; therefore, the entire level of temperature rises with each pulse. The temperature ultimately reached after a large number of pulses is related to the average heat flux for the cycle and the steady state heat transfer coefficient.

It is apparent that the use of steady state heat transfer rates when designing for heating transients in superconducting devices is overly conservative. Yet, for lack of more appropriate data, steady state boiling helium heat transfer data have been used in designs involving transient processes. An exception to the use of steady state data was the pulsed high energy inductive storage system designed by AERL[1] for the Air Force Aero Propulsion Laboratory. Heat flux for this design was estimated from the transient experimental data of Jackson.[2] This data is not sufficiently detailed; howeve for general application. More recently, experimental results have been report by two Japanese authors[3] at Yokohama National University, for the pulse heating of a thin lead film in a liquid helium bath. This study, however, gives only one point at the superconducting transition temperature of the lead film at a given power level, whereas, the entire temperature development is desired. The temperature rise after the onset of nucleate boiling until the establishment of steady temperatures has been experimentally determined by R. L. Bailey[4] at the Rutherford High Energy Laboratory, but the time scale was on the order of seconds and tenths of seconds and power densities were less than 25 mW/cm^2. In the wide range of existing and contemplated superconducting applications, pulses as brief as 0.05 ms and repetition rates as high as 200 per second are of concern.[5] In most applications, for economy of weight, material and liquid helium, the power levels will be pushed to the maximum safe limits as dictated by the ability of the coolant to dissipate heat due to transients, and these maximum heat fluxes may exceed 10 W/cm^2 for short pulses.

Therefore, it is clear that designers of superconducting equipment need a more complete picture of transient helium heat transfer phenomena than was available before the commencement of this study. This total program will provide experimental data and verified predictive methods covering heat fluxes up to 25 W/cm^2, measurement times starting at less than 0.05 ms extending to steady state, varying orientation of the heated surfaces, varying helium pressures and temperatures, and both static and forced flow of coolant. This paper covers the first phase of the program, namely the experimental results for a static coolant.

1.1. Acknowledgement

The author wishes to acknowledge the contributions of Dr. M. C. Jones who recognized the need for helium transient heat transfer data and supplied many of the ideas for measurement methods. Thanks are due also to Dr. Vincent Arp for many helpful suggestions and preliminary studies.

2. IDEALIZED TRANSIENT CONDUCTION HEAT TRANSFER

At the beginning of a heating transient, before convection currents or boiling have been established, conduction is the principle mechanism of heat transfer. The following set of assumptions and boundary conditions for solution of the heat conduction equation apply to the configuration of the test apparatus.

1) The heat source is an infinite plate in the X-Y plane at $Z = 0$. (This is equivalent to assuming that the thermal boundary layer is very thin relative to the width of the plate.)

2) Heating occurs only on the surface at the liquid solid interface.

3) Stagnant liquid helium is present at $Z > 0$.

4) A solid substrate exists at $Z < 0$.

5) Time during which heat is applied is short enough that convection currents are not established. This means there is not time for bubble formation and temperatures may exceed the equilibrium saturation temperature.

6) Radiation heat transfer is negligible.

7) Fluid and solid properties are constant.

8) Temperature difference due to Kapitza resistance may be considered separately.

The Fourier heat conduction equation in one dimension of the solid substrate is,[6]

$$\frac{\partial^2 \Delta T_s}{\partial Z^2} = \frac{1}{\alpha_s} \frac{\partial \Delta T_s}{\partial \theta} \qquad , \qquad Z \leq 0; \tag{1}$$

and for the fluid

$$\frac{\partial^2 \Delta T_f}{\partial Z^2} = \frac{1}{\alpha_f} \frac{\partial \Delta T_f}{\partial \theta} \qquad , \qquad Z > 0. \tag{2}$$

Boundary conditions for the substrate for a square wave heat pulse are,

$$\Delta T_s \ (Z,0) = 0 \tag{3}$$

$$\lim_{Z \to -\infty} \Delta T_s \ (Z,\theta) = 0 \tag{4}$$

$$P_s = K_s \left(\frac{\partial \Delta T_s}{\partial Z} \right)_{0,\theta} \qquad , \qquad 0 < \theta \leq \theta_P \tag{5}$$

$$P_s = 0 \qquad , \qquad \theta > \theta_P \ .$$

P_s is the part of the heater power being dissipated in the substrate. Boundary conditions for the fluid are,

$$\Delta T_f\ (Z,0) = 0 \qquad (6)$$

$$\lim_{Z \to \infty} \Delta T_f\ (Z,\theta) = 0 \qquad (7)$$

$$P_f = -K_f \left(\frac{\partial \Delta T}{\partial Z} \right)_{0,\theta} , \qquad 0 < \theta < \theta_P \qquad (8)$$

$$P_f = 0 \qquad , \qquad \theta > \theta_P \quad .$$

P_f is the part of the heater power being dissipated in the fluid. Also, the total heater power is

$$P = P_f + P_s \qquad (9)$$

where

$$P = \text{const} \quad 0 < \theta < \theta_P$$

$$P = 0 \qquad \theta \geq \theta_P, \quad \text{and}$$

$$\Delta T_s\ (0,\theta) = \Delta T_f\ (0,\theta) \ . \qquad (10)$$

The solution for the fluid is,[6]

$$\Delta T_f = \frac{P_f}{K} \left[2\sqrt{\frac{\alpha_f \theta}{\pi}}\ e^{\frac{-Z^2}{4\alpha_f \theta}} - Z\ \mathrm{erfc}\left(\frac{Z}{2\sqrt{\alpha_f \theta}} \right) \right] \qquad \text{for } 0 < \theta < \theta_P \qquad (11)$$

and

$$\Delta T_f = \frac{P_f}{K_f} \left[2\sqrt{\frac{\alpha_f \theta}{\pi}}\ e^{\frac{-Z^2}{4\alpha_f \theta}} - Z\ \mathrm{erfc}\left(\frac{Z}{2\sqrt{\alpha_f \theta}} \right) \right]$$

$$- \frac{P_f}{K_f} \left[2\sqrt{\frac{\alpha_f (\theta - \theta_P)}{\pi}}\ e^{\frac{-Z^2}{4\alpha_f (\theta-\theta_P)}} - Z\ \mathrm{erfc}\left(\frac{Z}{2\sqrt{\alpha_f (\theta - \theta_P)}} \right) \right] \text{for } \theta > \theta_P . \qquad (12)$$

The equations for the substrate temperature field are exactly analogous to (11) and (12) except that a subscript "s" appears instead of "f".

At the surface (Z = 0) we have

$$\Delta T_f(0,\theta) = \frac{2P}{\sqrt{\pi}} \frac{\sqrt{\theta}}{\left(\frac{K_s}{\sqrt{\alpha_s}} + \frac{K_f}{\sqrt{\alpha_f}}\right)} , \quad 0 < \theta < \theta_P \tag{13}$$

and

$$\Delta T_f(0,\theta) = \frac{2P}{\sqrt{\pi}} \frac{(\sqrt{\theta} - \sqrt{\theta-\theta_P})}{\left(\frac{K_s}{\sqrt{\alpha_s}} + \frac{K_f}{\sqrt{\alpha_f}}\right)} , \quad \theta > \theta_P \; . \tag{14}$$

Equation (13) gives the surface temperature rise during the heat pulse and equation (14) gives the temperature decay following the pulse.

Obviously the assumptions drastically limit the time during which equations (13) and (14) are valid. Comparison with experimental data help determine the conduction period duration.

3. KAPITZA RESISTANCE

Whereas fluid thermal resistance builds up as the boundary layer develops, Kapitza resistance apparently occurs the instant heat flow begins. Consequently, Kapitza resistance initially dominates the heat flow process at liquid helium temperatures, though only briefly. How long Kapitza resistance dominates depends upon its magnitude in relation to the increasing conduction resistance, and this in turn depends on the kind of material and physical condition of the surface as well as the temperature.

Since the exact nature of a surface is difficult to characterize[7] and may change with time, the influence of material and surface condition is difficult to predict. However, for a given heater the Kapitza conductance h_K (reciprocal of resistance) has been observed to obey approximately a T^3 relationship:[7]

$$h_K = \frac{P_f}{\Delta T_K} = h_{K,ref} \left(\frac{T}{T_{ref}}\right)^3 , \tag{15}$$

where $h_{K,ref}$ is a constant Kapitza conductance at reference temperature T_{ref}, P_f is the power being dissipated into the helium bath.

The heater surface temperature is

$$T = T_{bath} + \Delta T_f + \Delta T_K .$$

Therefore,

$$P = \frac{h_{K,ref}}{T_{ref}^{\ 3}} \left(T_{bath} + \Delta T_f + \Delta T_K\right)^3 \Delta T_K . \tag{16}$$

For low power levels ΔT_K can be estimated from the experimental data as the asymptote of the temperature curve as $\theta \to 0$. In this way it was determined that

$$h_K = 0.7 \ \mathrm{W/cm^2K} \quad \text{at approximately 4 K.}$$

Also,

$$\frac{h_{K,ref}}{T_{ref}^{\ 3}} = 0.011 \ \frac{\mathrm{W}}{\mathrm{cm^2K^4}} . \tag{17}$$

If T_{ref} - 1.9 K, $h_{K,ref}$ = 0.075 which is well within the possible range of values of h_K at 1.9 K shown by Snyder[7].

4. EXPERIMENTAL PROGRAM

The apparatus used is shown schematically in figure 1. A carbon thin film resistor[8] whose resistance is a known function of temperature serves both as an extremely fast response heater and thermometer. Its dimensions are approximately 5 mm parallel to current by 10 mm wide. The carbon film is deposited on a quartz substrate which has an extremely low heat capacity at liquid helium temperature so that most of the heat flows into the fluid and very little flows into the substrate. Voltage across the film and voltage across a standard resistor are recorded at intervals as close as 0.5 microsecond by a digital recording oscilloscope; 2048 points are recorded on each sweep of the oscilloscope on each of two channels. Power may be supplied as a step function or as pulses of any desired width or frequency. A minicomputer data acquisition system has been added to facilitate data processing and to permanently record the data. The heater surface temperature, power level and heat transfer coefficients are printed as a function of time.

For these tests the heater was submerged in a static helium bath. Insofar as possible power was applied as a step input or square wave pulse and ramp times of less than $1.5\ (10)^{-5}$ s were achieved. However, at high power levels resistance change due to large temperature rise causes power variation of up to $\pm$ 10 percent from the average. Tests covered the following parameter variations:

1. Power densities ranging from 0.05 to 20 W/cm^2 repeated for each variation of the other parameters.

2. Four orientations of the heater -- vertical, horizontal facing up, horizontal facing down, and 45° down.

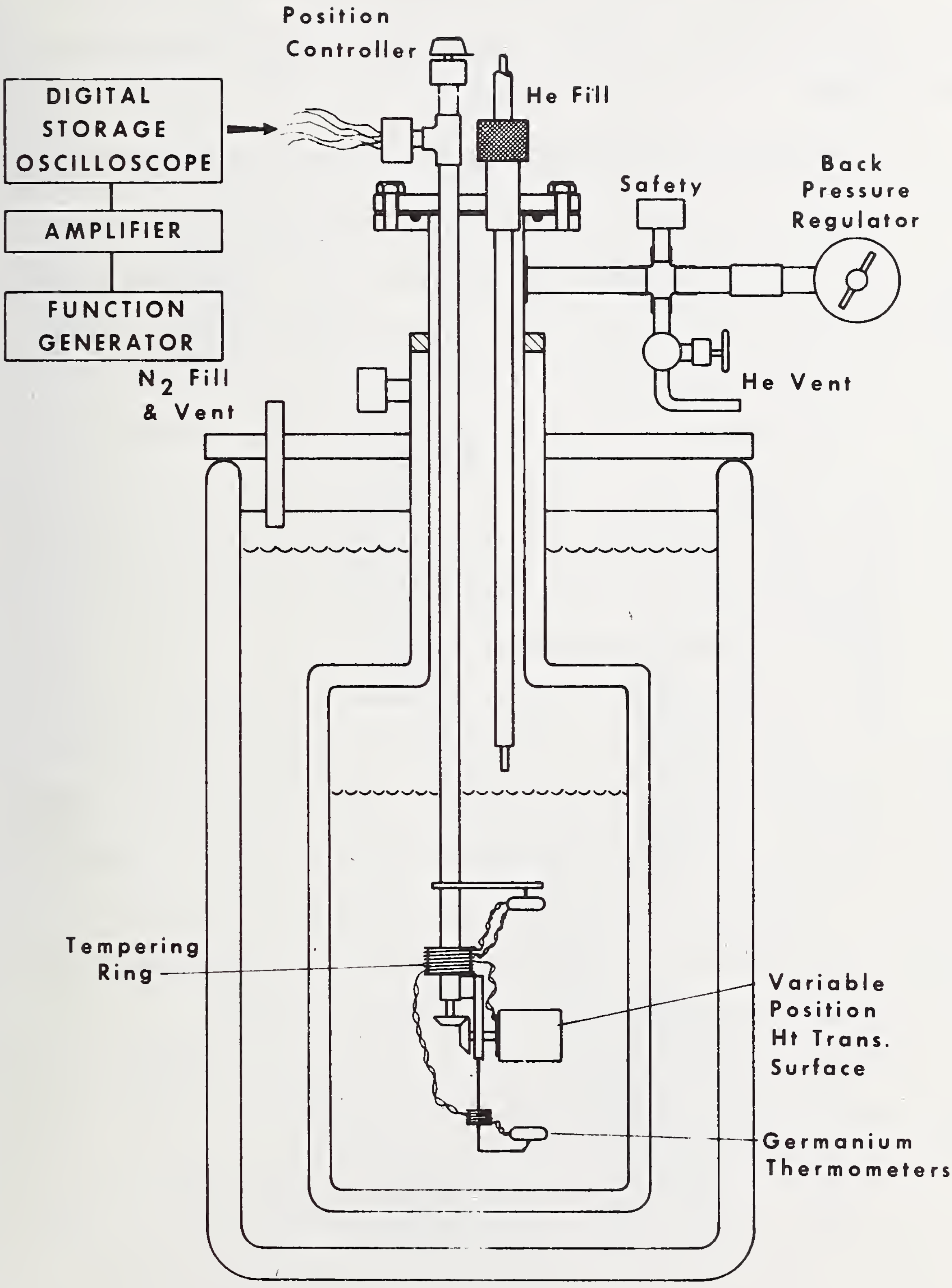

Figure 1. Pool pulsed heat transfer experimental apparatus.

