SURVEY AND EVALUATION OF TECHNIQUES TO AUGMENT CONVECTIVE HEAT TRANSFER

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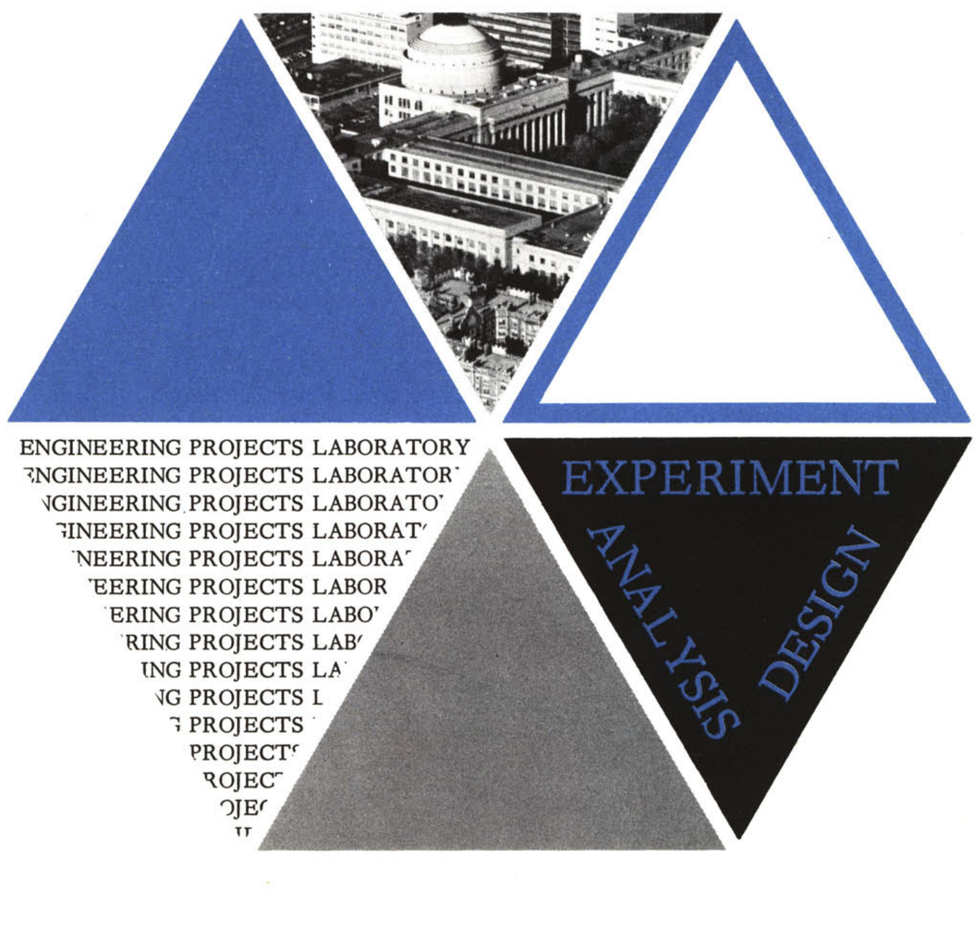

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SURVEY AND EVALUATION OF TECHNIQUES TO AUGMENT CONVECTIVE HEAT TRANSFER

by

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for

Massachusetts Institute of Technology

National Magnet Laboratory

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ABSTRACT

This report presents a survey and evaluation of the numerous techniques which have been shown to augment convective heat transfer. These techniques are: surface promoters, including roughness and treatment; displaced promoters, such as flow disturbers located away from the heat-transfer surface; vortex flows, including twisted-tape swirl generators; vibration of the heated surface or the fluid near the surface; electrostatic fields; and various types of fluid additives. Natural and forced convection situations for nonboiling, boiling, and condensation heat transfer are included. The conditions under which heat transfer is improved are summarized, and the efficiency of each technique is presented in terms of a performance criterion where possible.

ACKNOWLEDGMENTS

This study was supported by the National Magnet Laboratory of the Massachusetts Institute of Technology. H. L. Morton was a Sloan Fellow while preparing a preliminary survey as an S.M. Thesis in Mechanical Engineering.

The careful typing of Miss Lucille Blake is gratefully acknowledged. The assistance given by the staff of the M.I.T. Libraries is also appreciated.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

NOMENCLATURE

a	=	vibrational amplitude
C, n	=	constants
c_p	=	specific heat
D	=	channel diameter
D_e	=	channel hydraulic diameter
E	=	field strength
e	=	protrusion height
f	=	friction factor, vibrational frequency
G	=	mass velocity
h	=	heat-transfer coefficient
k	=	thermal conductivity
L	=	channel heated length, protrusion spacing
PL	=	sound pressure level
p	=	pressure
q	=	rate of heat transfer
q/A	=	heat flux
T	=	temperature
U	=	over-all heat-transfer coefficient
V	=	average velocity
w	=	mass flow
y	=	diameters per 180° tape twist
y/D	=	roughness parameter in Eq. (2)
x	=	vapor quality

α = roughness correlating parameter in Eq. (2)

Γ = $\frac{1}{\sigma} \frac{\partial \sigma}{\partial T}$

Δp = heated section pressure drop

ΔT = $T_w - T_b$

ϵ_o = permittivity of vacuum

ϵ^* = roughness parameter in Eq. (4)

κ = dielectric constant

μ = dynamic viscosity

ν = kinematic viscosity

ρ = density

ρ_e = electrical resistivity

σ = electrical conductivity

ϕ = heat flux

Dimensionless Groups

El = Electrostatic number = $\frac{\epsilon_o E^2}{\rho V^2} \Gamma (T_w - T_b)_i \left(\frac{\kappa^2 + 4\kappa - 2}{3}\right)$

Gz = Graetz number = $w\, c_p/kL$

Nu = Nusselt number = hD/k

Pr = Prandtl number = $c_p \mu / k$

Re = Reynolds number = VD/ν

St = Stanton number = Nu/Re Pr

Subscripts

a = augmentative data

BO = burnout condition

b = bulk fluid condition

cr = critical heat flux condition

d	=	displaced promoter data
e	=	extended surface data
f	=	film fluid condition $\left[(T_w + T_b)/2\right]$
i	=	condition at inlet of channel
o	=	non-augmentative data, condition at outlet of channel
P	=	evaluated at constant pumping power
Re	=	evaluated at constant Reynolds number
r	=	rough surface data
s	=	saturation condition, swirl flow data
v	=	vibration data
w	=	wall condition
Δp	=	evaluated at constant pressure drop

1. INTRODUCTION

1.1 Survey and Evaluation

Most of the ever-increasing research effort in heat transfer is devoted to analyzing what might be called the normal situation. For example, a cooling channel which is smooth, straight, and subject to no body forces other than gravity would be considered normal. However, the development of high-performance thermal systems has also stimulated interest in methods of augmenting heat transfer. The performance of conventional heat exchangers can be greatly enhanced by a number of augmentative techniques. On the other hand, certain systems, particularly those in space vehicles, may require an augmentative device for successful operation.

Several basic techniques have been developed which increase convective heat-transfer coefficients, usually at the expense of pumping power or external power supplied to the system. These are:

a. Surface promoters

b. Displaced promoters

c. Vortex flows

d. Surface or fluid vibrations

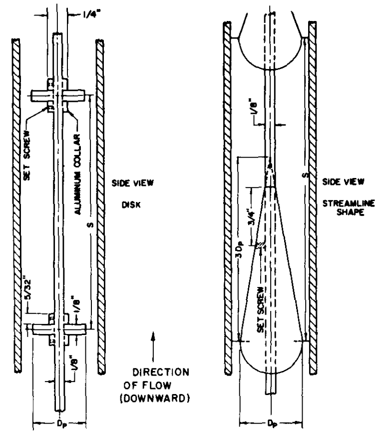
e. Electrostatic fields

f. Fluid additives

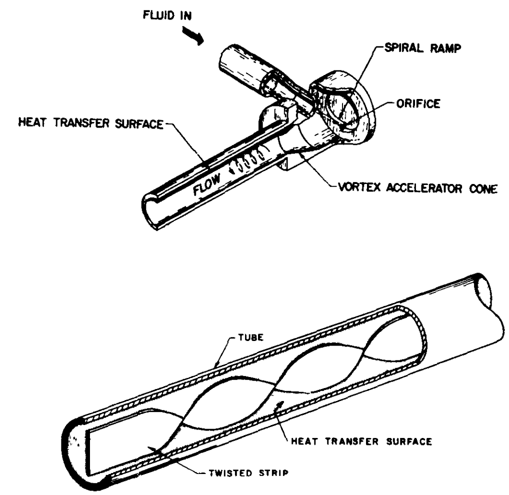
Figure 1 presents typical apparatus which have been used to demonstrate the effects of these techniques in the laboratory.

Potential augmentative techniques have frequently originated as nuisances. For example, varying amounts of surface roughness are present

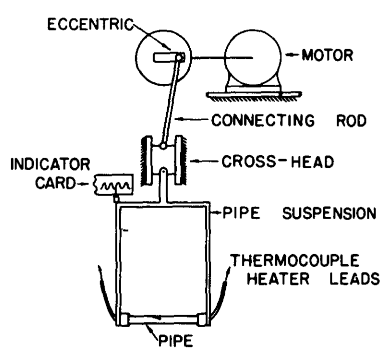
Rough Heated Surface (28)



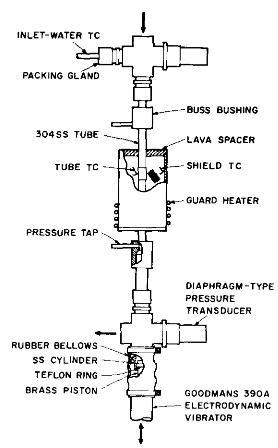
Displaced Promoters (44)



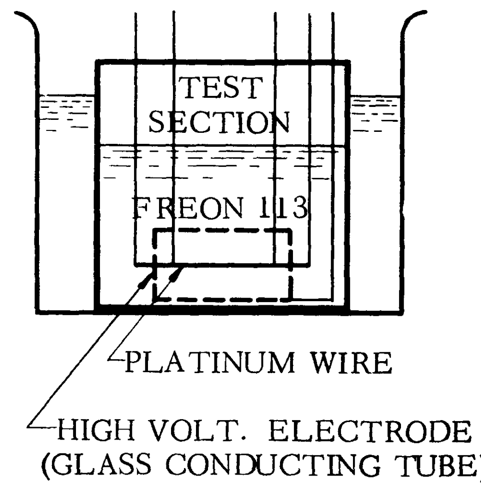
Vortex Generators (12)



Heated Surface Vibrator (72)



Fluid Vibrator (122)



Electrostatic Field Generator (133)

FIG. 1. EQUIPMENT USED IN AUGMENTATIVE EXPERIMENTS

in commercial piping. Since friction factors are substantially higher for rough tubes than for smooth tubes, it was natural to investigate whether there was an accompanying increase in the heat-transfer coefficient. Pulsations generated by reciprocating pumps were thought to be quite undesirable, until it was found that the pulsations boosted heat-transfer coefficients.

The designer of heat-transfer equipment has always been receptive to schemes which improve heat-transfer coefficients. It is only recently, however, that comprehensive experiments have been performed which clearly define the conditions under which an augmentative technique will improve heat transfer. Numerous investigations of each of these augmentative techniques have been reported; however, since these references are scattered throughout the engineering literature, a survey appears to be in order. A straight literature survey would be of limited usefulness to the engineer who contemplates using an augmentative technique. The conditions under which heat transfer can be improved and the efficiency of the method must also be summarized.

The effect of augmentation on heat transfer will in many cases be dependent on the mode of convective heat transfer. Natural convection and forced convection apply to both liquids and gases, whereas the several types of boiling are possible with liquids. Table I summarizes the types of convective heat transfer which have been investigated with the various augmentative techniques.

It is not sufficient to know simply that an augmentative scheme improves heat transfer. It may be possible that the normal system will perform equally well or even better for the same pumping power. Equal

TABLE I

INVESTIGATIONS OF AUGMENTATIVE TECHNIQUES

AUGMENTATIVE TECHNIQUE	NATURAL CONVECTION	Gases	Liquids	POOL BOILING	Surface-boiling heat transfer	Surface-boiling burnout	Bulk-boiling heat transfer	Bulk-boiling burnout	FORCED CONVECTION	Gases	Liquids	FORCED-CONVECTION BOILING	Surface-boiling heat transfer	Surface-boiling burnout	Bulk-boiling heat transfer	Bulk-boiling burnout	CONDENSATION
SURFACE PROMOTERS							√	√		√	√		√	√		√	√
DISPLACED PROMOTERS										√	√					√	
VORTEX FLOWS										√	√		√	√		√	
SURFACE VIBRATIONS		√	√				√			√	√						
FLUID VIBRATIONS		√	√			√	√	√		√	√		√	√			√
ELECTROSTATIC FIELDS		√	√				√	√			√				√	√	√
ADDITIVES			√			√		√					√	√			√

power consumption is an acceptable efficiency criterion; however, there are situations where an alternate criterion, such as equal pressure drop, is more useful. The final decision, however, will be made on the basis of minimum total cost, which includes manufacturing and pumping costs. Many studies neglect to report the complete information which will allow an estimation of the operating efficiency. For example, pressure-drop data are frequently omitted in channel flows with turbulence promoters, and transducer power requirements are often not reported for vibration studies. In the absence of such data, only the conditions under which heat transfer is improved can be presented.

This study is intended to be a concise, yet comprehensive, survey and evaluation of the augmentative field. A literature survey, consisting only of verified references, will be included in each section. The review articles which are available for several of the techniques will also be noted. The data from the investigations in each area will be evaluated and compared where possible. Experimental results will constitute the major part of the discussion; however, mention of the mechanism of augmentation will also be included.