3. Variations in bath pressure and temperature.

4. Repetition of square wave pulses with variations of pulse width and time between pulses. Finally, temperature decays following single pulses were recorded.

4.1. Preliminary Liquid Nitrogen Transient Heat Transfer Experiments

We performed tests at four power levels with liquid nitrogen for a check on measurement techniques and predictions for a fluid which is markedly different from helium in heat transfer related properties, and for which Kapitza resistance is negligible. Figure 2 shows the temperature rise of a vertical heater surface in liquid nitrogen as a function of time from the start of a step power input. The heat flux per unit area was nearly constant during the measurements and the four tests were at 5, 10, 15 and 18 W/cm^2. The temperature rise in the first $3(10)^{-2}$ s closely follows the non-boiling transient heat transfer calculations.

For 5, 10 and 15 W/cm^2 the heater temperature continued to rise in proportion to $(time)^{1/2}$ and for 9 to 15 K above the steady state nucleate boiling level, then dropped back down to that level. The time required to finally reach steady state nucleate boiling was approximately 0.2 s. A power density of 18 W/cm^2 exceeded the maximum nucleate boiling heat flux and the heater began a slow transition to a film boiling temperature which might have burned out the heater if the heating had been allowed to continue.

The heater temperature rise during the non-boiling transient heat transfer period was calculted from equation (13) in which

$$\frac{K_s}{\sqrt{\alpha_s}} + \frac{K_f}{\sqrt{\alpha_f}} = 0.0906 \frac{Ws^{1/2}}{cm^2 K} \text{ for } LN_2$$

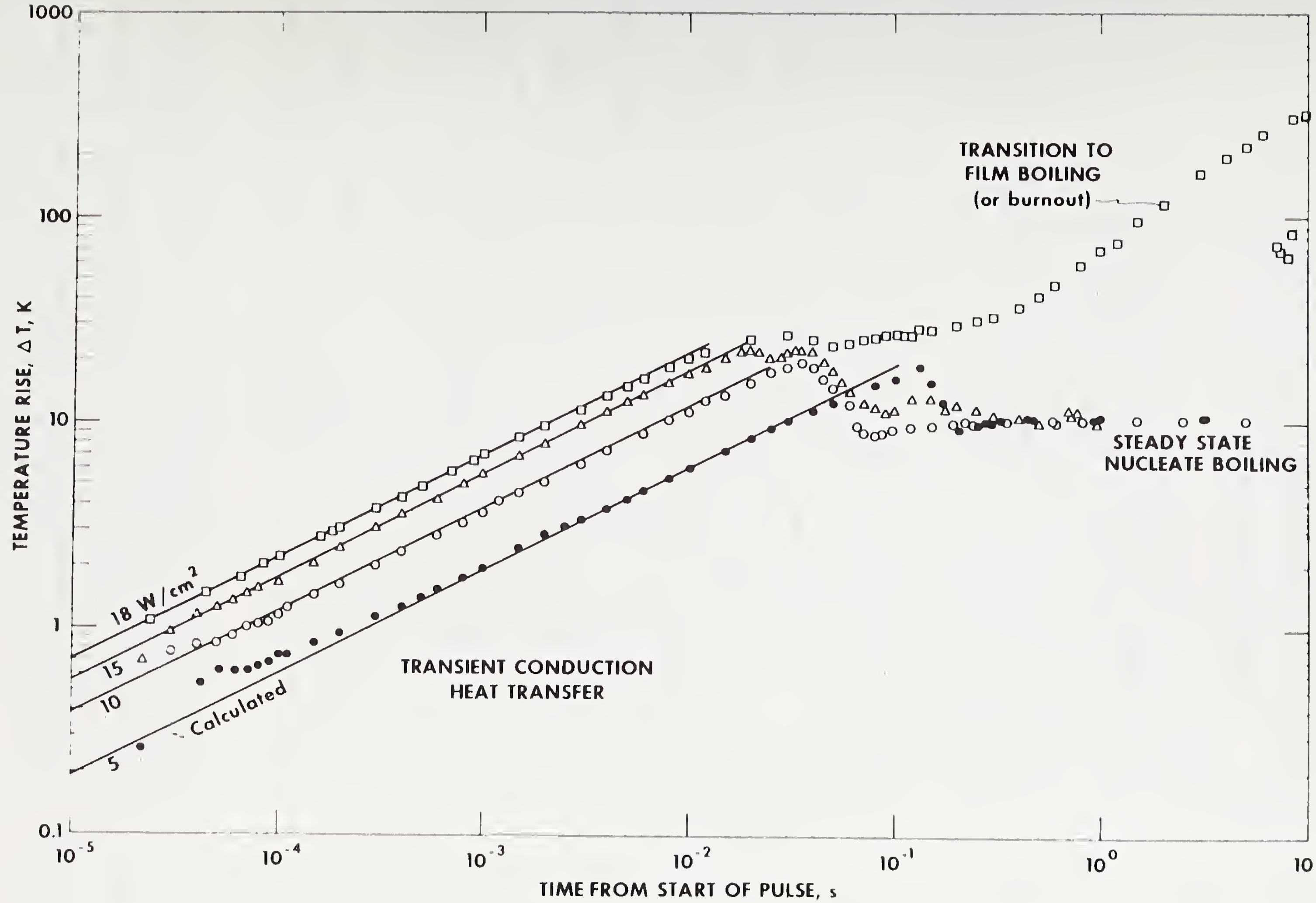

Figure 2. Transient heat transfer from a vertical heater to liquid nitrogen.

and quartz substrate. At liquid nitrogen temperature Kapitza resistance is negligible; therefore $\Delta T = \Delta T_c$.

4.2. Liquid Helium Transient Heat Transfer Experiments with a Vertical Heater

4.2.1. Accuracy

The carbon thermometer resistance at helium temperature increased by 1.3 percent over the duration of the test program; however, this slow drift did not affect the test results since the thermometer was calibrated at room temperature and by nitrogen and helium vapor pressure measurements each time the apparatus was cooled from room temperature and the helium temperature calibration was re-established before each test. Systematic error in calibration is no more than $\pm$ 0.01 K, and experience with pulses repeated seconds apart and months apart indicate that heater-to-bath temperature difference measurements were repeatable within $\pm$ 5 percent. While instantaneous heater power is accurate to $\pm$ 0.1 percent, the variation during a pulse from the stated average power due to heater resistance change ranges from $\pm$ 1 percent at 0.1 W/cm^2 to a maximum of $\pm$ 8 percent at 20 W/cm^2. Time measurements are initiated by the triggering of the digital oscilloscope at the beginning of a pulse; however, the time to achieve essentially steady power is approximately five microseconds later than the triggering time. No corrections were made in the plotted data for the finite power rise time.

The liquid helium results are complicated by a significant Kapitza resistance which is added in series with the thermal boundary layer resistance. Because of the Kapitza resistance the heater temperature does not rise simply in proportion to $\sqrt{\theta}$ during the non-boiling transient heat transfer period. For helium

$$\Delta T = \Delta T_K + \Delta T_f,$$

where ΔT_f is given by eq. (13). For helium the quantity

$$\frac{K_s}{\sqrt{\alpha_s}} + \frac{K_f}{\sqrt{\alpha_f}} = 0.0118 \frac{Ws^{1/2}}{cm^2 K} . \qquad (18)$$

The surface temperature step ΔT_K due to Kapitza resistance is given by equation (15).

The non-boiling trnasient heat transfer calculations using eqs. (13), (2), (3) and the constant (4) are shown in figure 3. The earliest time at which temperature can be measured is about $20(10)^{-6}$ s. It can be seen that except at low power levels the non-boiling transient heat transfer period is over by $20(10)^{-6}$ s; however, the calculated temperatures at $\theta < 20(10)^{-6}$ s are consistent with the measured temperatures. The non-boiling transient period leads either to steady state nucleate boiling as in the 0.1 to 0.6 W/cm^2 curves or to a metastable nucleation period at higher power levels. The length of time the metastable nucleation period lasts decreases as the power level increases. (The steady state nucleate boiling at power levels below 0.6 W/cm^2 might be thought of as a "metastable" nucleation period which lasts indefinitely). At power levels of 1 W/cm^2 and above, the metastable nucleation period leads to a period of relatively slow transition to stable film boiling. At the highest power levels of 9, 15 and 25 W/cm^2 the transition to film boiling has already begun at the earliest possible measurement times. The transition times become longer at higher power so that steady state film boiling is reached at about 0.1 s for all power levels.

The steady state nucleate and film boiling power level versus temperature rise relationships are shown along with data from previous research[9] in figure 4. The new data, including the value of the peak steady state nucleate boiling heat flux, compares well with the previous data and Kutateladze correlation but with nucleate boiling ΔT slightly on the high side. It is interesting to note that the metastable nucleation data falls on an extension of the nucleate boiling curve above the steady state peak heat flux. The curve of Kapitza ΔT_K versus P is also shown on this curve. Obviously the nucleate boiling ΔT must be greater than ΔT_K, so the very large Kapitza resistance of the carbon film may account for the slightly higher than average nucleate boiling ΔT of the new data.

Heat transfer coefficients derived from these data are discussed in Section 4.6.

4.3. The Effect of Heater Orientation on Transient Helium Heat Transfer.

The orientation of the heating surface is known to be an important factor in steady state natural convection and boiling heat transfer because of the effect of buoyancy and orientation in inducing currents and in expelling

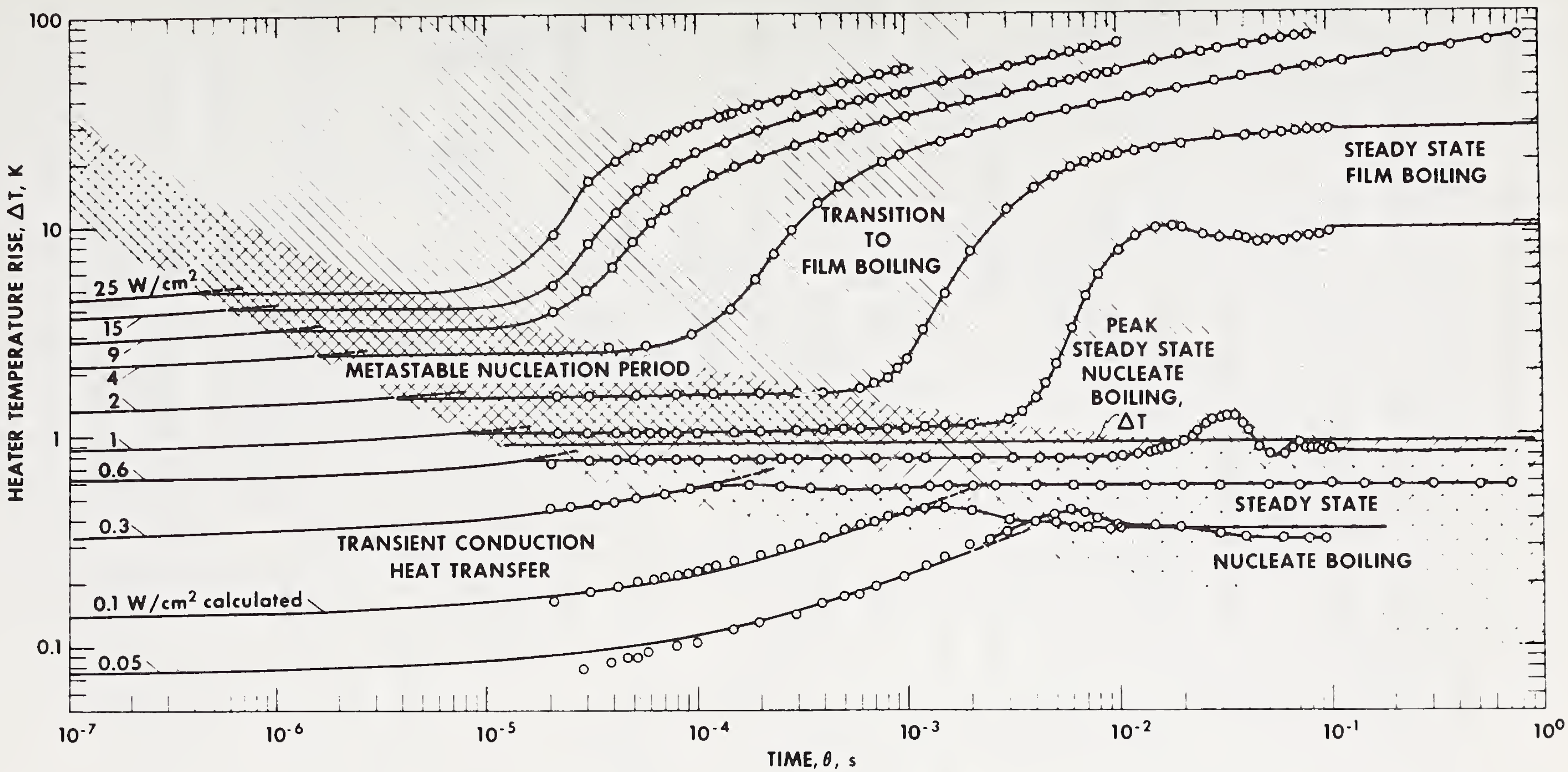

Figure 3. Transient heat transfer to a saturated liquid helium bath at 4 K from a vertical heater.

vapor bubbles. Particularly in boiling, bubbles formed on a vertical or upward facing surface tend to be expelled and swept away; whereas, bubbles forming on a downward facing surface tend to be trapped and form a thick insulating film which results in poorer cooling. Data such as shown in figure 3 for the vertical heater have been obtained for the horizontal facing up, facing down, and 45 degree facing down positions. Comparisons are shown in figure 5 for power levels of 0.1 W/cm^2 in the steady state nucleate boiling regime, and 1.0 and 5 W/cm^2 in the film boiling regime. Little effect of orientation is seen in the transient regime; however, the downward facing heater finally rose markedly above the other orientations particularly in steady state film boiling. The effect is less pronounced in the nucleate boiling range but still exists there. In general, vertical and horizontal facing up positions are nearly the same; 45 degree tilted downward gives the medium temperature, and a heater facing straight down produces the highest temperatures.