1.2 Performance Criterion

Investigators presenting both heat-transfer and pressure-drop data for forced-flow systems with augmentative devices have usually evaluated the device according to some performance criterion. For the present work, since there is no standard measure of performance, it is necessary to select an appropriate criterion and apply it to the experimental data whenever possible. It appears most useful to compare the heat-transfer performance of the augmented and unaugmented systems on the basis of equal pumping power or equal pressure drop. The heat-transfer performance is characterized by the heat-transfer coefficient or burnout* heat flux. Equal pumping power in the heat exchanger is the most general criterion since it implies that pumping costs are unaffected by adoption of the augmentative scheme. However, equipment limitations may make it necessary to compare on the basis of equal pressure drop. With a centrifugal pump, for example, the pressure drop is relatively independent of flow rate. On the basis of equal power, the augmentative pressure drop will normally increase. The full benefit of the augmentative device can then only be realized by installation of a new, higher-head pump. If the equipment cannot be replaced, a comparison on the basis of equal pressure drop would be more useful as it gives the attainable improvement. In any case, the comparison on the basis of equal pumping power will appear to be the most favorable.

The present efficiency criterion for a nonboiling forced-convection system then becomes

$$(h_a/h_o)_P = f(\text{Re, Pr, promoter geometry}). \qquad (1)$$

Calculation details and additional comments are given in the Appendix. Assume, for example, that $(h_a/h_o)_P = 2$ for a particular promoter geometry. For a given exchanger the heat-transfer rate could be doubled for a constant temperature difference, or for the same q, the ΔT could be halved. If, on the other hand, the flow rate is maintained constant, the length could be halved for the same q and ΔT. For the case of constant exchanger

*Burnout and critical heat flux are used interchangeably in this report. The exact definition of the critical condition will vary with each investigation.

geometry, there may be a gain in performance even if $(h_a/h_o)_P < 1$. The flow rate for the augmented case will be less than that for the unaugmented case, and if the pressure drop in the remainder of the system is significant, an over-all saving in pumping cost can be achieved. Since the efficiency can be evaluated in general only for the heated section, this same comment applies if entrance and exit losses are a significant part of the exchanger pressure drop.

The final decision will, of course, be made on the basis of minimum total cost (e.g. (1)*). The present criterion considers only a portion of the operating cost, the pumping power. The remainder of the operating cost, including cost of the fluid and maintenance, as well as the fixed costs, including depreciation and taxes, can only be estimated for a particular system. For the surface promoter, displaced promoter, and vortex flow techniques, the installation cost is relatively small so that the pumping power becomes the dominant cost. On the other hand, a vibration or electrostatic-field device may be applied to a system where the flow is low, in which case the augmenter power and initial cost are most important.

*Numbers in parentheses refer to References listed beginning on page 117.

2. SURFACE PROMOTERS

The first three augmentative techniques--surface promoters, displaced promoters, and vortex flow--have frequently been lumped into the general catagory of turbulence promoters. It is thought, however, that a clearer survey and evaluation can be made if these techniques are considered separately.

Surface roughness was one of the first techniques to be considered seriously as a means of augmenting forced-convection heat transfer. Initially it was speculated that elevated heat-transfer coefficients might accompany the relatively high friction factors characteristic of rough conduits. However, since the commercial roughness is not well defined, artificial surface roughness has been employed. Surface roughness of either the protrusion or depression type can be obtained by machining. Protuberances can also be of the attached type, such as wire coils inserted inside tubes.

An extensive literature survey on rough surfaces by Bhattacharyya (2) was recently received. Most of the available experimental data, including correlations and analogy solutions, are summarized in this presentation.

2.1 Nonboiling Forced Convection

2.1.1 Flow Inside Tubes

Although extensive friction data are available for commercial tube and pipe, there appear to be few investigators who have measured both heat transfer and friction for the rougher commercial piping. The tests of Nunner (3) summarized in Fig. 2 indicate that $(h_r/h_o)_P$ is close to unity for several samples of commercial pipe.

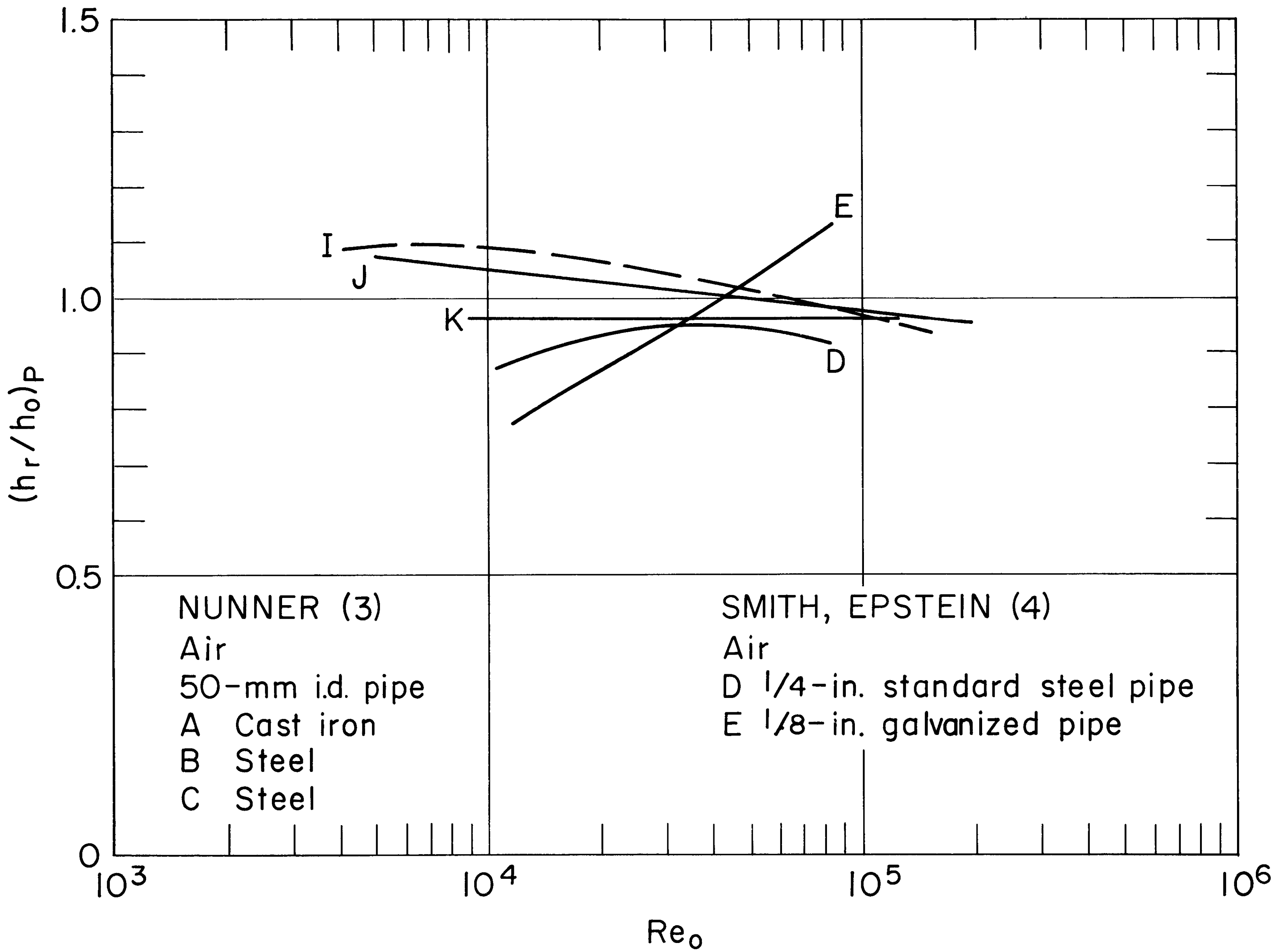


FIG. 2. PERFORMANCE OF TUBES WITH COMMERCIAL ROUGHNESS

Smith and Epstein (4) conducted a more extensive study of small-diameter commercial pipe. As seen by the curves in Fig. 2 for pipes with large and small apparent roughness, there is no clear trend as far as the performance is concerned.

It would seem that commercial roughness is quite random, and in general one cannot count on a favorable performance factor. Of course, if conventional heat-transfer correlations are used together with rough-pipe friction factors, the design will be somewhat conservative.

Systematic investigations of artificial roughness have appeared at regular intervals in the literature. A guide to many of these investigations is presented in Table II. It is seen that a great variety of roughness elements has been tested; however, it probably is safe to say that the optimum geometry has yet to be established. All of these schemes have been successful mechanically, at least on an experimental basis. As a result, they are worthy of consideration for commercial systems, providing that their performance is satisfactory.

One of the first studies of heat transfer and friction in tubes with well-defined roughness was conducted by Cope (5) in 1941. A special knurling process was used on three pipes to form geometrically similar, pyramid-type rough surfaces. Average measurements were made for cooling water in the test pipes. Small temperature differences contributed to uncertainty in heat-transfer coefficients; however, the rough-pipe data should be fairly reliable since smooth-pipe data are in reasonable agreement with conventional correlations. Figure 3 indicates that this type of roughness is advantageous only at low Reynolds numbers. The roughest surface appears to be most favorable.

TABLE II EXPERIMENTAL INVESTIGATIONS OF ARTIFICIAL ROUGHNESS--
NONBOILING HEAT TRANSFER AND FRICTION

Investigators	Description
Cope (5)	Cooling of water; internally knurled tubes
Sams (6)	Heating of air at high ΔT; square threads inside tubes
Lancet (7)	Heating of air; rectangular channel with machined roughness
Dipprey, Sabersky (8)	Heating of water; sand-grain-type roughness in tubes
Nunner (3)	Heating of air; rings of various cross section inserted inside tubes
Koch (9)	Heating of air; axially supported rings inserted inside tube
Nagaoka, Watanabe (10)	Heating of water; wire coils inside tubes
Sams (11)	Heating of air; wire coils inside tubes
Kreith, Margolis (12)	Heating of water and air; tubes with wire-coil inserts
Edwards, Sheriff (13)	Heating of air; rectangular channel, one surface heated and lined with wire
Kemeny, Cyphers (14)	Heating of water; annulus, inner heated surface with spiral grooves or protrusions
Brauer (15)	Heating of water; annulus; inner surface heated
Bennett, Kearsey (16)	Heating of superheated steam; annulus, inner tube heated and grooved
Draycott, Lawther (17)	Heating of air; annuli and tube bundles with threads, knurls, and coiled wires
Kattchee, Mackewicz (18)	Heating of nitrogen (water for friction tests); wire coils outside rods in a bundle

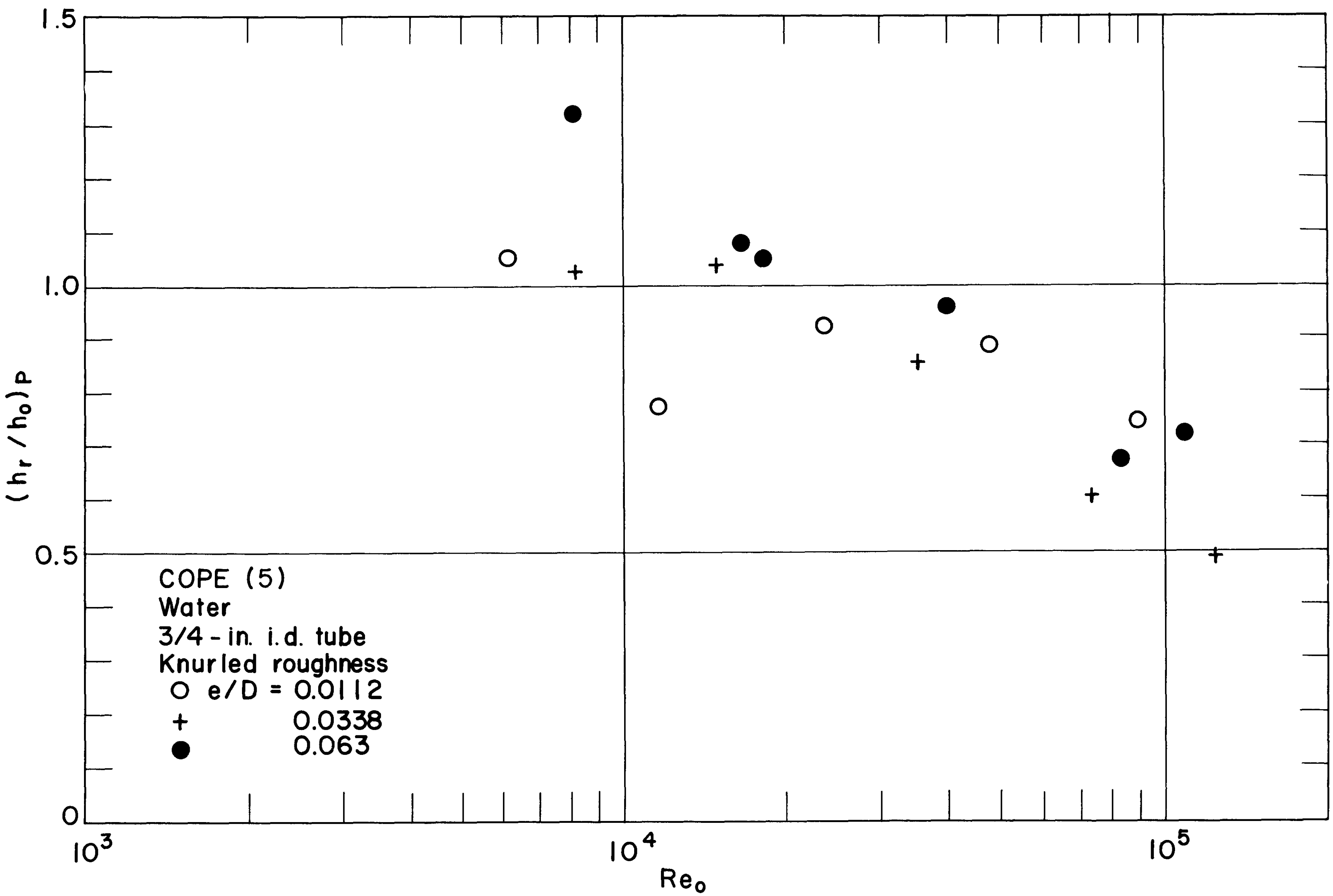


FIG. 3. PERFORMANCE OF TUBES WITH KNURLED ROUGHNESS

Machined roughness was also considered by Sams (6) who heated air at high temperature differences. As indicated in Fig. 4, the performance of this tube decreases with increasing Re, and there is no particular effect of temperature level. There appears to be very little advantage to this type of roughness.