4.4. Pressure Effects

In all tests of figure 6 the bath bulk temperature was maintained at approximately 4.02 K; the pressure is raised to 0.151 MPa (▲ symbol) and 0.310 MPa (□ symbol) as opposed to atmospheric pressure, 0.086 MPa (○ symbol). For a heat flux of 0.1 W/cm^2 the temperatures for the three pressures follow nearly the same transient conduction curve; however, the higher pressures lead to decidedly higher steady state temperatures as would be expected; the increase in surface temperature in going from 0.086 MPa to 0.151 MPa is approximately the same as the increase in saturation temperature between those pressures. It should be noted that the highest pressure, 0.310 MPa is above the critical pressure, yet the steady state temperature is consistent with the boiling curves

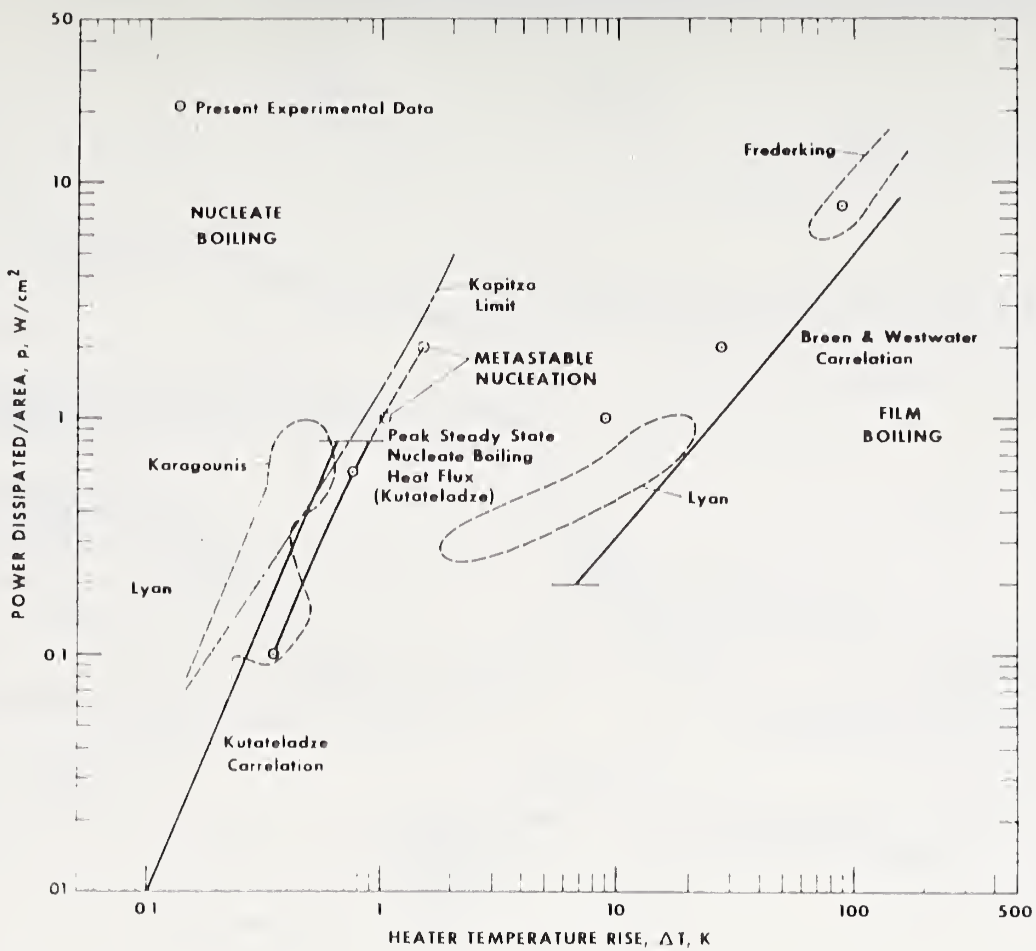

Figure 4. Comparison between the present experimental data in steady state boiling and previous research as discussed in Reference 9.

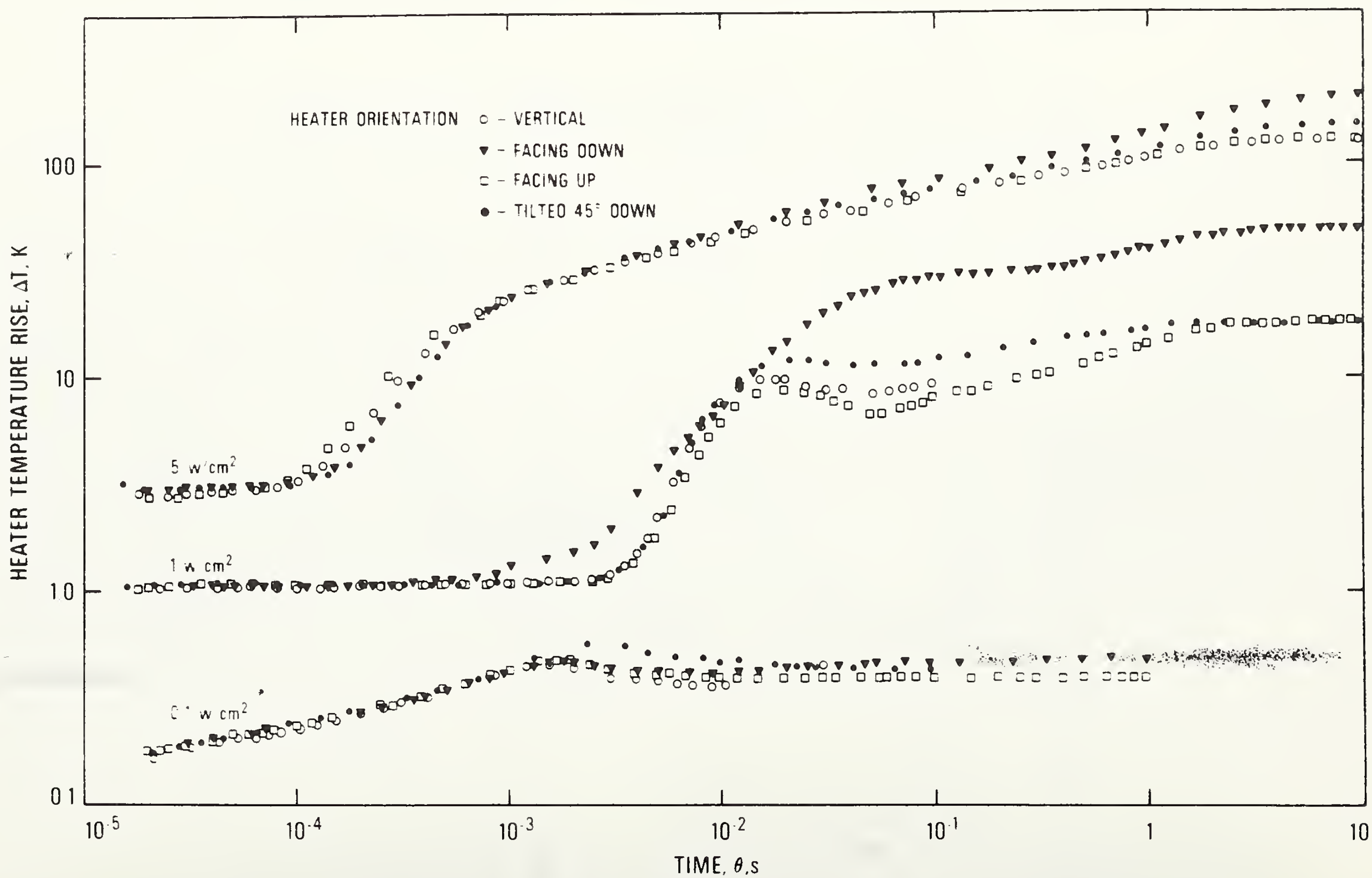

Figure 5. Comparison of heater temperature rise for the four orientations.

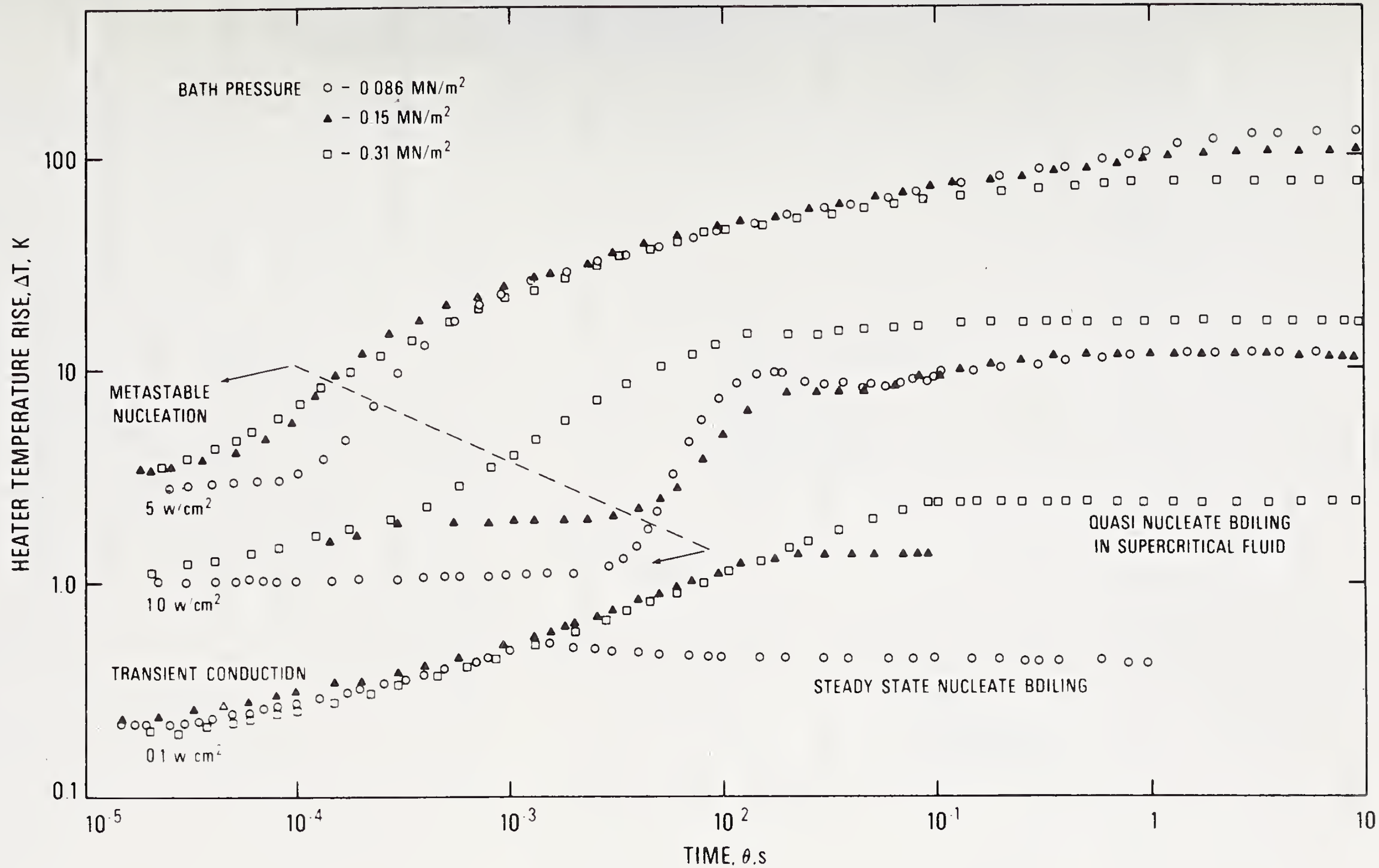

Figure 6. Effect of bath pressure on the temperature history for a vertical heater in 4 K liquid and supercritical helium.

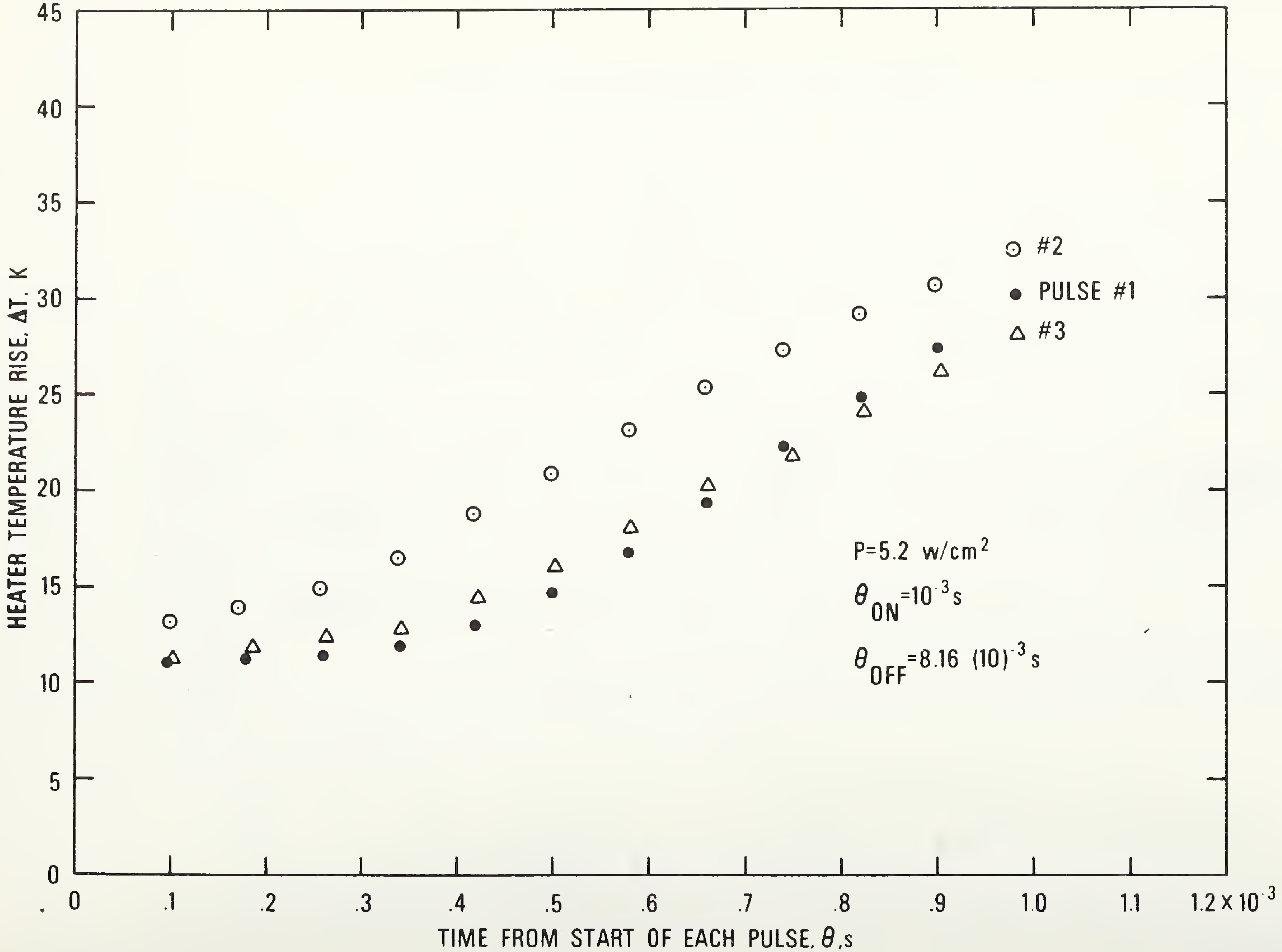

Figure 7. Temperature rise during repeated square wave pulses of 5.2 W/cm^2 for a pulse duration of 10^{-3}s and 8.16(10)$^{-3}$s between pulses.

lower pressures. The same phenomena can be seen in the metastable nucleation region to the left of the transition to film boiling at 1 and 5 W/cm^2. However, at 0.310 MPa the transient conduction curve leads directly to the transition to film boiling with no period of apparent nucleation. At 5 W/cm^2 only a small flat region can be observed at times less than 10^{-4} s for 0.086 MPa, and the two higher pressures are already in transition to film boiling at the earliest measured time.

It is interesting that the trend of steady state film boiling temperatures at 5 W/cm^2 is reversed from that of nucleate boiling. This is perhaps due to improved heat transfer through the vapor or supercritical fluid film at higher pressure and that the film would be thinner at higher pressure. This does not explain the behavior at 1 W/cm^2 in which the highest pressure produces the highest temperature as in nucleate boiling. However, the fact that the steady state heat transfer at 1 W/cm^2 exhibits some characteristics of nucleate boiling is not particularly disturbing since then heat flux is only slightly above the nucleate boiling peak and some regions of the surface are probably undergoing nucleate boiling.

4.5. Repetition of Pulses and Temperature Decay

When heating pulses are repeated the temperature rises while the power is being applied and decays asymptotically toward the bath temperature while the power is off. If succeeding pulses begin before the surface has fully regained equilibrium with the bulk fluid the entire level of temperature rises with each pulse until a stable condition is reached in which the total energy of each pulse is dissipated during each cycle. Figures 7 is an example of one of these temperature histories during 1 ms heating pulses of 5.2 W/cm^2 and an off-time of $8.16(10)^{-3}$ s. These tests show, for this example, that sustained pulses would produce a peak temperature of approximately 30 K above the bath temperature.

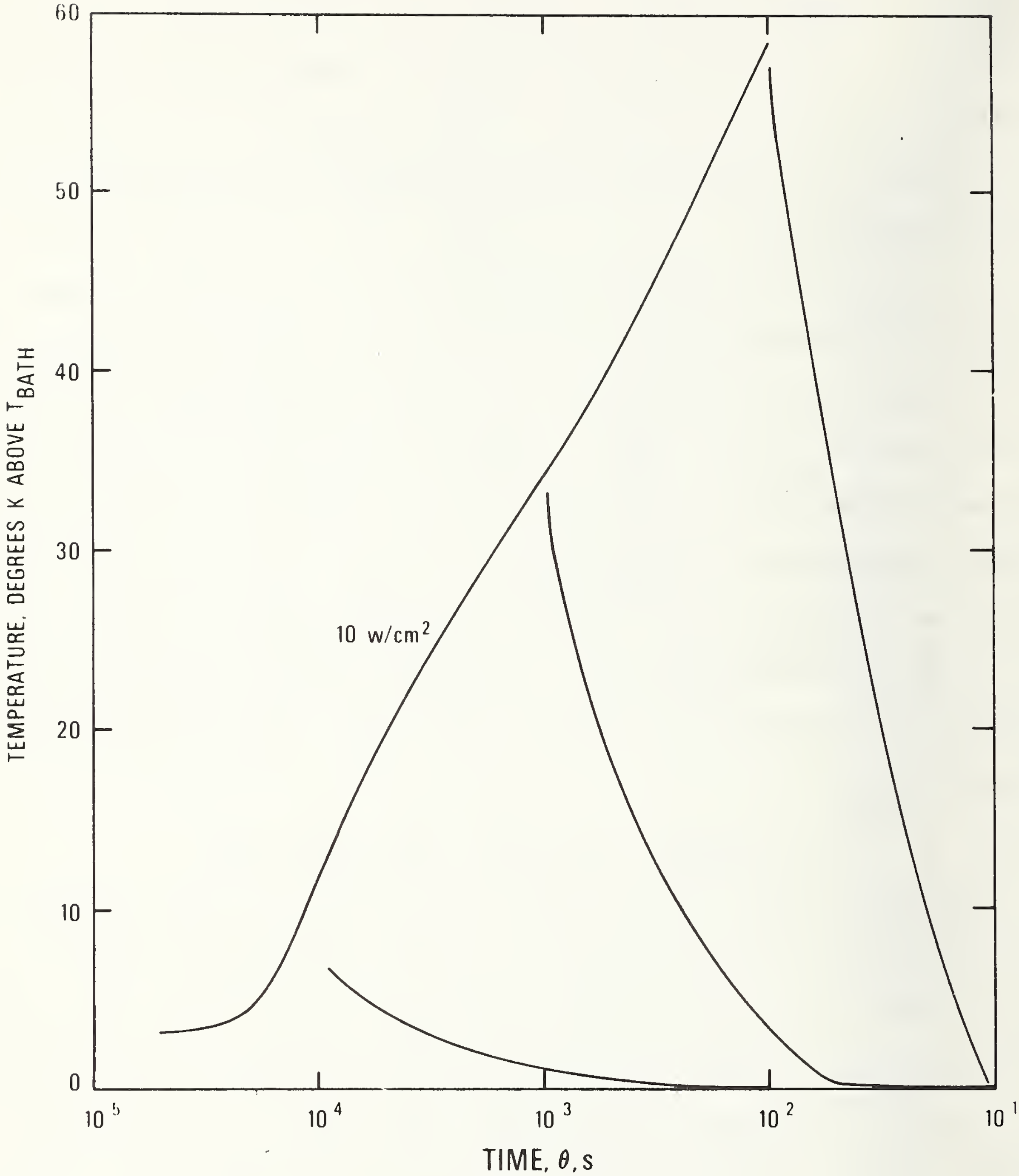

Figure 8. Temperature decay following 10 W/cm^2 pulses of 10^{-4}, 10^{-3}, and 10^{-2}s duration.

Figure 8 shows a typical temperature decay following square wave heat pulses. The nature of these decay curves depends on not only the heat flux during the pulse but on the temperature reached at the end of the pulse and the duration of pulse. All of these factors affect the boundary layer temperature profile as the decay begins. These figures confirm the previous interpretation of the initial temperature step as being due to Kapitza resistance. Following the pulse, when the power is essentially removed (except for a small measurement current) and Kapitza resistance should disappear, the temperature does in fact return to the bath temperature.

4.6. Heat Transfer Coefficients

Heat transfer coefficients were calculated for all of the experimental data. Some of these are shown in figure 9 for a vertical heater and atmospheric pressure, along with curves calculated from,

$$h = \frac{P_f}{(\Delta T_c + \Delta T_K)} \tag{20}$$

where ΔT_c and ΔT_K are from equations (13) and (17).

At the lowest power levels of .051 and 0.1 W/cm^2 the experimental data is seen to follow the transient calculations until steady state nucleate boiling is reached. The steady state condition is preceeded in both those cases by a dip in heat transfer coefficient corresponding to the slight temperature overshoot seen in figure 3. It is seen that the time to reach steady state varies all the way from 10^{-5} s to 1 s.

At 0.6 W/cm^2 a vestige of the transient curve is seen just before nucleate boiling begins at about $3(10)^{-5}$ s; however, at higher power levels transient conduction curves apparently intersect the boiling or metastable boiling curves before the earliest measurement time. The assumed metastable nucleation curve for 24 W/cm^2, and extrapolation of those curves for 2 W/cm^2 and 8 W/cm^2 to the transient conduction intersections are shown in dashed lines in figure 9. It should be emphasized that the extrapolated curves are rough estimates.

The Kapitza resistance of the carbon heat transfer surface may be representative of materials used for insulation but is considerably higher than typical base metals. The heat transfer coefficients for heaters with negligible Kapitza resistance has been estimated by subtracting ΔT_K from equation (17) from the observed ΔT. The transient conduction heat transfer coefficient then becomes independent of P. These curves in figure 10 indicate that heat transfer coefficients at very early times may be three times as large for materials with negligible Kapitza resistance as for the carbon film.

Negligible Kapitza-resistance heat transfer coefficients such as shown in figure 10 are compared with the experimental data of Jackson[2] in figure 11. The 0.031 cm manganin wire used as heater and thermometer by Jackson may account for the rather large differences between the two sets of data since surface curvature has a pronounced effect on bubble nucleation when the radius of curvature is on the order of the bubble radii. Even though there are points of agreement on the curves of 1.5 to 2.2 W/cm^2 the slope $dh/d\theta$ is considerably steeper in the present data and Jackson's data indicates larger dependence of h on heat flux.

5. CONCLUSIONS

Transient heat transfer experiments using fast response carbon thin film heater thermometers in a static helium bath show the effects of heat flux, heater orientation and pressure on surface temperature and heat transfer

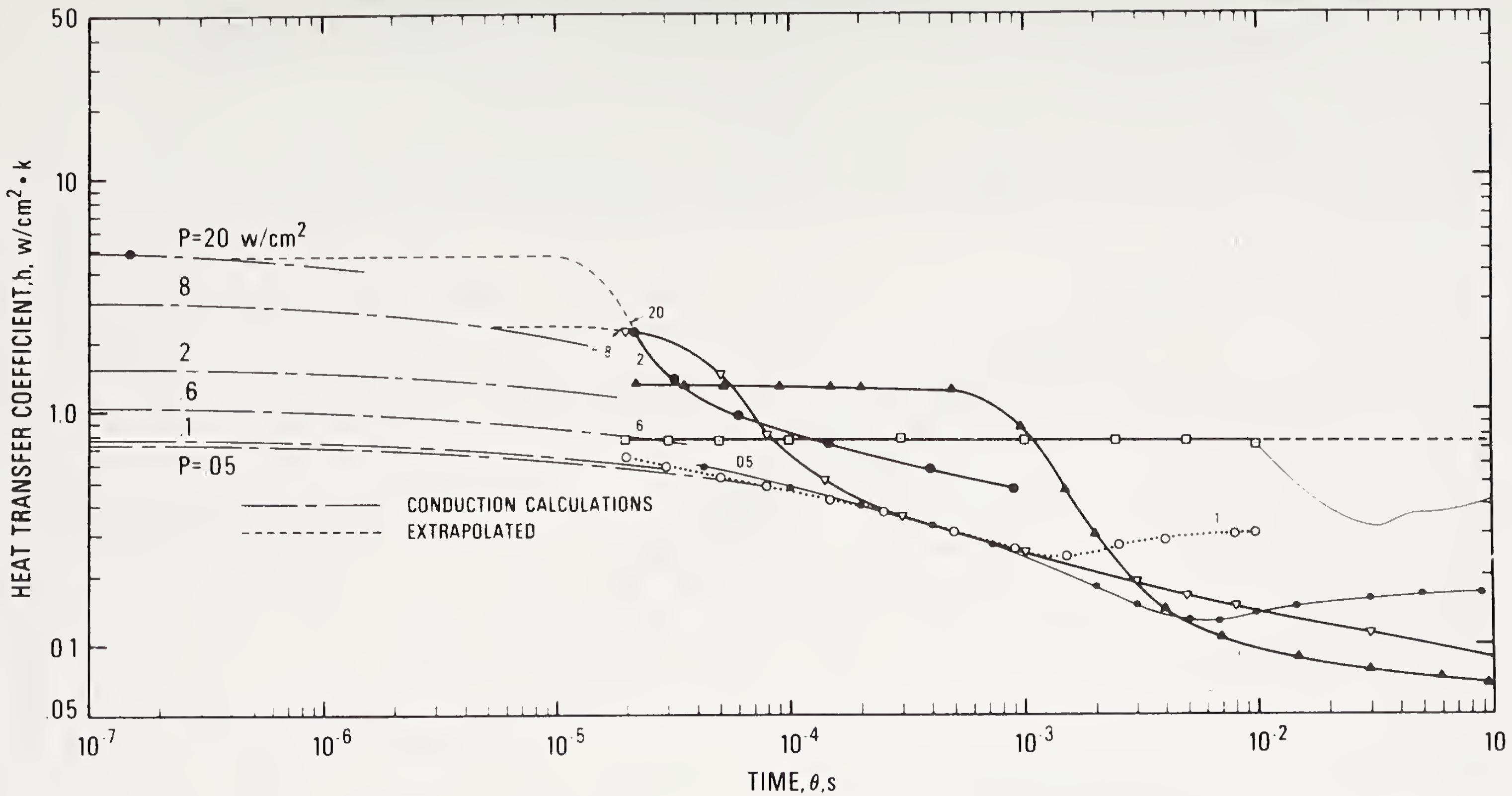

Figure 9. Transient heat transfer coefficients for a vertical carbon film heat transfer surface in saturated liquid helium at 4 K.