A unique sand-grain-type roughness was produced in tubes and extensively investigated by Dipprey and Sabersky (8). A wide range of Prandtl numbers was covered with the heating of water. Figure 5 indicates that this type of roughness has excellent characteristics with $(h_r/h_o)_P$ approaching 2. The effect of Re concurs with the above results only for the roughest surface. The data for all surfaces indicate a substantial increase in performance as Pr is increased.

Lancet (7) performed tests with a roughened rectangular duct of small hydraulic diameter. The relatively large protrusions ($e/D_e = 0.24$) caused substantial increases in heat transfer and friction. The performance factor based on a hydraulically smooth channel was approximately 1.4 at $Re_o = 15000$; however, the author was unable to obtain a hydraulically smooth surface. With the channel of $D_e = 0.04$ in., even minute polishing scratches caused an appreciable increase in the friction factor.

Nunner (3) presented a thorough study of two-dimensional attached roughness elements with air as the working fluid. The elements were rings of rectangular or round shape which were inserted at various spacings in the test section. Comparative data for these artificial roughnesses are shown in Fig. 6. There is again a distinct decrease in $(h_r/h_o)_P$ as Re is increased. It is also apparent that there is an

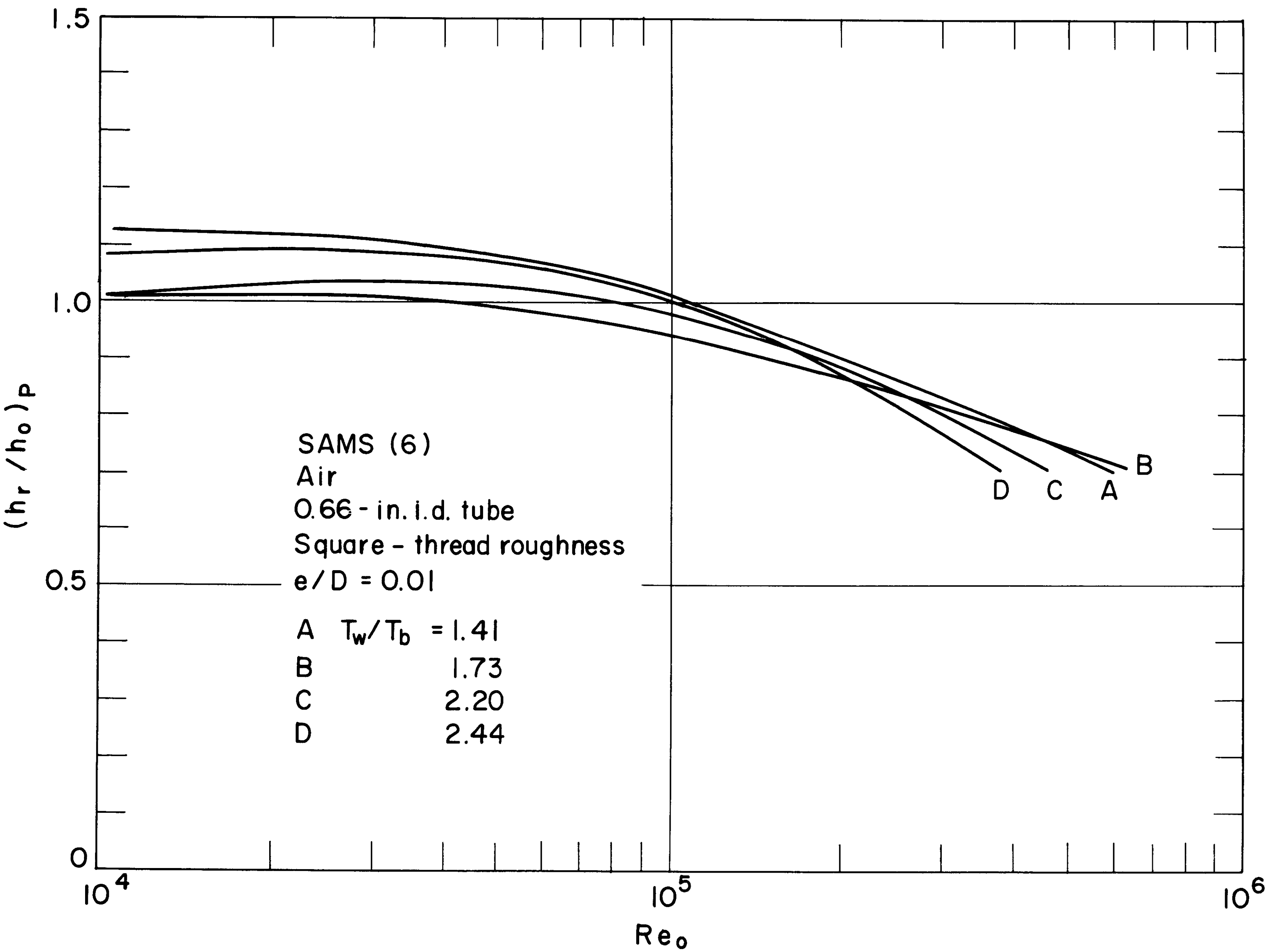


FIG. 4. PERFORMANCE OF TUBES WITH SQUARE-THREAD ROUGHNESS

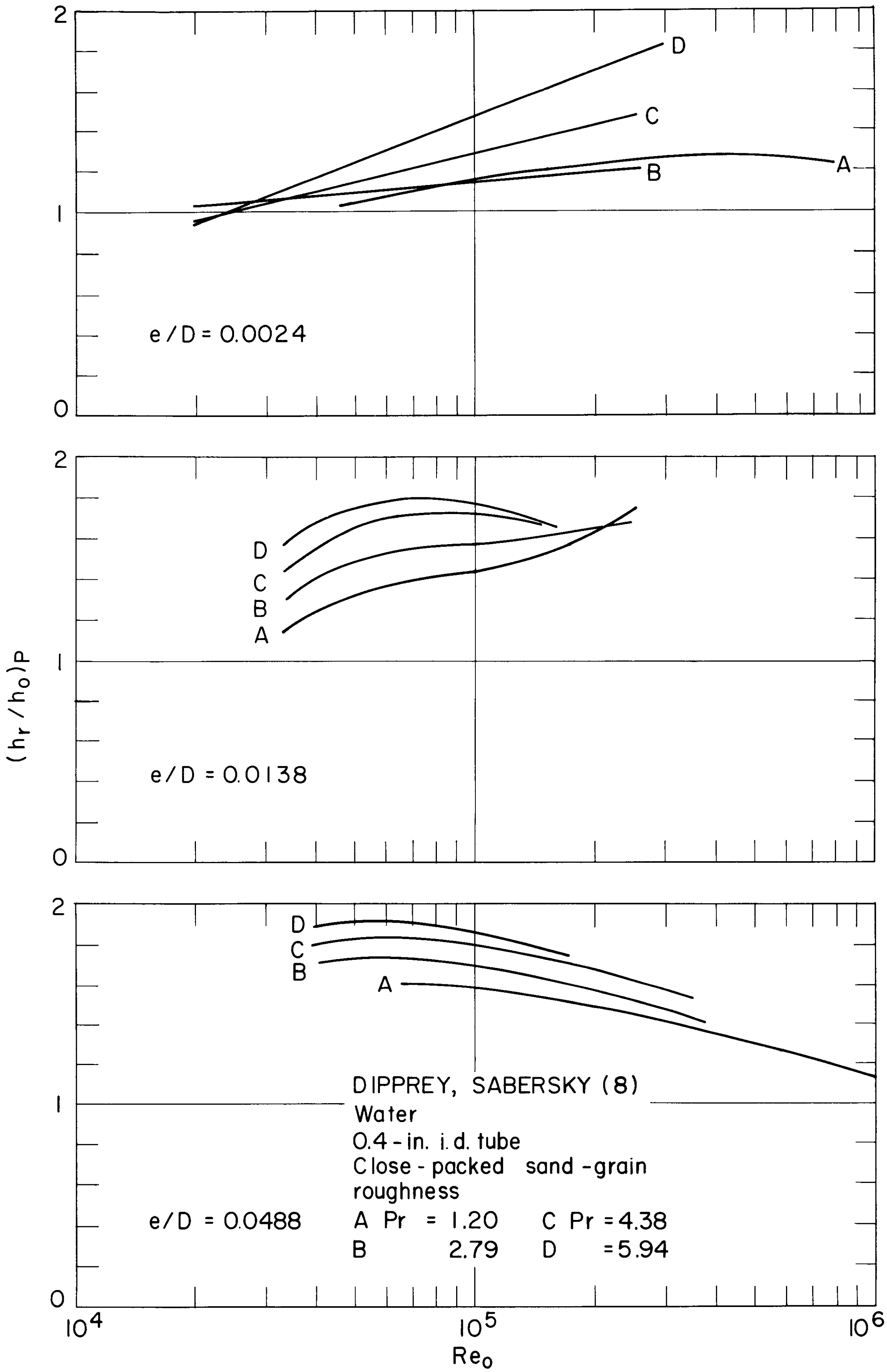


FIG. 5. PERFORMANCE OF TUBES WITH SAND-GRAIN ROUGHNESS

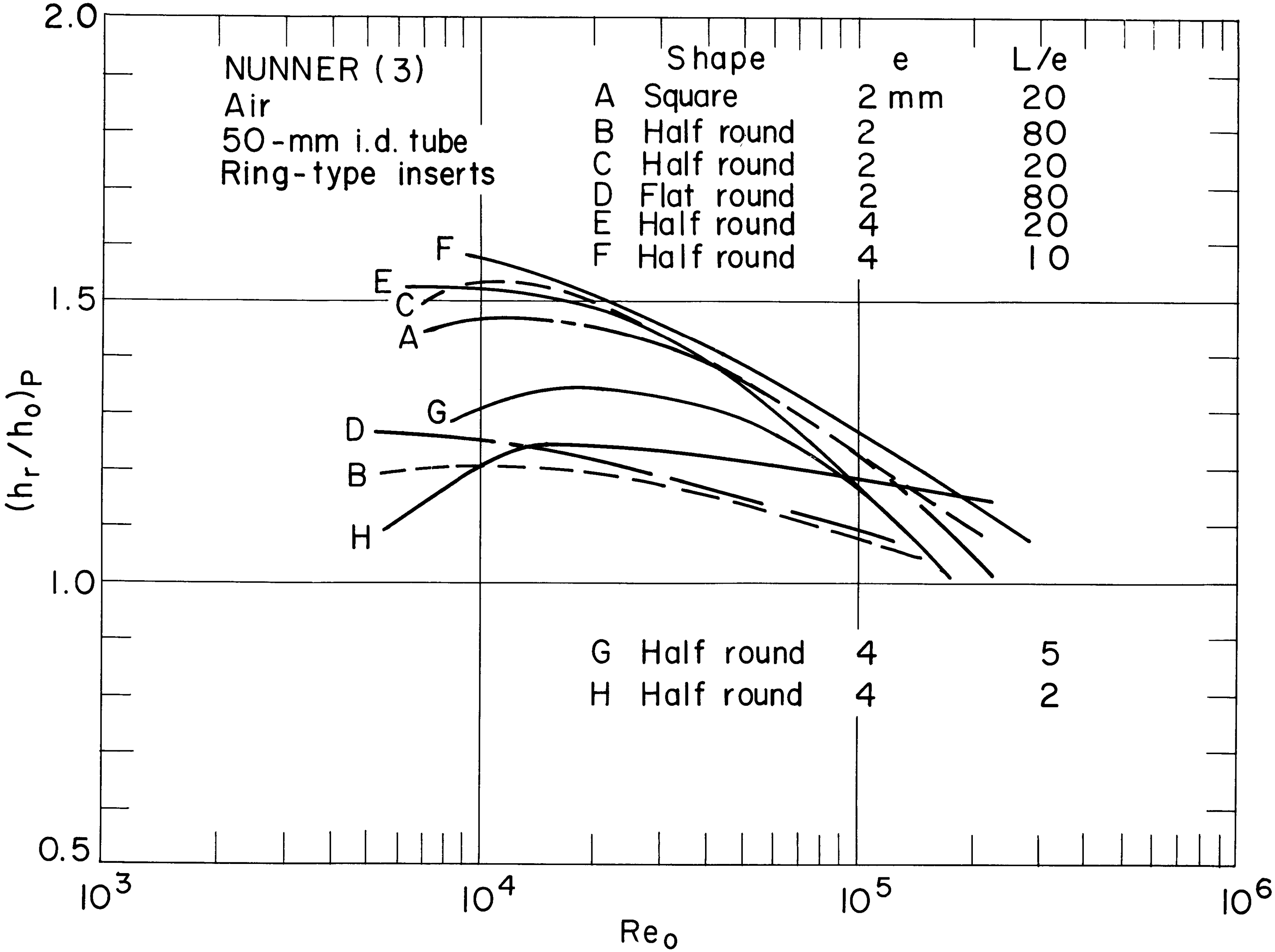


FIG. 6. PERFORMANCE OF TUBES WITH SMALL RING-TYPE INSERTS

optimum spacing-to-thickness ratio for the several shapes considered. For L/e = 10-20, an impressive performance factor of over 1.5 is noted.