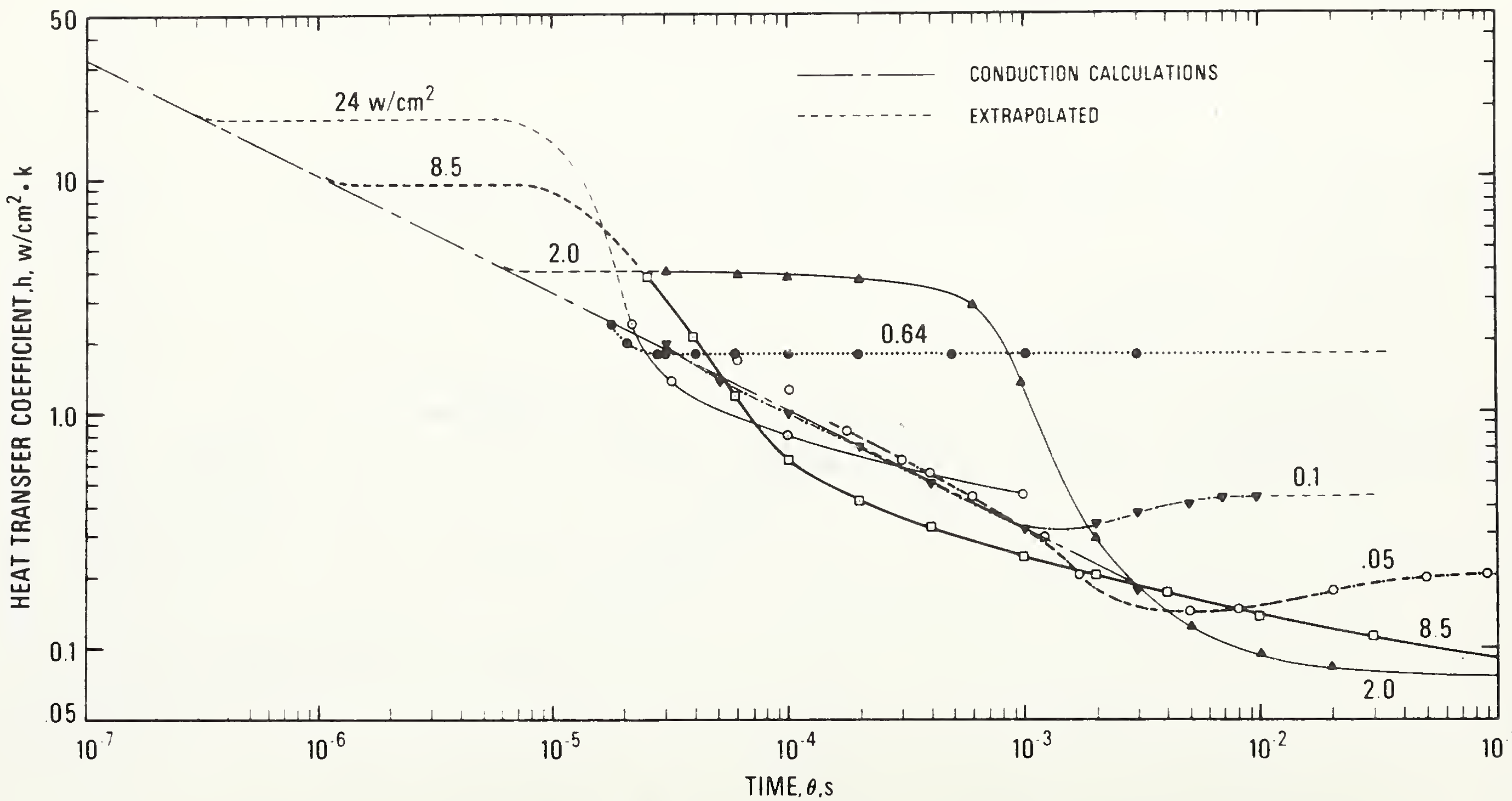

Figure 10. Transient heat transfer coefficients for a heat transfer surface having negligible Kapitza resistance.

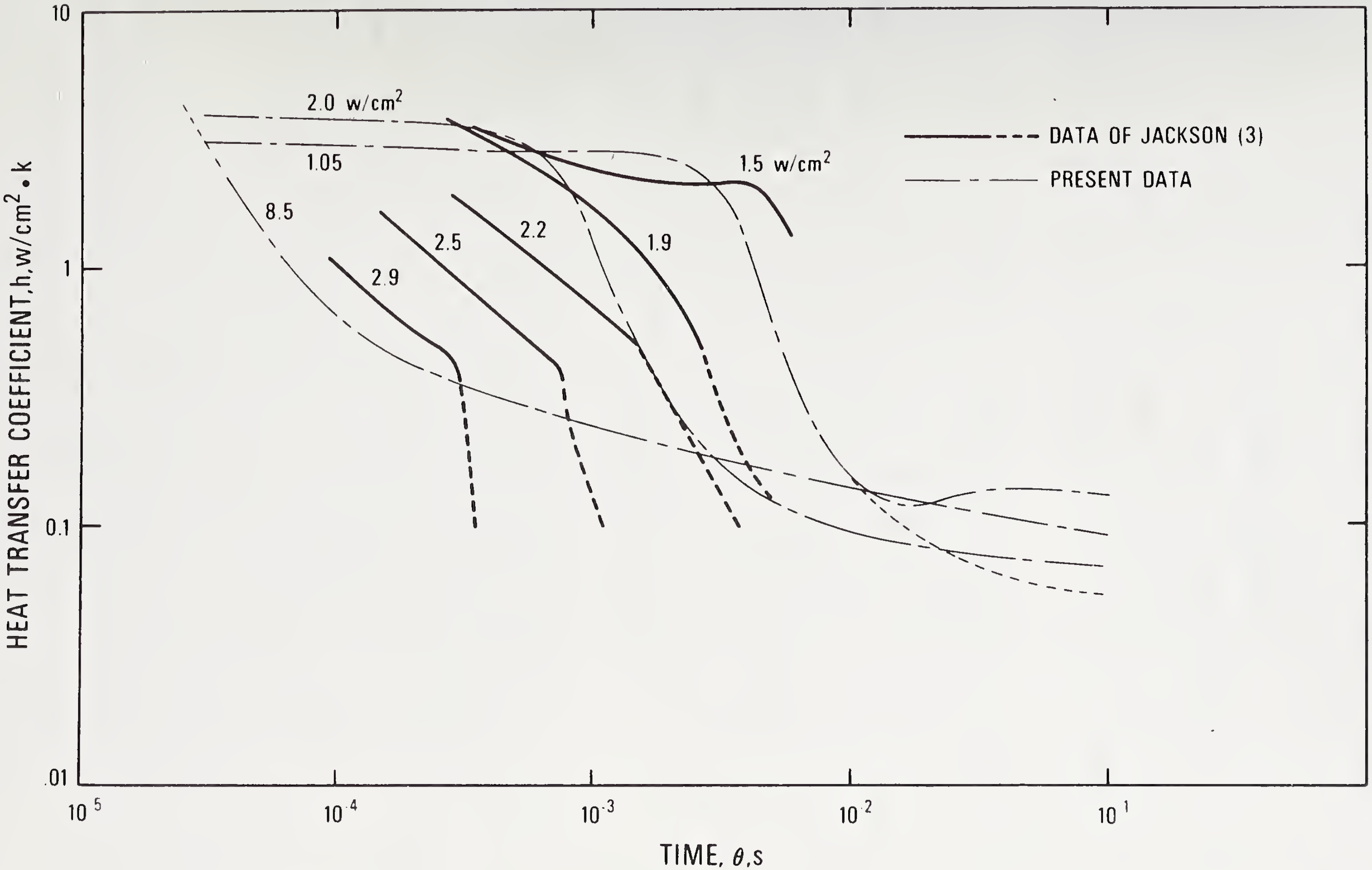

Figure 11. Comparison between the heat transient heat transfer coefficient data of Jackson [3] and the present data.

coefficients. For heat flux below the steady state peak nucleate boiling limit the surface temperatures follow calculations based on pure conduction (allowing for Kapitza resistance) until, after a small overshoot, the temperature reaches the steady state nucleate boiling level. Above the steady state nucleate boiling peak the transient conduction period leads to an apparent metastable nucleate boiling period followed by a transition to film boiling. All of these events vary systematically with heat flux. Steady state temperatures fall within the range of previous experimental data which, granted, allows considerable latitude.

A downward facing orientation of the heater increases the steady state temperature rise 20 to 60% over the other orientations, but has little effect during transients. Likewise, raising the coolant helium pressure has little effect during the transient conduction period but raises the steady state and metastable nucleate boiling temperature levels. The effect of pressure is reversed in steady state film boiling, apparently because measured pressure improves heat transfer through the vapor (or low density supercritical fluid) film.

Kapitza resistance for materials such as the carbon heat transfer surface used in these experiments may override much of the benefit which might have been gained from the initial high helium conduction heat transfer. For that reason the type of electrical insulating material or metal covering the superconductor and contacting the helium coolant will have a significant effect on transient cooling. Investigation of surface material effects as well as forced convection effects are planned for a continuation of this program.

Steady state nucleate boiling is reached in 10^{-5} to 10^{-2} s, but steady state film boiling conditions are not reached until 10^{-1} to 1 s from the beginning of a heat pulse. Since the initial heat flux is at least an order of magnitude higher than steady state, the time variations in heat transfer coefficients should be considered in the design and analysis of superconducting equipment if these transition times comprise a significant part of anticipated heating pulses.

6. NOMENCLATURE

c	heat capacity, j/g•K
h	heat transfer coefficient, $W/cm^2 \cdot K$
h_K	Kapitza conductance, $W/cm^2 \cdot K$
$h_{K,ref}$	Kapitza conductance at T_{ref}
K_f	thermal conductivity for the fluid, W/cm•K
K_s	thermal conductivity for the substrate, W/cm•K
P	power going into the fluid, W/cm^2
P_s	power going into the substrate, W/cm^2
T	temperature, K
T_{bath}	bath temperature, K
Z	coordinate normal to the heating surface, cm
α_f	thermal diffusivity of the fluid, cm^2/s
α_s	thermal diffusivity of the substrate, cm^2/s
ΔT_f	temperature of the fluid above the bulk temperature, K
ΔT_s	temperature of the substrate above the bulk fluid temperature, K
θ	time from the start of a heating pulse, s
θ_P	duration of a pulse, s

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NBSIR 78-877, Part Three

MAXIMUM PRACTICAL EFFICIENCY OF HELIUM TEMPERATURE REFRIGERATORS

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ABSTRACT

An ideal refrigerator using a perfect gas working fluid is defined which gives the efficiency of a refrigerator as a function of compressor and expander efficiency, heat exchanger temperature difference, and heat exchanger pressure drop. Although not suited to detailed hardware design, this approach clearly relates the overall cycle efficiency to component efficiencies. In contrast, computer studies of specific cycles using real fluid properties are usually such that the details tend to overshadow major trends. The results of the study show that in an efficient cycle the major losses are in the compressor and the cold end expansion device. For current compressor and expander efficiencies the maximum practical helium temperature refrigerator efficiency is about 37 percent of Carnot.

Key words: Component efficiency; cryogenics; efficiency; helium; maximum refrigerator efficiency; perfect gas analysis; refrigerator.

MAXIMUM PRACTICAL EFFICIENCY OF HELIUM TEMPERATURE REFRIGERATORS

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I. Introduction

We know the efficiency of large helium temperature refrigerators is limited by the efficiency of the available compressors, expanders, and heat exchangers. But which component arrangement or cycle will yield maximum refrigerator efficiency and which component in the refrigerator contributes most to the refrigerator inefficiency? Answers to these questions can be found by using a computer and real gas properties to investigate various refrigeration cycles that use components with variable efficiencies. However, the results of such a computer study are usually clouded by the fact that a) One is never quite sure that a totally different and unconsidered cycle may not yield higher efficiencies and b) The computer studies are usually such that the details tend to overshadow the more obvious major trends. We describe here a simple defined refrigeration cycle using an ideal gas working fluid which determines both the major sources of inefficiency in a refrigerator and the maximum refrigerator efficiency based on the efficiencies of available components. Although this approach overlooks many of the details required to design a real refrigerator such as heat exchanger temperature pinches, non-ideal fluid refrigeration effects and economic and reliability considerations, it offers the advantage of clarity and simplicity. The overall refrigerator efficiency is directly related to the individual component efficiencies and the maximum efficiency obtainable for a given set of component efficiencies is given.

II. EFFICIENCY OF DEFINED REFRIGERATOR WITH PERFECT HEAT EXCHANGER

A defined refrigerator cycle -- the schematic is shown on figure 1 -- has an isothermal compressor, a theoretical isothermal expander as the final expansion device, heat exchangers, and a variable number of expanders located at intervals in the heat exchanger to absorb heat exchanger losses. The isothermal expander is a theoretical device to obtain cooling from a perfect gas and it is analogous to the final expansion device of a refrigerator where the constant temperature cooling is a result of evaporating liquid at constant pressure. The expanders used to intercept heat exchanger losses are situated so the inlet temperature of each expander is higher than the discharge temperature of the preceeding expander by the temperature difference of the heat exchanger. The heat exchanger temperature difference is proportional to the absolute temperature as defined by a constant, $C = \Delta T/T$. This arrangement minimizes heat exchanger losses [1]. The isothermal expander is supplied by a separate compressor so its operating pressure ratio can be maintained constant while the number of loss absorbing expanders in the heat exchanger can be varied along with their operating pressure ratio.

Refrigerator efficiency is defined as the ideal compressor power divided by the actual compressor power minus the recoverable power from the expanders, or

$$\eta = \frac{P_I}{P_a - P_e} \times 100, \qquad (1)$$

where

η = refrigerator efficiency,

P_I = ideal compressor power required to produce a set amount of refrigeration,

P_a = actual compressor power required to produce the same amount of refrigeration,

and P_e = recoverable power from the expanders.