Koch (9) extended Nunner's work with a similar apparatus. His aperture-type inserts occupied a substantial portion of the tube cross section, and it was necessary to use thin wire supports. It is evident from Fig. 7 that these promoters are not particularly efficient, probably due to the large values of e. It is to be noted, however, that any fin-effect would be small due to the loose fitting assembly. It can be noticed that an optimum L/e ratio of about 10 also exists for these inserts.

Several investigations of coiled wire promotors have been reported. These attached roughness elements are particularly easy to fabricate since they are simply helical springs inserted tightly into tubes. There is certainly some spiral motion induced by these coils; however, it would appear that the primary effect would be that of surface roughness.

Representative performance data of Nagaoka and Watanabe (10), Sams (11), and Kreith and Margolis (12) are presented in Fig. 8. In spite of the rather well-defined geometry there is a substantial spread in the data. The tightness of the coil fit could explain some of this; however, coils of circular cross section are rather poor fins due to the small contact area. The data of Kreith and Margolis are somewhat suspect due to the increases in performance at higher Re. One would expect that the roughness elements would have less effect as the turbulence becomes fully developed.

Edwards and Sheriff (13) investigated increases in h and f in the vicinity of single wires placed in a rectangular channel. It was concluded

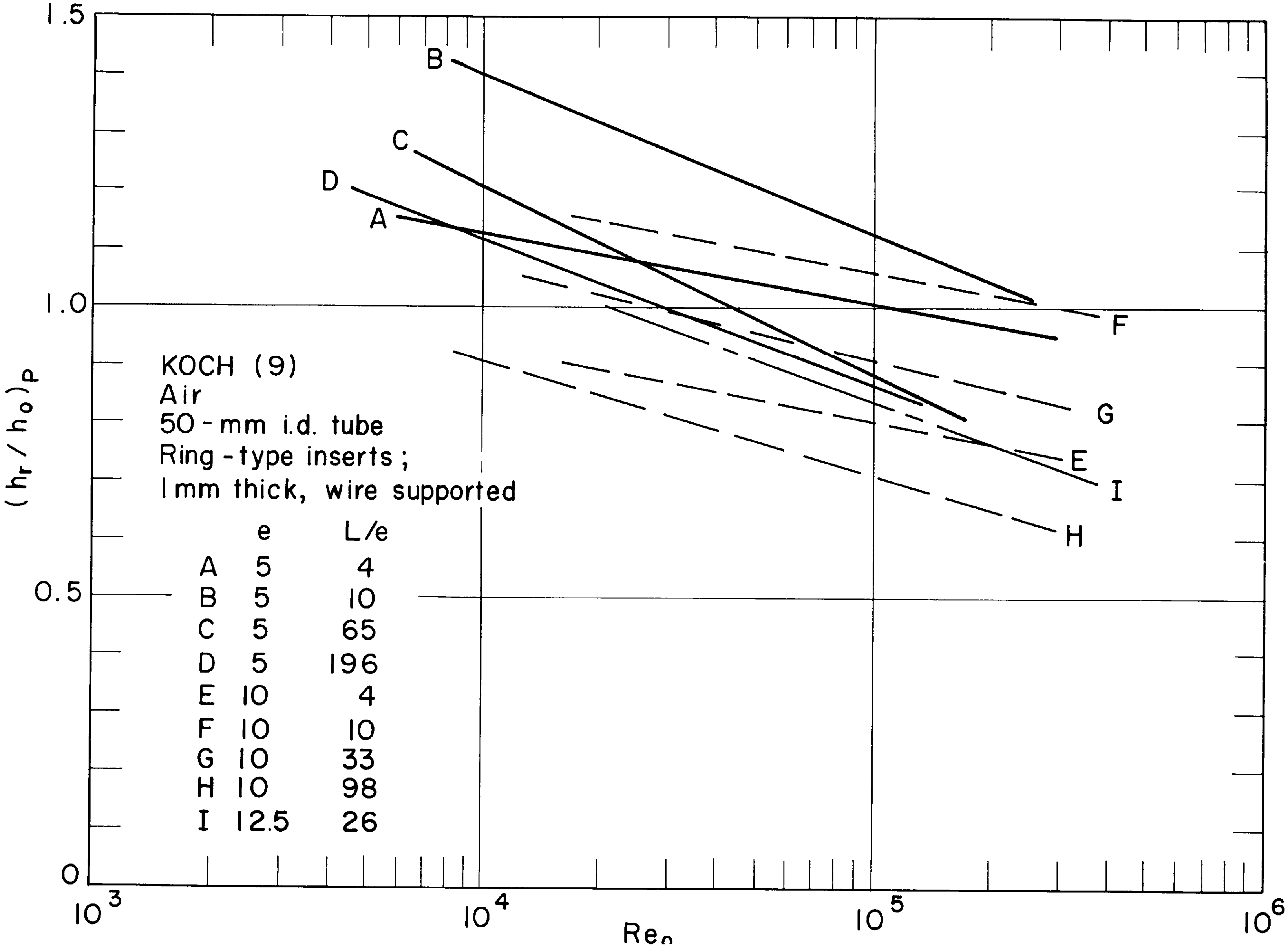


FIG. 7. PERFORMANCE OF TUBES WITH LARGE RING-TYPE INSERTS

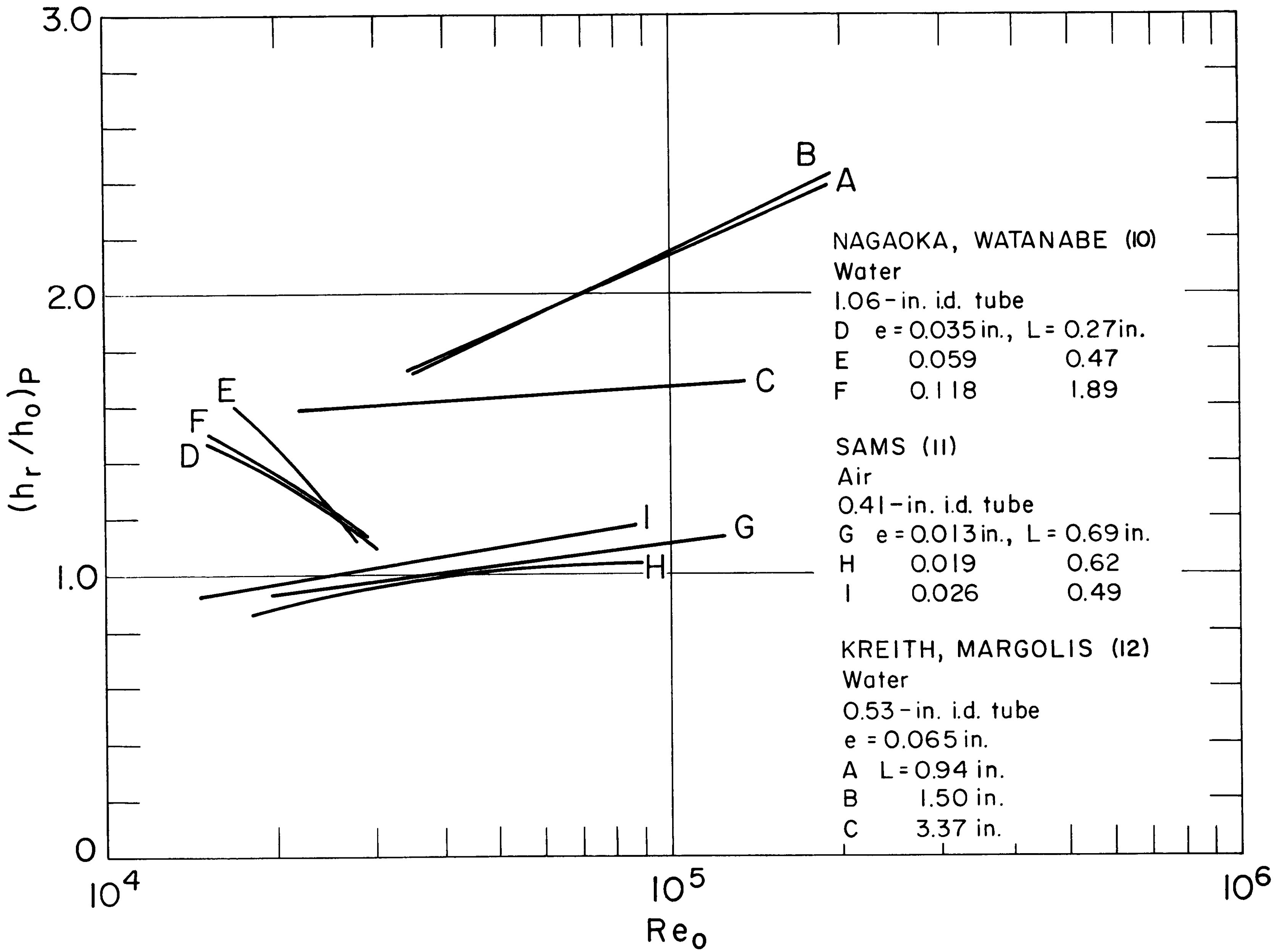


FIG. 8. PERFORMANCE OF TUBES WITH WIRE-COIL ROUGHNESS ELEMENTS

that the wire height must exceed the laminar sublayer thickness before it becomes effective. It appears that more studies of this type will be needed to establish the optimum coil parameters.

The frequently used annular geometry presents a different situation for the application of surface roughness. Machined surfaces are relatively easy to produce, and increased friction affects only a portion of the wetted surface. The results of Kemeny and Cyphers (14) for a helical grove and a helical protuberance are given in Fig. 9. The grooved surface is not effective in general, although there is a tendency to improve with increasing Re. Apparently a relatively shallow groove has little effect until the free-stream turbulence penetrates into the groove. The protruding roughness is seen to be very effective at lower Re. The inferior performance of the coiled wire assembly compared to the integral protrusion is probably due to poor contact between the wire and the groove.

The recent results of Bennett and Kearsey (16) for superheated steam flowing in an annulus are included in Fig. 9. The comparison was based on actual friction data. These investigators were unable to achieve smooth-tube performance due to machining marks and support structures.

The data of Brauer (15) for a similar system illustrate the importance of protrusion spacing. As shown in Fig. 10, the optimum L/e for the annular geometry appears to be about three, which is lower than the apparent optimum for tubes.

An extensive investigation of rough surfaces in complex geometries is summarized by Draycott and Lawther (17). Annuli were used to survey the friction and heat-transfer characteristics of twenty-one machined

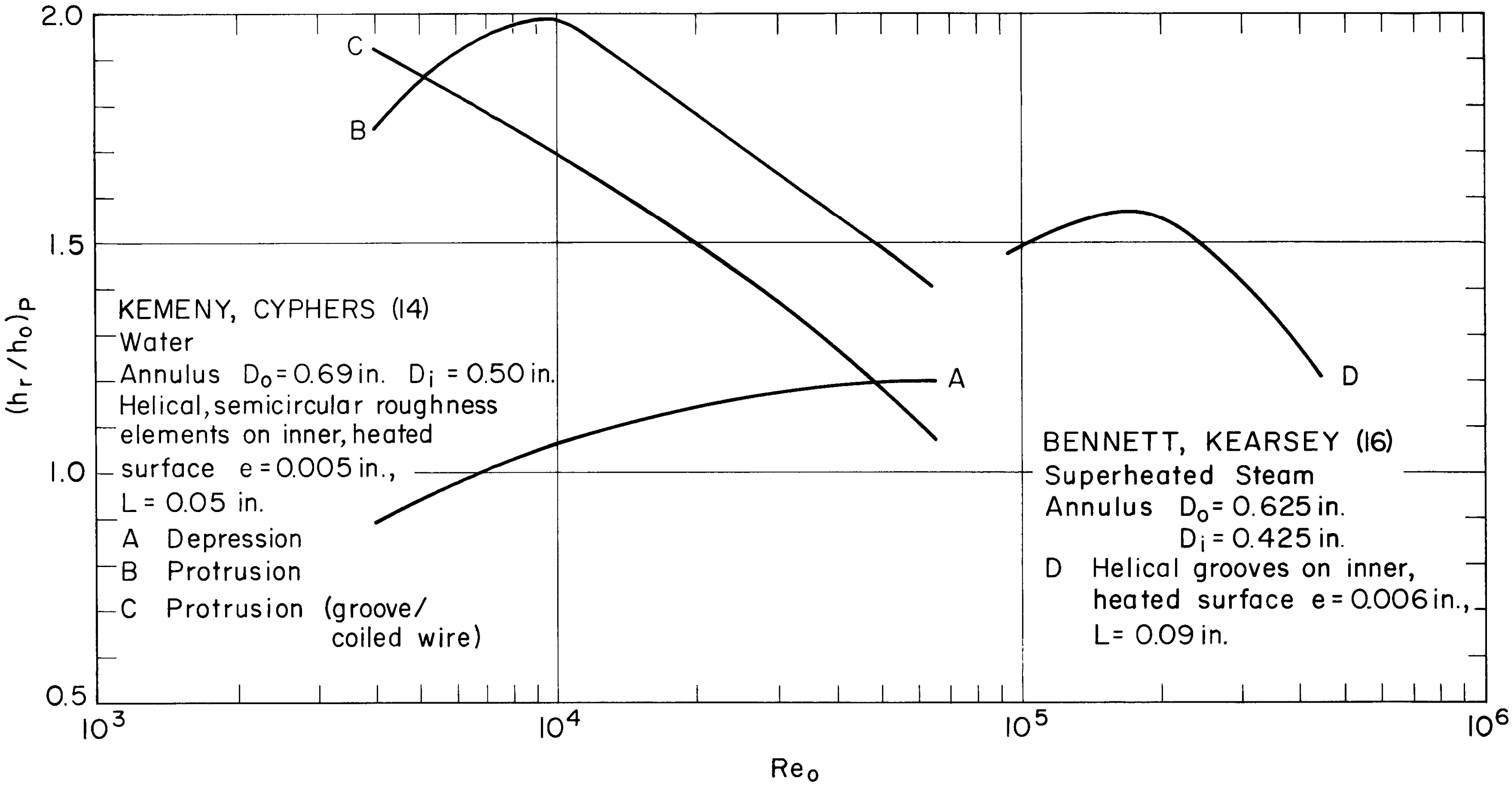


FIG. 9. PERFORMANCE OF ANNULI WITH HEATED-SURFACE ROUGHNESS ELEMENTS

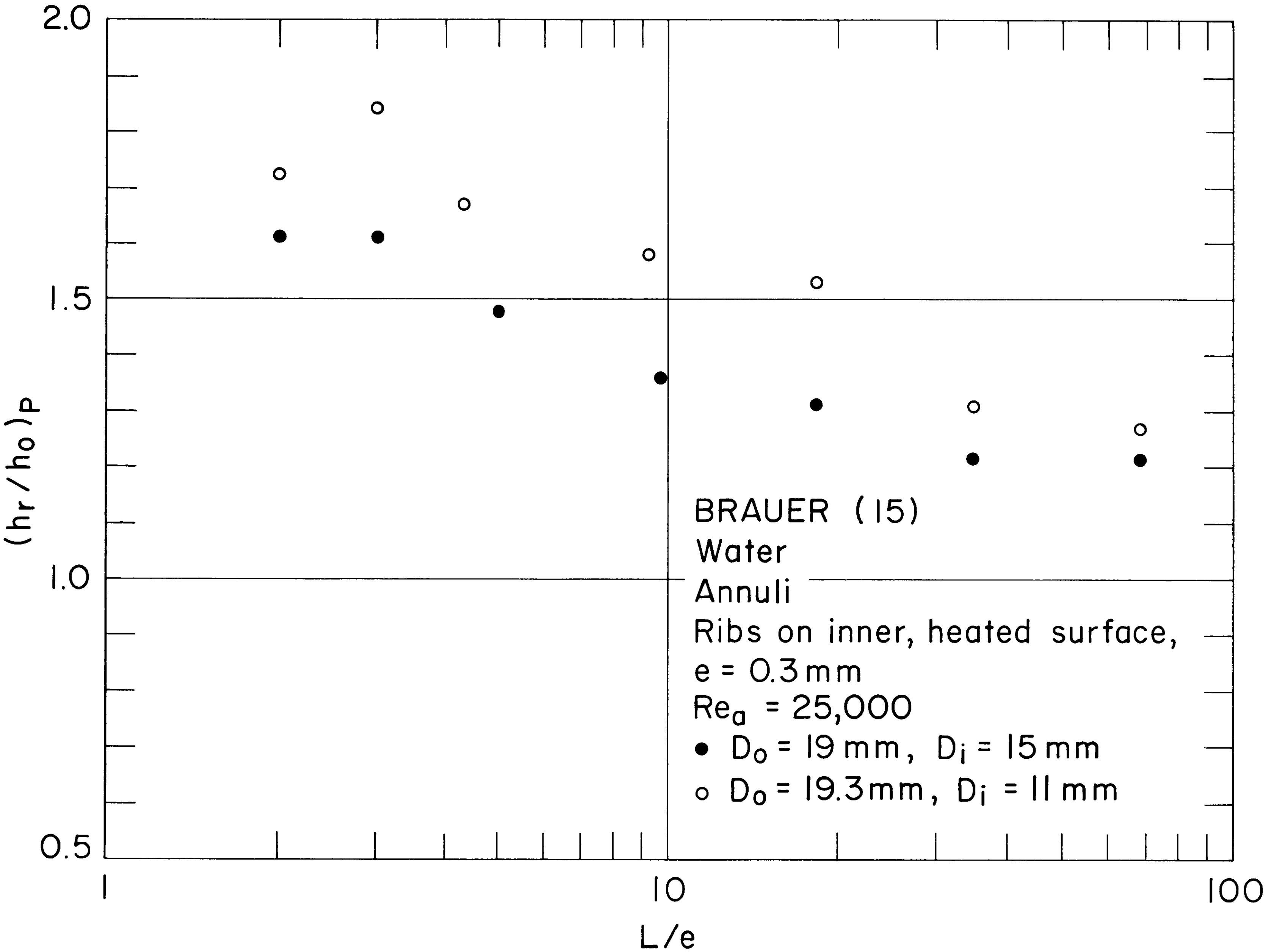


FIG. 10. INFLUENCE OF PROTRUSION SPACING ON ROUGH ANNULUS PERFORMANCE

and wire-wound heater elements. Certain of these surfaces were selected for use in a 7-rod cluster. Some of the surfaces were apparently quite favorable from a performance standpoint; however, the data given in this summary are not adequate for a proper evaluation.

In a recent study, Kattchee and Mackewicz (18) wrapped small-diameter wire around each of nineteen tubes comprising a cluster. Nitrogen was used for heat-transfer measurements, and water was utilized in obtaining friction data. All twelve combinations of diameter and pitch gave a favorable improvement in heat transfer as shown in Fig. 11. It is interesting to note that there is an apparent optimum L/e of about twenty which corresponds to Nunner's results for flow inside tubes.

2.1.2 Analogy Results

Several attempts have been made to relate heat-transfer coefficients to friction factors by means of the analogy between heat and momentum transfer. A general analogy solution for rough surfaces would be highly desirable since it would eliminate the need to perform time-consuming heat-transfer experiments for the many types of roughness.

Pinkel (19) found that air data of Sams (6) for square-thread roughness could be correlated by

$$Nu_f = 0.023\ Re_f^{0.8}\ Pr_f^{0.4}/\alpha\ , \tag{2}$$

where α is an involved function of Re_f, Pr_f, f_f, and y_e/D, a factor representing the effective roughness. Evaluation of pertinent quantities at the film temperature was found to compensate for a wide range of T_w/T_b. The roughness factor is a function of the several dimensions that characterize the roughness; however, it must be established

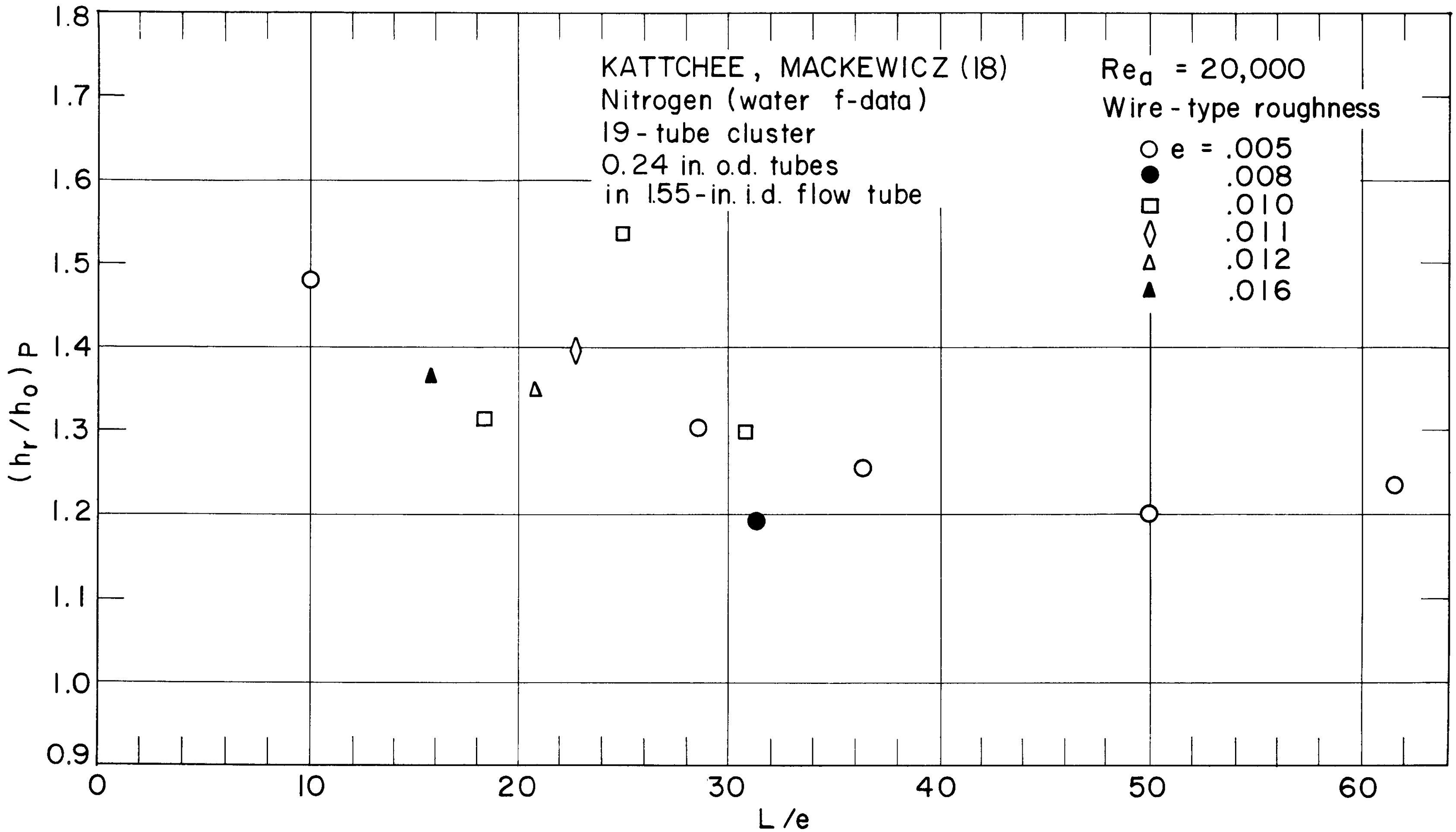


FIG. 11. INFLUENCE OF PROTRUSION HEIGHT AND SPACING ON TUBE BUNDLE PERFORMANCE

empirically from friction data. Furthermore, it is not clear that the same roughness parameter applies to both heat transfer and friction.

Nunner (3) proposed a two-zone analogy where the roughness was postulated to produce a form drag, or shear-stress discontinuity, at the junction of the laminar sublayer and turbulent core. The final equation,

$$St = \frac{f/2}{1 + 1.5\ Re^{-1/8} Pr^{-1/6} \left[(f/f_o)_{Re} - 1 \right]} , \qquad (3)$$

implies that there is a unique relation between heat transfer and friction which is independent of the type of roughness. Nunner's data for ring inserts and $Pr \simeq 0.7$ were well correlated by this equation; however, data for other types of roughness and higher Pr did not agree.

More recently, Dipprey and Sabersky (8) presented a similar but more general analogy which can be expressed as

$$\frac{(f/2\ St) - 1}{(f/2)^{1/2}} = f\,(\epsilon^*, Pr) - f'(\epsilon^*) . \qquad (4)$$

The functions f and f' were considered to be the same for each type of geometrically similar roughness. They must, however, be obtained from both friction and heat-transfer data.

It has been well established, then, that there is no unique relation between heat transfer and friction for rough surfaces. The analogy solutions are useful only to the extent that they facilitate extrapolation and interpolation of limited data.

2.2 Boiling

2.2.1 Saturated Pool Boiling

Surface condition has long been recognized as an important variable in boiling heat transfer. Pool boiling in particular has been extensively investigated with regard to the effects of heater material and surface preparation. Since recent studies by Bernath (20), Pinchera (21), and Ivey and Morris (22) thoroughly survey this area, only one set of representative data is included here.

Figure 12 presents data of Berenson (23) for saturated pool boiling of pentane on a copper surface subjected to various finishing operations. It is seen that a substantial decrease in the nucleate-boiling wall superheat can be obtained by moderate roughening of the heated surface. However, the critical heat flux is virtually independent of surface finish. Film-boiling coefficients are also relatively unaffected. Careful measurements by Berenson also indicated that there was a substantial effect of heater material on saturated pool boiling. For the same surface finish, nucleate boiling heat transfer was greater for a copper surface than for a nickel or an inconel surface. Critical heat fluxes and film boiling were not effected. This effect cannot be attributed entirely to the material, however, since the same finishing operation will produce different nucleation sites in different materials.