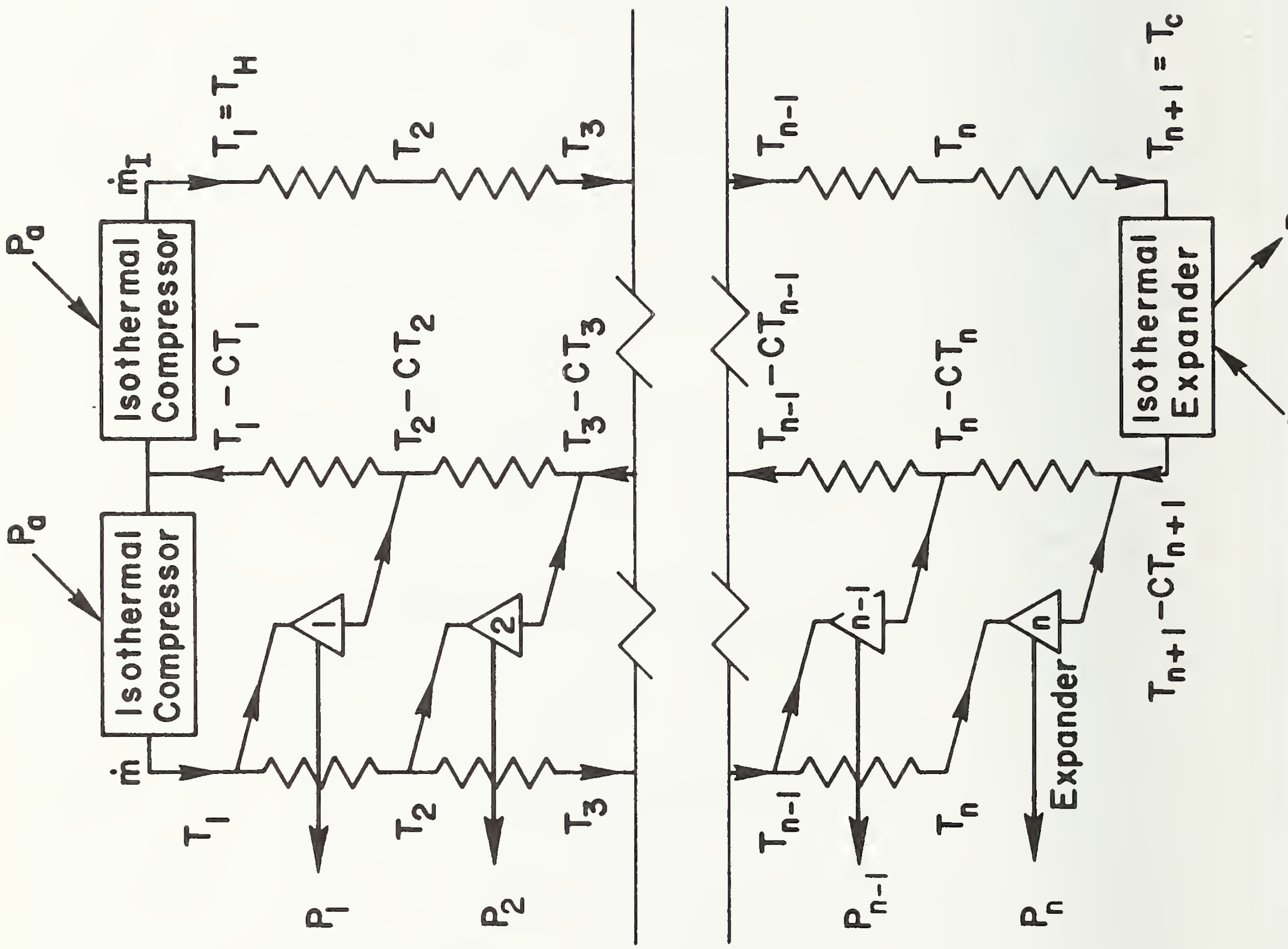

Figure 1. Defined refrigerator cycle.

The ideal power required by the compressor in an ideal refrigerator is,

$$P_I = Q(T_H - T_c)/T_c, \tag{2}$$

where

Q = refrigeration load,

T_H = rejection temperature (300 K),

and T_c = refrigeration temperature (4.2 K).

Actual compressor power required to supply the isothermal expander is

$$P_a = \frac{\dot{m}_I R T_H \ln Pr_c}{\eta_c} \tag{3}$$

where

$\dot{m}_I$ = isothermal expander mass flow rate,

R = gas constant,

Pr_c = compressor pressure ratio (P_H/P_L),

and η_c = compressor isothermal efficiency.

When the heat exchanger is perfect, no loss absorbing expanders are required so the total mass flow rate ($\dot{m}$) equals the mass flow through the isothermal expander or

$$\dot{m}_I = \frac{Q}{\eta_I R T_c \ln Pr_I}, \tag{4}$$

where

η_I = efficiency of the isothermal expander,

and Pr_I = pressure ratio of the isothermal expander.

Recoverable power from the isothermal expander is equal to the refrigeration load or $P_I = Q$. Since no expanders are required to absorb the heat exchanger losses, the cycle efficiency can be found by combining equations (1), (2), (3) and (4). This combination results in the equation

$$\eta = \frac{\dfrac{T_H - T_c}{T_c}}{\dfrac{T_H \ln Pr_c}{\eta_c \eta_I T_c \ln Pr_I} - 1} \times 100. \qquad (5)$$

This equation reduces to 100 percent when $Pr_c = Pr_I$ (resulting from no pressure drop in the system), and when the compressor and isothermal expander are 100 percent efficient, or $\eta_c = \eta_I = 1$. The effect of pressure drop on refrigerator efficiency is found for this simple system by increasing the compressor pressure ratio above the isothermal expander pressure ratio or $Pr_c = APr_I$; where $A = (1.0 + \Delta P/P)/(1.0 - \Delta P/P)$, and $\Delta P/P$ is the ratio of the pressure difference between the warm and the cold end to the absolute pressure in that stream. The efficiency of a simple refrigerator including some pressure drop effects but perfect heat exchange is shown in figure 2. From the general trend of the efficiency versus pressure ratio, the maximum refrigerator efficiency occurs at an infinite pressure ratio. Because an infinite pressure ratio is unrealistic and because a pressure ratio of 50:1 is well above the knee in the curve, we chose to maintain the isothermal expander pressure ratio constant at 50 for the remainder of the investigation. From equation (5), it is apparent that the compressor and isothermal expander efficiency have an equal effect on refrigerator efficiency.

III. EFFECT OF IMPERFECT HEAT EXCHANGER ON DEFINED REFRIGERATOR EFFICIENCY

The effect of heat exchanger temperature difference on refrigerator efficiency is found in a similar fashion. The additional compressor power required when expanders are needed to absorb the heat exchanger losses (loss absorbing expanders) can be found by using equation (3) but the compressor mass flow rate is now a function of the number of expanders and their pressure ratio. Pressure ratio is determined by the number of loss absorbing expanders because they are placed end to end.

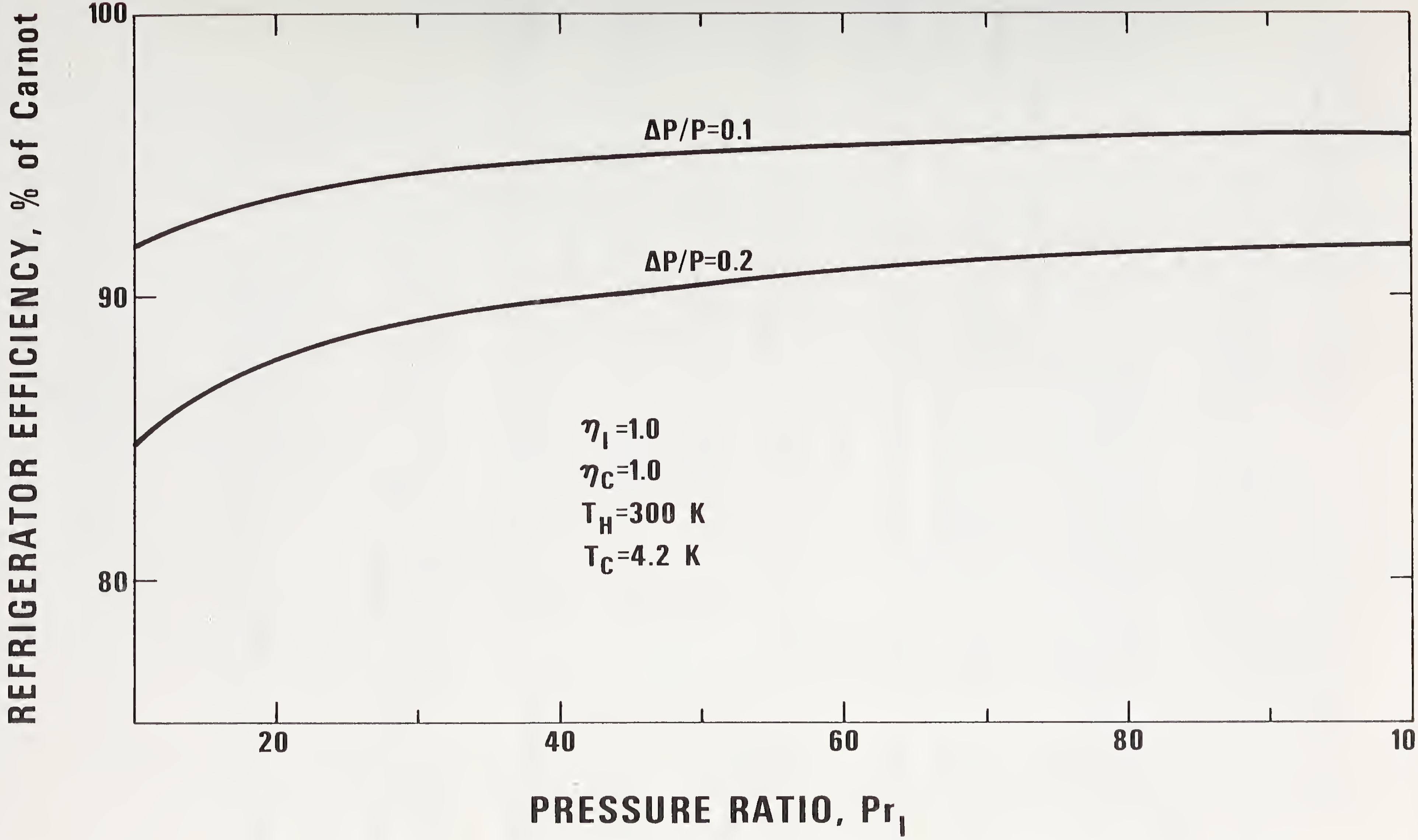

Figure 2. Defined refrigerator efficiency as a function of cycle pressure ratio. All refrigerator components are ideal except for some heat exchanger pressure drop.

The temperature drop across an expander operating with a perfect gas is

$$T_{in} - T_{out} = \eta_p T_{in}(1 - Pr_L^{-\alpha}), \tag{6}$$

where

T_{in} = expander inlet temperature,

T_{out} = expander outlet temperature,

η_p = isentropic efficiency of the loss absorbing expander,

Pr_L = pressure ratio across the expander (P_H/P_L),

α = $(k - 1)/k$ (0.4 for helium),

and k = ratio of specific heats (1.667 for helium at ambient temperature and zero pressure).

From figure 1, expression for T_2, T_3, ---- T_{n+1}, can be written based on equation (6) as,

$$T_2 = T_H[(1 - \eta_p + \eta_p Pr_L^{-\alpha})/(1 - C)], \tag{7}$$

$$T_3 = T_2[(1 - \eta_p + \eta_p Pr_L^{-\alpha})/(1 - C)], \tag{8}$$

and $$T_{n+1} = T_n[(1 - \eta_p + \eta_p Pr_L^{-\alpha})/(1 - C)], \tag{9}$$

where C = constant used to define heat exchanger temperature difference, $C = \Delta T/T$.

Assuming the same efficiency for all the precooling expanders and substituting to maintain T_H in the equation yields,

$$T_{n+1} = T_C = T_H[(1 - \eta_p + \eta_p Pr_L^{-\alpha})/(1 - C)]^{N_{exp}} \tag{10}$$

where N_{exp} = number of loss absorbing expanders.

Solving for Pr_l yields,

$$Pr_L = \left[\frac{\eta_p}{(1 - C)(T_C/T_H)^{1/N_{exp}} + \eta_p - 1}\right]^{1/\alpha} . \tag{11}$$

Note that the value of the fraction within the brackets must be positive so that as the efficiencies of the heat exchanger and expander are decreased more loss absorbing expanders are required.

The mass flow through the compressor supplying the loss absorbing expanders is the sum of the mass flows through the individual expanders. The total expander flow rate is determined by an energy balance around each heat exchanger expander combination as follows:

$$\dot{m}_{N_{exp}} = \dot{m}_I \frac{C}{1 - C}, \tag{12}$$

$$\dot{m}_{N_{exp}} + \dot{m}_{N_{exp-1}} = m_I\left[\frac{1}{(1 - C)^2} - 1\right], \tag{13}$$

and $$\dot{m}_{N_{exp}} + \dot{m}_{N_{exp-1}} + \dot{m}_{N_{exp-2}} = m_I\left[\frac{1}{(1 - C)^3} - 1\right]. \tag{14}$$

Thus, the total compressor flow for a variable number of loss absorbing expanders located end to end becomes,

$$\dot{m} = \dot{m}_1 + \dot{m}_2 ---- \dot{m}_{N_{exp-2}} + \dot{m}_{N_{exp-1}} + \dot{m}_{N_{exp}} = \left[\dot{m}_I \frac{1}{(1 - C)^{N_{exp}}} - 1\right] \tag{15}$$

where

$\dot{m}$ = total compressor mass flow rate

$\dot{m}_I$ = isothermal expander mass flow rate from equation (4),

and subscripts

1 = first or warmest loss absorbing expander ----,

N_{exp} = final or coldest loss absorbing expander.

The most efficient refrigerator will use the expander output power to reduce the power required by the compressors. The maximum recoverable power from the isothermal expander is equal to the refrigerator capacity, or

$$P_I = Q. \tag{16}$$

The maximum recoverable power from the loss absorbing expanders involves a summation. The power from a single expander operating with a perfect gas is,

$$P = \dot{m}C_p\Delta T, \tag{17}$$

or for expander number 1, figure 1, the recoverable power is,

$$P_1 = \dot{m}_1 C_p(T_H - (T_2 - CT_2)), \tag{18}$$

where

P_1 = power output of the warmest loss absorbing expander,

$\dot{m}_1$ = mass flow rate through the warmest expander,

and C_p = specific heat at constant pressure.

By substituting the temperatures from equations (7,8,9) and the expander mass flow rates from differences between equations (12, 13, 14, 15) into equation (18) while substituting to maintain T_H, the expander output power becomes:

$$P_1 = \frac{C}{(1-C)^{N_{exp}}} m_I\ C_p\ T_H(\eta_p - \eta_p Pr_L{}^{-\alpha}), \tag{19}$$

$$P_2 = \frac{C}{(1-C)^{N_{exp}}} m_I\ C_p\ T_H(\eta_p - \eta_p Pr_L{}^{-\alpha})(1 - \eta_p + \eta_p Pr_L{}^{-\alpha}), \tag{20}$$

$$P_3 = \frac{C}{(1-C)^{N_{exp}}} m_I\ C_p\ T_H(\eta_p - \eta_p Pr_L{}^{-\alpha})(1 - \eta_p + \eta_p Pr_L{}^{-\alpha})^2, \text{ --- }, \tag{21}$$

and $$P_{N_{exp}} = \frac{C}{(1-C)^{N_{exp}}} m_I\ C_p\ T_H(\eta_p - \eta_p Pr_L{}^{-\alpha})(1 - \eta_p + \eta_p Pr_L{}^{-\alpha})^{N_{exp-1}}; \tag{22}$$

where

P = expander power output,

η_p = expander isentropic efficiency,

Pr_L = pressure ratio across the expanders,

N_{exp} = total number of loss absorbing expanders,

and subscripts

1 = warmest loss absorbing expander ---,

N_{exp} = coldest loss absorbing exapnder.