These results are in general agreement with theory. Nucleate boiling characteristics are dependent on the cavity size distribution, and in general larger cavities require lower superheat to nucleate. The critical heat flux is usually considered to be caused by hydrodynamic limitations. At the critical heat flux there is interference or

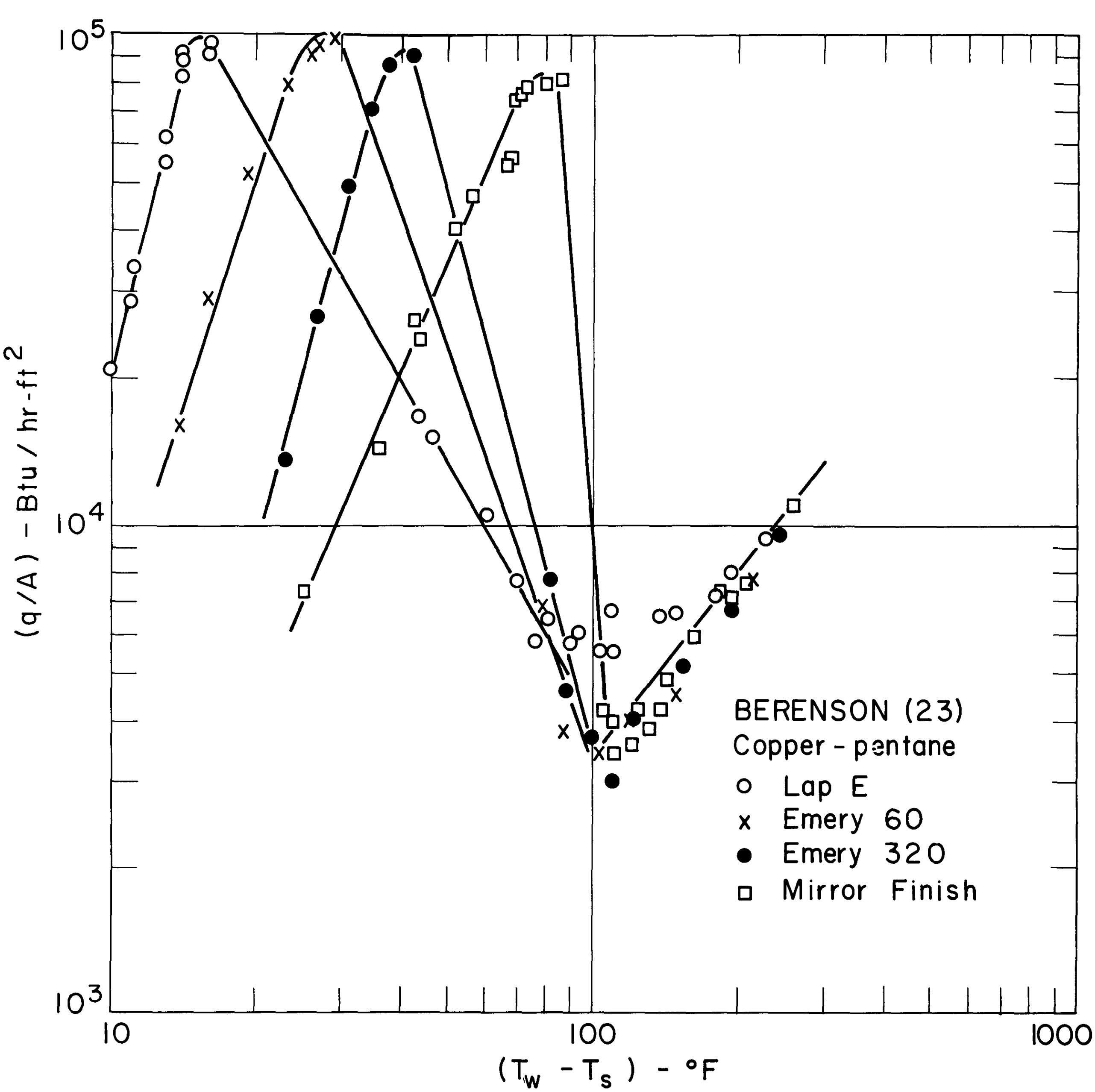


FIG. 12. INFLUENCE OF SURFACE FINISH ON SATURATED POOL BOILING

coalescense of the bubble streams issuing from each cavity resulting in vapor blanketing at the surface. The vapor generation is apparently the same for all types of clean surfaces at the critical heat flux. Since conduction through the vapor film is the mechanism in film boiling, no effect of surface condition should be discernible.

Fouled and oxidized surfaces, as well as aluminum heaters in general, exhibit somewhat higher critical heat fluxes. Attempts have been made to explain this in terms of surface phenomena. However, the effect is not really significant since the increase is only about 10 percent, which is similar to the usual experimental scatter.

A novel technique for promoting pool boiling was recently proposed by Young and Hummel (24). Teflon spots, either on the heated surface or in pits, were found to promote nucleation as shown in Fig. 13. Relatively low superheat was required to activate the nonwetting cavities present at the spots. Since the resulting bubbles were generally large compared to the distance between spots, the area of influence of the bubbles included the whole heated surface, with the net result that the average superheat for the surface was low. The effect on the critical heat flux is still inconclusive as few data have been obtained. Further details of this research were recently presented by these investigators (25).

2.2.2 Forced-Convection Surface Boiling

Forced-convection boiling is also affected by surface condition although systematic studies similar to those in pool boiling are not available. Figure 14 presents surface-boiling data for similar flow conditions and experimental technique, but with nickel and stainless-steel

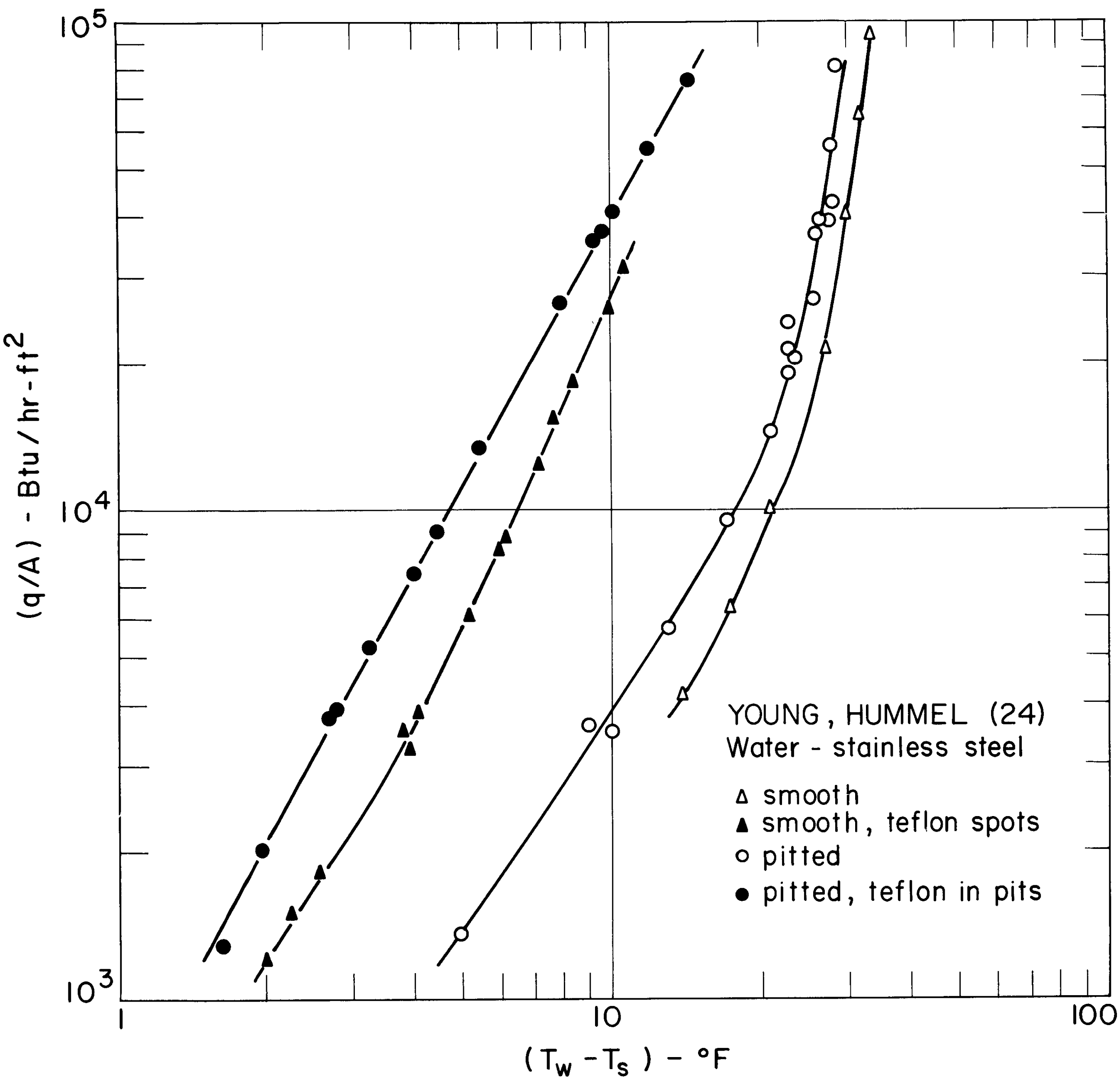


FIG. 13. INFLUENCE OF SURFACE TREATMENT ON SATURATED POOL BOILING

tubes. The fully developed boiling region for each set of data is described by the usual relation

$$q/A = C(T_w - T_s)^n . \tag{5}$$

The constant and to some degree the exponent are different for the two test sections. The more desirable features of the nickel tube are probably due to differences in both material and mechanical treatment of the surface. It is emphasized that both tubes were used in the as-drawn condition and that neither was especially rough.

In contrast to the pool boiling results there is a substantial increase in surface-boiling burnout with machined roughness. Data of Durant, et al. (28) are presented in Fig. 15 as rough-to-smooth burnout-flux ratio versus subcooling. The effects of roughness type and velocity appear to be relatively small for the range considered; however, a definite subcooling trend is evident. No clear explanation for this behavior is evident, although it could be connected with the effect of subcooling on bubble size. At low subcooling the relatively large bubbles could be broken up and prevented from coalescing and blanketing the surface by the additional free stream turbulence caused by the roughness. At high subcooling, the small bubbles could form patches without extending beyond the protuberances. Since no boiling friction data were taken, it is not possible to compare these data on the basis of equal pumping power.

2.2.3 Bulk Boiling

The effects of surface promoters on bulk-boiling heat-transfer coefficients do not appear to have been extensively investigated since the primary object has been to increase critical heat fluxes for boilers.

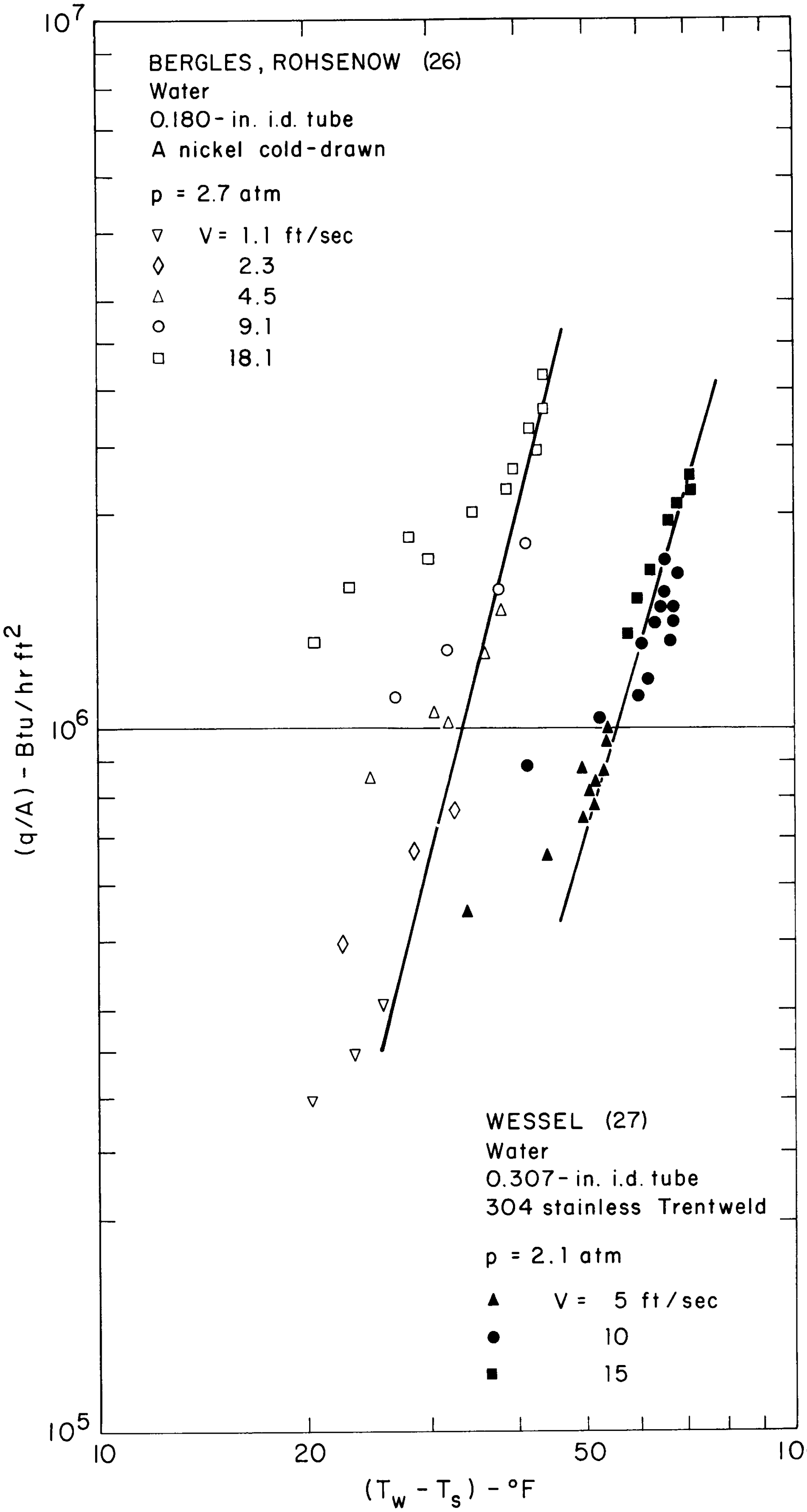


FIG. 14. EFFECT OF SURFACE MATERIAL ON FORCED-CONVECTION SURFACE BOILING

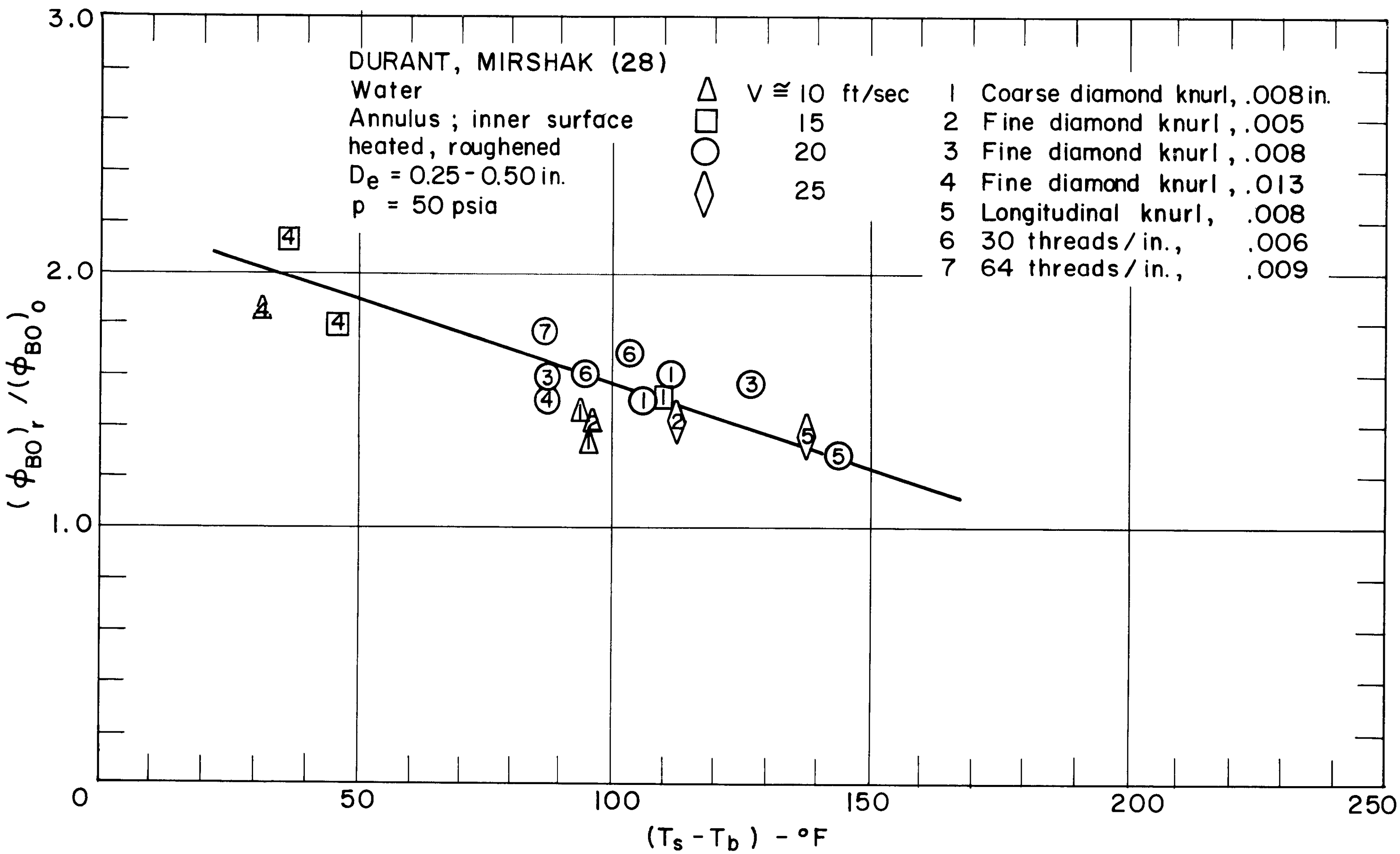


FIG. 15. EFFECT OF SURFACE ROUGHNESS ON SURFACE-BOILING BURNOUT

Two of the configurations investigated by Bernstein, et al (29) belong in this augmentative category. Tests were run with water at constant mass velocity and several heat fluxes. The dual-diameter assemblies, made by joining short sections of different diameter tubing, were found to have made higher heat-transfer coefficients than straight tubes at vapor qualities greater than 50 percent. A slotted helical insert, which was used to simulate internal threads, had higher coefficients at qualities greater than 80 percent. The pressure drop characteristics of these tubes were reasonably good; however, both were difficult to manufacture and thought to be unreliable for high-temperature and high-purity operation. The critical heat fluxes for both these assemblies are considerably higher than those for straight tubes under comparable conditions.

Swenson, Carver, and Szoeke (30) summarized heat-transfer tests with tubes having various machined configurations on the inside surface. Helical ribs were found to be the most effective since they delayed the transition to film boiling until very high qualities were reached.

Additional data are available for bulk-boiling burnout. Janssen, Levy, and Kervinen (31, 32) performed tests with an annular test section where the inner, heated rod was sandblasted ($\sim 7.6\,\mu$). Although these investigators concluded that the surface roughening had an adverse effect on burnout, an examination of their tabulated data shows that this is not the case. Figure 16 shows that the burnout flux with the rough surface is relatively unaffected; if anything, it is increased slightly at the higher flow rate.

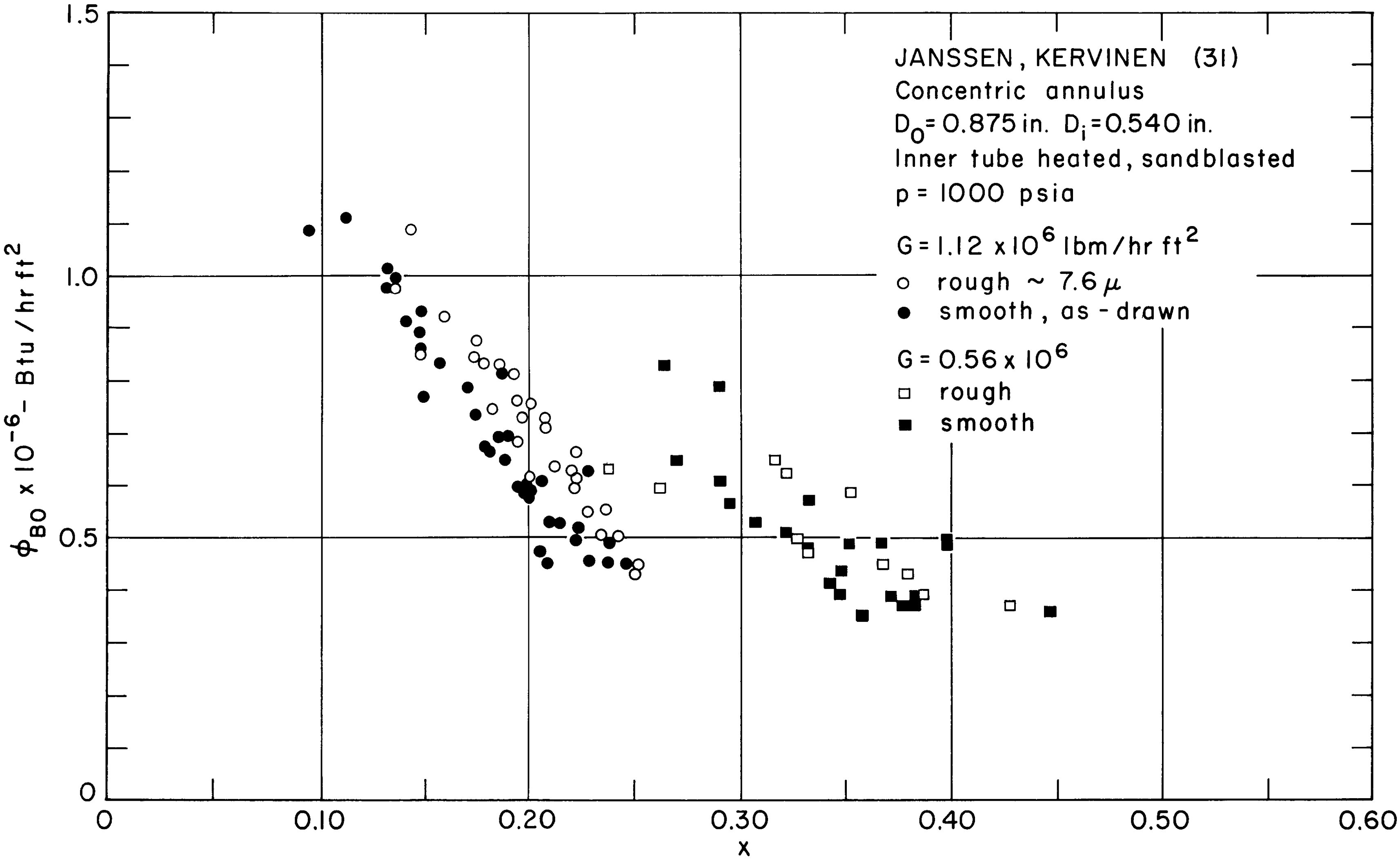


FIG. 16. EFFECT OF SURFACE ROUGHNESS ON BULK-BOILING BURNOUT

These results are in substantial agreement with SNECMA data (33) for rod-bundle burnout tests. Over a wide range of flow rates, the burnout flux for shot-blasted inconel tubes of 5 μ roughness was about 40 percent higher than those for polished tubes of 0.2 μ roughness.

Machined roughness has been considered in an investigation at GEAPD. Quinn (34) reported preliminary results to the effect that machined protuberances, $e = 0.0025$ in. and $L = 0.050$ in., on the heated tube of an annulus increased both critical heat fluxes and film boiling heat-transfer coefficients. A subsequent report (35) indicates that the critical flux is increased only in a certain range of velocity and quality. Wall temperature oscillations in the critical region are generally reduced for the finned surfaces. Improvements in critical heat flux and temperature stability appear to be a strong function of fin size.

In any case, the effect of roughness on quality burnout will depend largely on the flow regime; for example, roughness would be expected to influence the liquid film in annular flow.

2.3 Condensation

Surface treatment is particularly well known in connection with condensation. Dropwise condensation yields heat-transfer coefficients five to ten times as high as film condensation. However, dropwise condensation occurs only when the surface is treated with a suitable promoter that prevents the condensate from wetting the surface. For practical applications the promoters must be reasonably permanent.

The more important promoters are discussed at some length in the standard reference work of McAdams (36).

2.4 Extended Surfaces

It is appropriate at this point to comment briefly on fins as an augmentative technique. Certain of the surface roughnesses, as well as many of the twisted-tape assemblies discussed later, depend to some extent on the fin-effect for their improvement in heat transfer.

In the present study, heat-transfer coefficients, heat fluxes, etc., have been evaluated using the base area with no allowances for protuberances or depressions. There appears to be no justification for separating out the fin-effect unless one wishes to investigate the mechanism of a particular augmentative scheme.

In general it is desirable to take advantage of the fin-effect whenever possible. For example, good contact between attached-type roughness elements and the heated surface would be preferred. Thus the wire-coil inserts should be of semi-circular instead of round cross section and should fit tightly into the channel.

The use of extended surfaces is a well-established and much-used augmentative technique. There is little point in discussing this subject in detail here since it has been treated extensively in standard works, such as McAdams (36), Knudsen and Katz (37), Kays and London (38), and Kraus (39). In general, the use of extended surfaces will affect the entire design of a heat exchanger due to structural considerations. It would thus be somewhat misleading to compare, for example, a ruffled-fin surface with a simple circular-tube surface without considering the application and the fabrication. For the present work it is desirable, however, to compare the performance of simple finned systems with the other augmentative techniques. In spite of the large repertory of analytical solutions, a proper evaluation of most finned systems can be made only by taking actual friction and heat-transfer data.

Circular tubes with rather complex internal fins are being produced commercially (e.g. (40)) . Design data for a wide variety of geometries were presented recently by Hilding and Coogan (41). As shown in Fig. 17 several of the arrangements have excellent performance factors. On the basis of these tests, it appears that the assemblies with the largest surface areas perform the best. It is expected that an optimum area would exist, however. The manufacturing complexity and cost of these fins will be an important consideration in the over-all optimization. Similar results have been obtained for transverse and longitudinal fins in annuli. Tubes with external, transverse fins are, of course, extensively used for both natural- and forced-convection cross flow.

Extended surfaces have been found to be quite useful in boiling situations also. Extensive data have been reported for commercial evaporators with horizontal tubes where boiling takes place outside the externally finned tubes. Katz, et al. (42), for example, indicate that nucleate boiling coefficients for the finned tubes are generally greater than those for plain tubes at low ΔT, although the reverse might be expected to hold at high ΔT due to vapor blockage. Due to the increased area of the finned surface, it has been possible to get at least twice the heat transfer with fins for the same ΔT and length of tube. Internal axial fins were found to improve the heat-transfer coefficient and critical heat flux for bulk boiling at high quality (29). Considerable difficulty was encountered in fabricating these fins in high-pressure boiler tubes.