Equations (19, 20, 21, and 22) are summed to establish the total recoverable power from the loss absorbing expanders or

$$P_{LA} = \frac{C}{(1-C)^{N_{exp}}} m_I C_p T_H(\eta_p - \eta_p Pr_L{}^{-\alpha}) \sum_{n=1}^{n=N_{exp}} (1 - \eta_p + \eta_p Pr_L{}^{-\alpha})^{n-1} \tag{23}$$

Total compressor power required by the refrigerator is the sum of the compressor power required for the isothermal expander loop (Pr_I = 50) and the compressor power required by the loss absorbing expanders or

$$P_a = \dot{m}_I \frac{RT_H \ln Pr_I}{\eta_c} + \left[\dot{m}_I \frac{1}{(1 - C)^{N_{exp}}} - 1\right] \frac{RT_H \ln Pr_L}{\eta_c} \tag{24}$$

The total recoverable expander power is the sum of the recoverable power from the loss absorbing expanders and the isothermal expander, or

$$P_e = \dot{m}_I C_P \frac{C}{(1-C)^{N_{exp}}} T_H(\eta_P - \eta_P Pr_L^{-\alpha}) \sum_{n=1}^{n=N_{exp}} (1-\eta_p + \eta_p Pr_L^{-\alpha})^{n-1} + Q \tag{25}$$

The overall refrigerator efficiency can now be calculated by

$$\eta = \frac{Q(\frac{T_H - T_c}{T_c})}{P_a(\text{equation 24}) - P_e(\text{equation 25})} \tag{26}$$

This equation includes the effect of component inefficiency and heat exchanger temperature difference. The effect of pressure drop can also be included by changing the compressor pressure ratios as they were changed previously. Increasing the pressure ratio across the compressor that supplies the isothermal expander by the factor A = (1.0 + ΔP/P)/(1.0 - ΔP/P) above the pressure ratio of the isothermal expander and increasing the pressure ratio of the compressor that supplies the work absorbing expanders above that calculated by equation (11) results in the maximum effect of pressure drop on cycle efficiency. This maximum effect was used in the data presented herein. A minimum effect of pressure drop on refrigerator efficiency would be to reduce only the pressure ratio of the isothermal expander below that of the compressor supplying it. The minimum and maximum effect were very nearly the same differing in the third significant digit.

IV. RESULTS

The ideal refrigerator efficiency with a perfect heat exchanger -- no pressure drop and no temperature difference -- is shown in figure 3. The pressure ratio of the isothermal loop was maintained at 50 during the calculation and no loss absorbing expanders were required. The maximum refrigerator efficiency was very nearly equal to the product of the compressor and isothermal expander efficiency.

The effect of pressure drop and heat exchanger temperature difference on refrigerator efficiency is shown on figures 4 and 5. Again the refrigerator efficiency is plotted versus expander efficiency but in this instance the refrigerator has loss absorbing expanders and their adiabatic efficiency is numerically equal to the isothermal expander efficiency. The cycle efficiency varied slightly with the number of loss absorbing expanders and if the maximum efficiency occurred with six or less expanders that point was plotted, otherwise the cycle efficiency for the minimum number of expanders required to produce a positive fraction in equation (11) was plotted.

The refrigerator efficiency depends primarily on the compressor and cold end expander efficiency in contrast to a liquefier where the efficiency of the precooling expanders has a significant effect [2]. The dependence of the

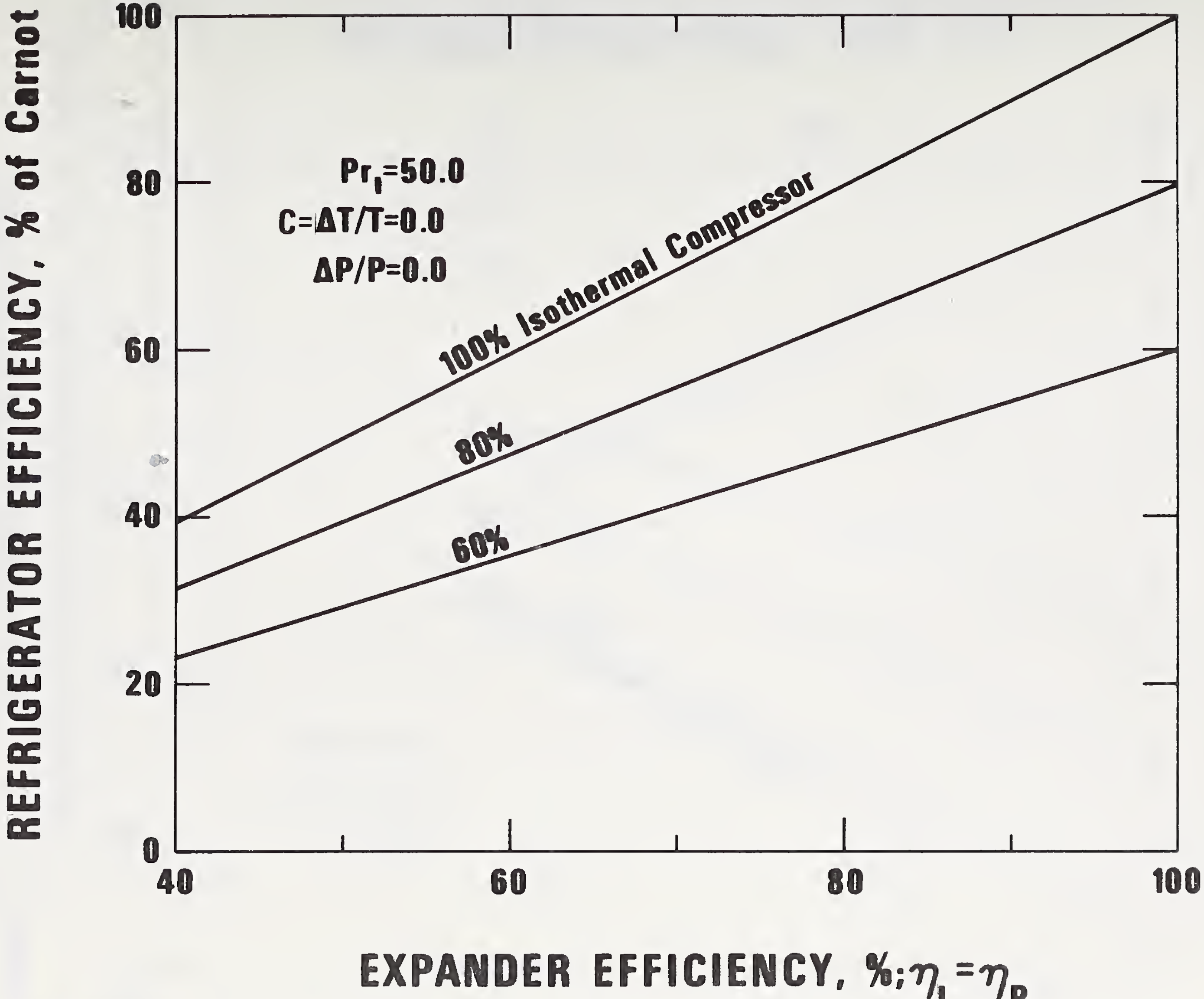

Figure 3. The dependence of the defined helium temperature refrigerator efficiency on expander efficiency. No heat exchanger temperature difference or pressure drop.

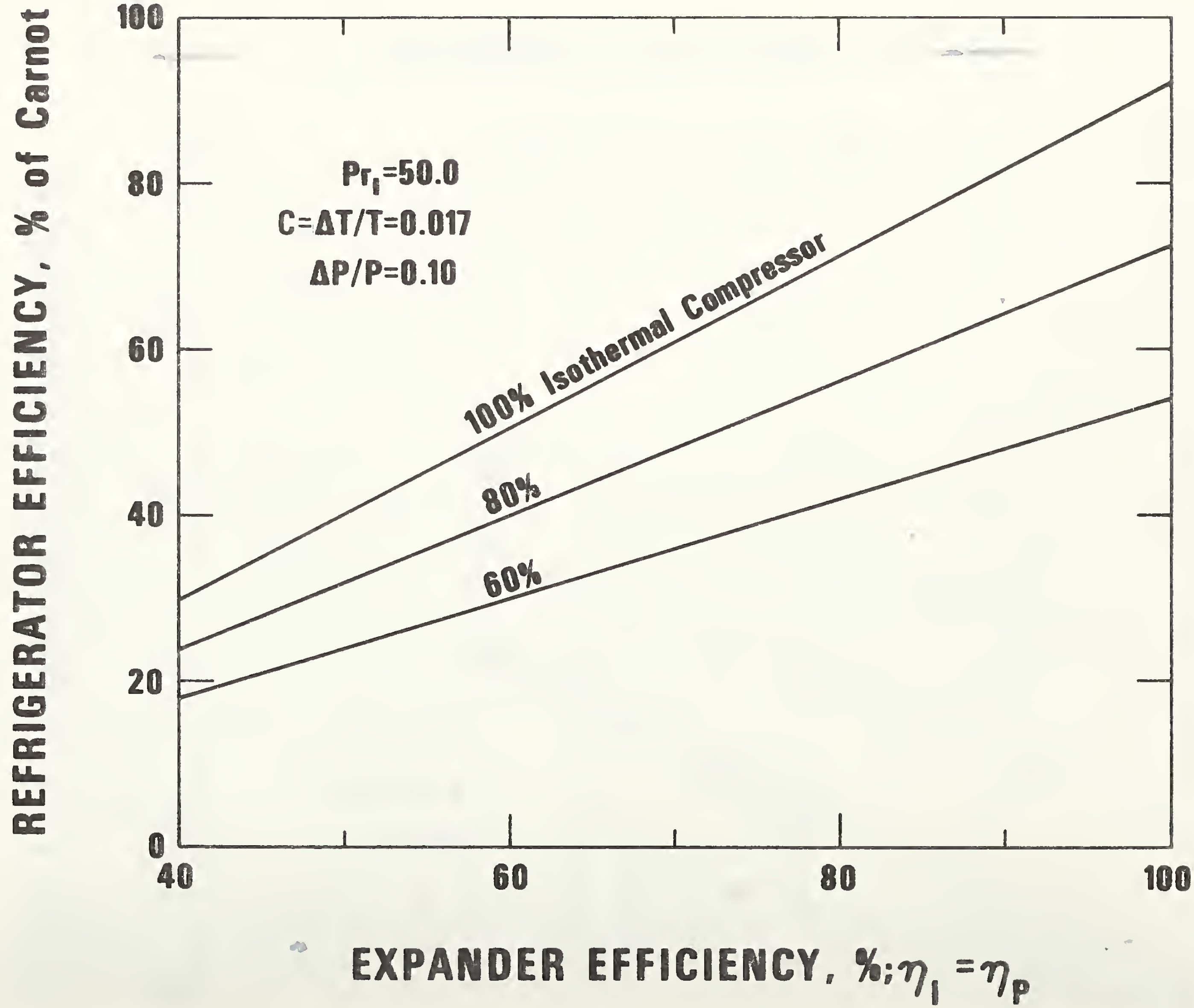

Figure 4. The dependence of the defined helium temperature refrigerator efficiency on expander efficiency. Includes the effect of heat exchanger temperature difference and pressure drop.

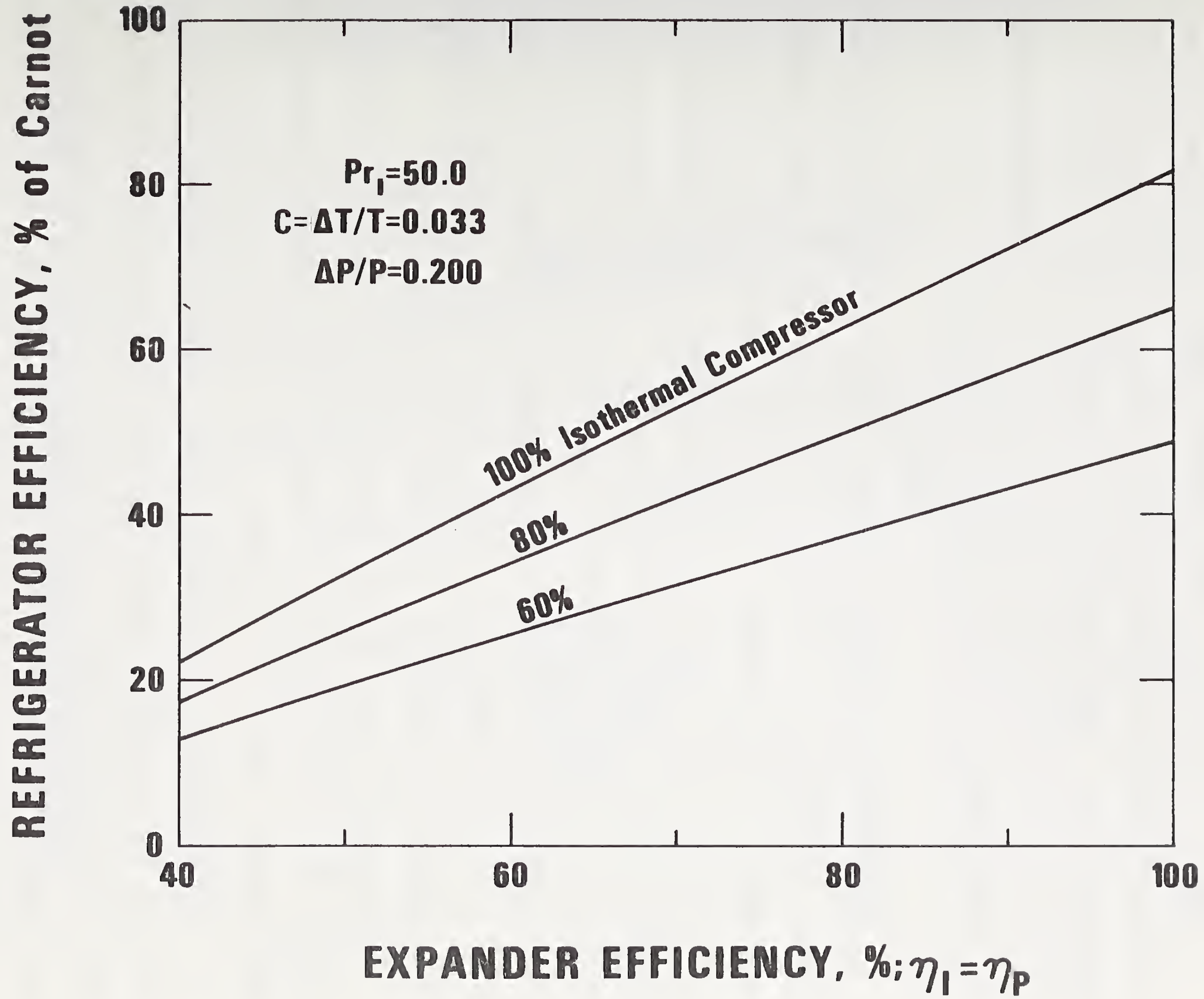

Figure 5. The dependence of the defined helium temperature refrigerator efficiency on expander efficiency. Includes the effect of heat exchange temperature difference and pressure drop.

refrigerator efficiency on the efficiency of the loss absorbing expanders is shown on figure 6. In this figure, the compressor efficiency and the isothermal expander efficiency are constant while the efficiency of the loss absorbing expanders varies. Comparing figure 6 with figures 4 and 5 shows that the efficiency of the isothermal expander affects the refrigerator efficiency more than the efficiency of the loss absorbing expanders.

V. CONCLUSIONS

The efficiencies derived using the relatively simple defined cycle can serve as a guide for the efficiency study of cycles using real components with real gases. If the efficiency of the real gas cycle is near the maximum found by the defined cycle, the real cycle is using the available components in their optimum operating locations and pressure ratios. On the other hand a significantly lower real gas cycle efficiency indicates that a new cycle may achieve higher efficiencies. Of course, the design of any practical unit is a compromise, so the real refrigerator efficiency may be lower than the defined cycle efficiency due to reliability or capital cost considerations.

The defined cycle also shows that the overall refrigerator efficiency is most directly effected by the efficiency of the compressor and the isothermal final expander. Thus, it is apparent that the efficiency of the cold end of a real refrigerator has a significant effect on overall refrigerator efficiency. Replacing the final Joule Thomson expansion valve with a more thermodynamically efficient expansion ejector or a wet expander (liquid in its exhaust), should significantly increase the efficiency of a refrigerator. If it is preferrable to keep the Joule Thomson expansion valve (JT valve), its inlet temperature should be reduced as closely as possible to the refrigeration temperature by the use of a cold expander. Using a cold expander-JT valve combination will also reduce the optimum pressure ratio of the JT valve.

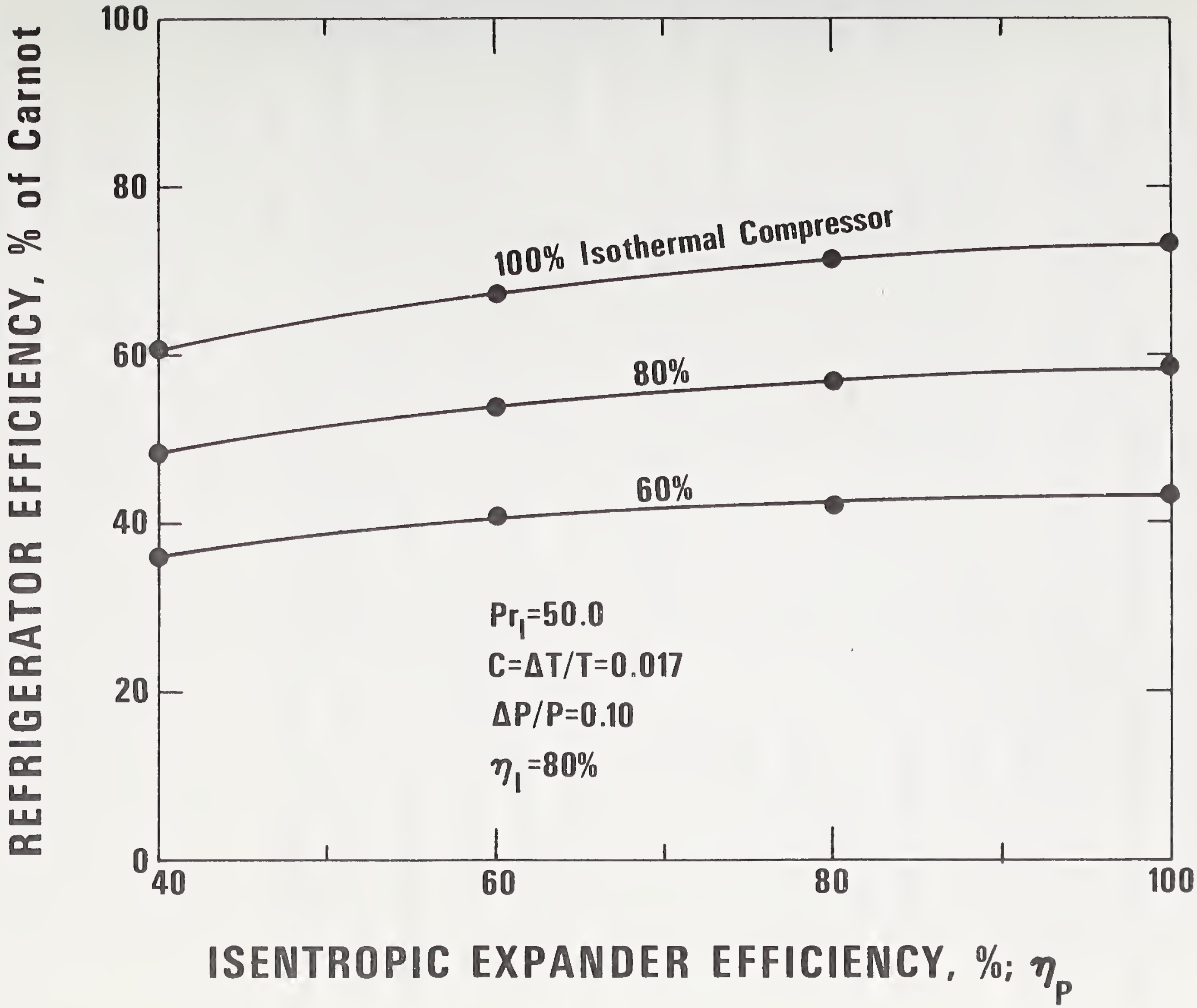

Figure 6. The dependence of the defined helium temperature refrigerator efficiency on the loss absorbing expander efficiency.

The compressor efficiency has a major influence on the refrigerator efficiency, but because the compressor has a long development history, increasing its efficiency above the current 60 percent of isothermal is probably difficult and expensive.

The results from the defined refrigeration cycle indicate that with a 60 percent isothermal compressor, 80 percent adiabatic precooling expanders, and an 80 percent isothermal expander the maximum refrigerator efficiency attainable is 48 percent (figure 2). Losses due to system pressure drops and heat exchanger temperature differences reduces the expected maximum efficiencies to 42 percent (figure 3) and 37 percent (figure 4). A reasonable maximum efficiency for a practical refrigerator may lie between 37 to 42 percent without considering other losses such as heat leak to the cold components or possible irreversibilities resulting from the properties of a real gas. Even with the irreversibilities of a real gas a refrigerator efficiency of 37 percent should be possible if the refrigerator is designed to achieve the maximum efficiency without compromises to reduce the capital cost of the refrigerator or to increase the reliability of the refrigerator.

VI. NOMENCLATURE

A proportionality constant used to define pressure drop

C proportionality constant used to define heat exchanger temperature difference, $C = \Delta T/T$

C_p constant pressure specific heat

k ratio of specific heats, C_p/Cv

$\dot{m}$ mass flow rate

$\dot{m}_I$ isothermal expander mass flow rate

N_{exp} number of expanders

P recoverable expander power

P_a actual compressor power

P_e recoverable power from the expanders

P_{eI} recoverable power from the isothermal expander

P_H high pressure

P_L low pressure

P_{La} total recoverable power from the loss absorbing expanders

Pr_c compressor pressure ratio

Pr_I pressure ratio of the isothermal expander

Pr_L pressure ratio of the loss absorbing expanders

Q refrigeration load

R gas constant

T_c refrigeration temperature (4.2 K)

T_H rejection temperature (300 K)

T_{in} inlet temperature

T_{out} outlet temperature

α $(k - 1)/k$ (0.4 for helium)

η refrigerator efficiency

η_c compressor isothermal efficiency

η_I efficiency of the isothermal expander

η_P adiabatic efficiency of the loss absorbing expanders

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NBSIR 78-877, Part Four

Helium Refrigerator Reliability Data Bank

T. R. Strobridge

In 1973, NBS was commissioned by the Fermi Laboratory to visit the five large industrial helium plants in Kansas and Oklahoma to gather data and assess the operating history of helium expansion machines. Two of the facilities separate helium from natural gas only, while there are four helium liquefiers at the other three locations. These, the world's largest helium liquefiers, are roughly equivalent to a refrigerator of up to 3 kW capacity at 4.2 K. The unpublished result was one of the first surveys of operating and failure history for helium liquefier expansion engines, heat exchangers, compressors, purification equipment and auxiliary equipment of this size; however, the data was only used as a basis for recommendations, and was never published.

Recently, ERDA-Power Transmission let two identical contracts to private industry to assess and study existing concepts and methods of cryogenic refrigeration for superconducting transmission cables. As part of the work, many of the major laboratories in this country and Europe were visited or contacted to gather operating and failure histories of helium refrigerators and liquefiers [Kadi and Longsworth], [Manatt, et al.]. Thus, the failure rate data has been extended and much other useful information was presented. In addition, both reports concluded that a refirgerator reliability data bank should be established to collect as much information as possible so that the maximum benefit may be realized from the reliability engineering sciences. NBS has also informally suggested on past occasions that this data bank should be established.

At the start of this program, our technical staff was uncertain about possible legal problems in the collection and public dissemination of this type of data. In discussing this broad legal question, we found that such a program already exists: The Government-Industry Data Exchange Program (GIDEP).

1. GIDEP

The GIDEP (Government-Industry Data Exchange Program) is a cooperative activity between Government and Industry participants seeking to reduce or eliminate expenditures of time and money by making maximum use of existing knowledge. The program provides a means to exchange certain types of technical data essential in the research, design, development, production and operational phases of the life cycle of systems and equipment.

In 1959, the Commanders of Army, Navy and Air Force ballistic missle programs began the Interservice Data Exchange Program (IDEP) which also included the ballistic missle contractors. The National Aeronautics and Space Administration (NASA) and the Canadian Department of Defence noted the value of the program and joined in 1966.

In 1970, by agreement of the Joint Logistics Commanders of the Military Services and NASA, the Navy assumed the responsibility for consolidated program management of the renamed Government-Industry Data Exchange Program (GIDEP). Pursuant to that same agreement, the program has been greatly expanded both in scope and quantity of data handled. Today, GIDEP is an efficient, working system for communications and the exchange of specialized information among government and industry activities.

The program is centrally managed and funded by the Government. Its participating organizations are: The United States Army, Navy, Air Force, Marine Corps, Defense Supply Agency, General Services Administrations, National Aeronautics and Space Administration, Federal Aviation Administration, Energy Research and Development Administration, and the National Security Agency as well as the Canadian Department of Defence and hundreds of industrial organizations.

Previously GIDEP was restricted to government activities and government contractors. Now, because of government emphasis on commercial off-the-shelf items, any activity which uses and/or generates the types of data GIDEP exchanges may be considered for membership. The program specifically excludes classified and proprietary information.

Participants in GIDEP are presently provided access to the four major data banks listed below. The proper utilization of these data banks can assist in the improvement of quality and reliability and reduce costs in the development and manufacture of complex systems and equipment.

Engineering Data Bank	Metrology Data Bank
Failure Rate Data Bank	Failure Experience Data Bank

2. Participation

It was suggested to us that the best way to get our Refrigeration Reliability operation started is to attend the scheduled GIDEP meeting and workshop during the first week of October, 1977. This was attended not only by the GIDEP staff, but also by industrial and government users of their services. We arranged to meet with the GIDEP Officer-in-Charge during this meeting.

In the meantime we have visited the GIDEP representative of a local aerospace company who has been very cooperative in discussing details of the program operation and in opening his files for our study. We further have discussed the operation with the NBS legal advisor, who tells us that collecting the data for input to the reliability sciences and even disseminating the data with generic names is permissible within the context of the GIDEP program.

After attending the workshop during the first week of October, the next step of our work will be to design the data collection system for this particular project within the GIDEP framework and format.

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15. SUPPLEMENTARY NOTES

16. ABSTRACT *(A 200-word or less factual summary of most significant information. If document includes a significant bibliography or literature survey, mention it here.)*

This is the first year-end report on a program of studies on cryogenic engineering data in support of magnetic fusion energy projects. The report is divided into four parts: (1) an assessment of the cryogenic engineering data base used in the MFE community, and recommendations for needed work for that base, (2) experimental progress on measurement of transient helium heat transfer; the data are of importance for magnet stability analysis, (3) presentation of a newly developed general technique for analyzing the efficiency of helium refrigerators of any configuration and thereby identifying sources of inefficiency, and (4) progress towards setting up a data bank on refrigeration system reliability. The technology assessment, item (1), is a revision of a preliminary version dated May 13, 1977, with inclusion of feedback received from both within and outside the MFE community.

Both the technology assessment, Part One, and the efficiency study, Part Three, have been formatted for individual publication subsequent to this report. The technology assessment will be distributed to those who received copies of the preliminary edition. The efficiency study will be submitted for publication in Cryogenics. Because of our fairly recent publication on transient heat transfer (Part Two, appendix A), it is appropriate to do further research in this field before writing it up for a follow-up publication in an outside journal.

17. KEY WORDS *(six to twelve entries; alphabetical order; capitalize only the first letter of the first key word unless a proper name; separated by semicolons)*

Efficiency; heat transfer; helium; pressure measurement; refrigeration; stability; superconducting magnets; transient heat transfer.

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