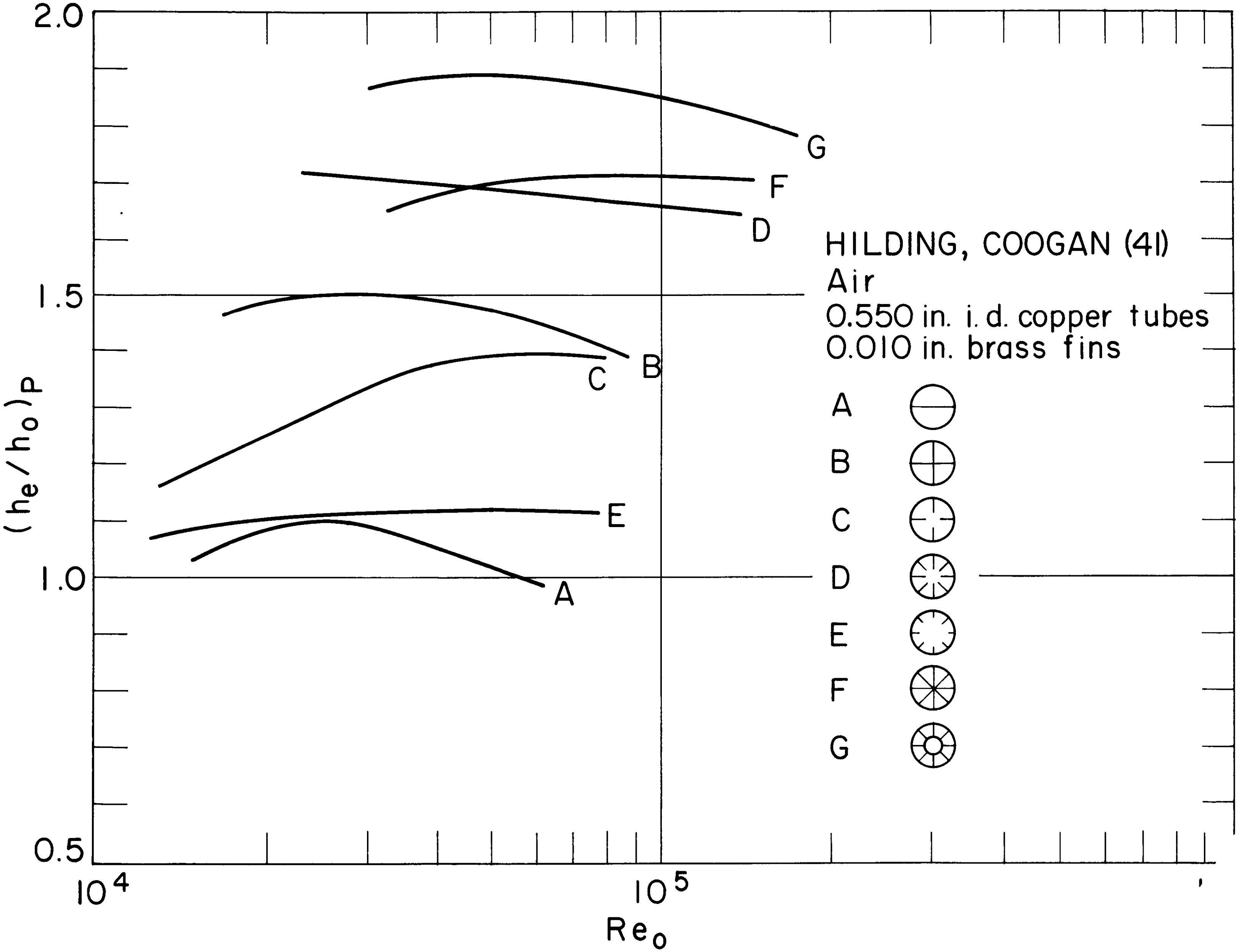


FIG. 17. PERFORMANCE OF TUBES WITH INTERNAL FINS

The finned surfaces, then, are to be regarded as direct competitors of the augmentative schemes discussed at length in this report.

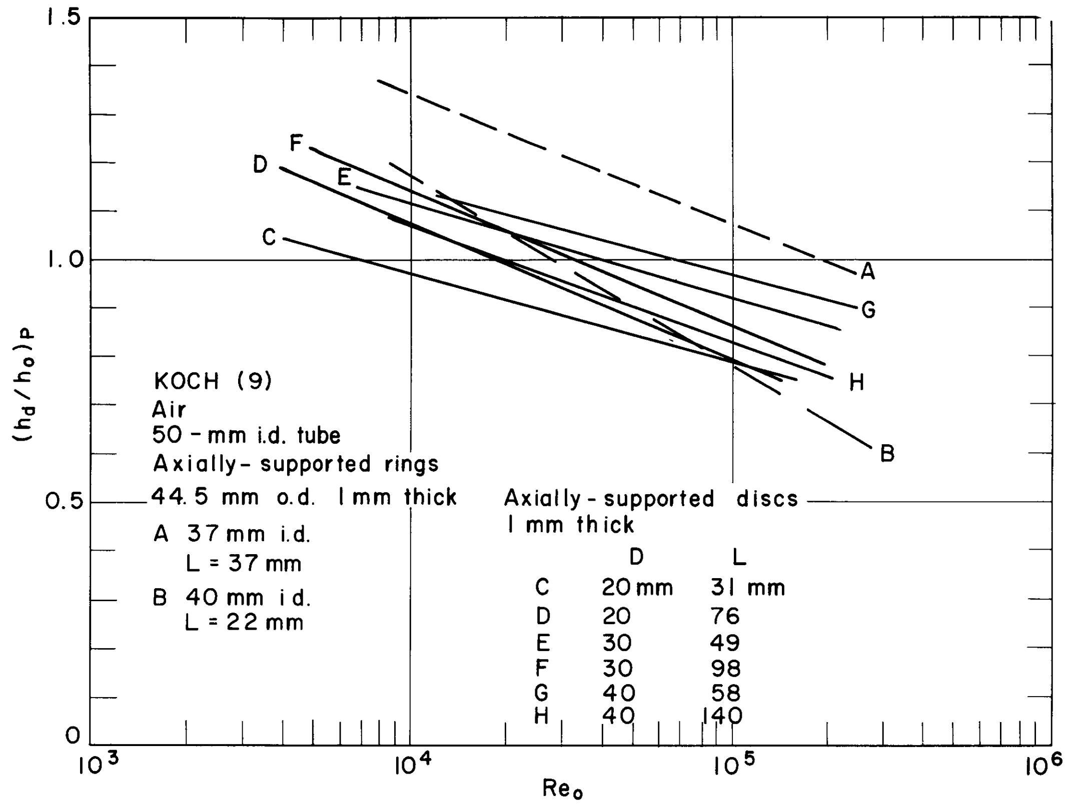
3. DISPLACED PROMOTERS

The turbulence level in a convective system can also be elevated by disturbing the flow near the heated surface. Axially located bluff bodies and streamlined shapes as well as different packing materials have been inserted in tubes. Packing materials, such as Rashig rings, will not be considered here.

3.1 Nonboiling

Axial inserts have been considered as turbulence promoters in two comprehensive studies. Koch, in addition to his investigation of bluff inserts located at the heated surface, considered thin rings and discs located in the bulk flow (9). The evaluation of his results is shown in Fig. 18. Rings are seen to substantially improve heat transfer in the lower Reynolds number range; however, the improvement is quite sensitive to ring dimensions. Discs are less effective, and there is no particular trend of size or spacing within the range of variables tested.

Evans and Churchill (43, 44) also considered axial discs with heat transfer to water in forced convection. As seen in Fig. 18, the results are generally unfavorable. The performance data are somewhat lower than those of Koch; this can probably be attributed to differences in geometry and perhaps in Pr. An optimum disc spacing at L = 4D is discernible, and there appears to be some advantage to the larger diameter discs. Evans also investigated the effect of axially located streamline shapes. As shown in Fig. 19 there appears to be little to recommend such elaborate shapes for turbulence promoters. The data fall rather close together; however, there again appears to be an optimum spacing at L = 4D.



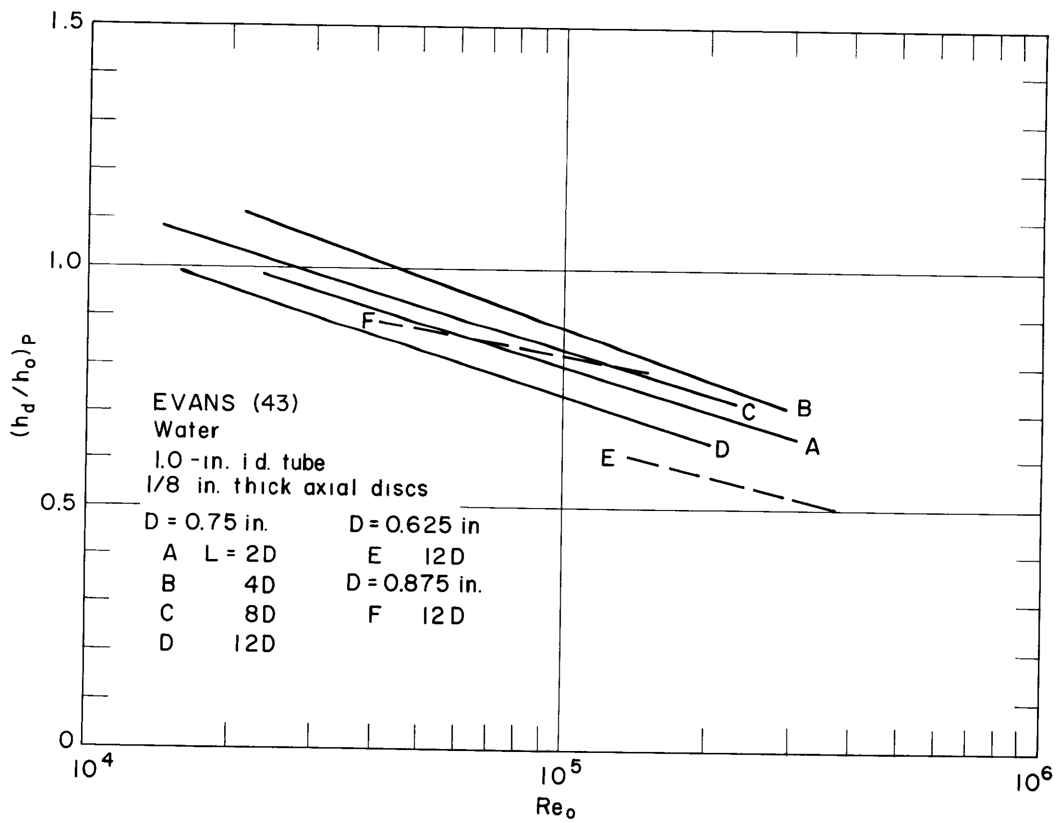


FIG. 18. PERFORMANCE OF TUBES WITH AXIAL DISC-TYPE TURBULENCE PROMOTERS

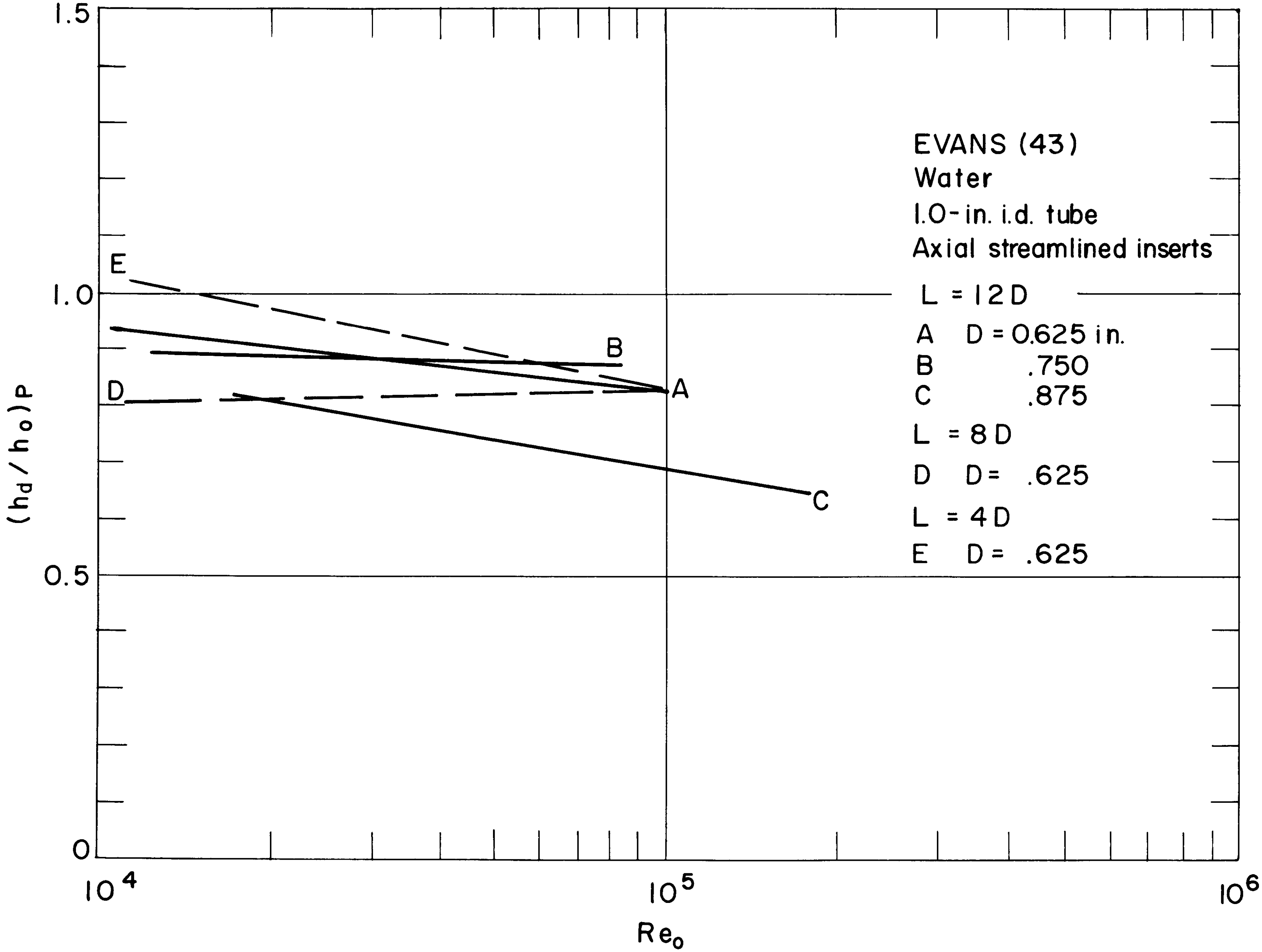


FIG. 19. PERFORMANCE OF STREAMLINED AXIAL TURBULENCE PROMOTERS

3.2 Boiling

Janssen, Levy, and Kervinen (31, 32) reported on bulk boiling burnout with displaced turbulence promoters. Flow-disturbing rings were located on the outer tube of an annular test section. Burnout fluxes for quality boiling with the rough liner are seen in Fig. 20 to be as much as 60 percent greater than those for the smooth liner. It is interesting to note that there is a reversal of the usual flow effect at the higher velocities with the rough liner. These investigators explain both the increased burnout and flow-effect reversal by noting that the roughness elements force the liquid toward the heated surface.

These results were so encouraging that a similar approach has been used in another study at GEAPD. Rings of stainless-steel wire, $e = 0.044$ in. and $L = 1$ in., were spot-welded to the channel wall of a two-rod assembly. As reported by Quinn (45) both critical heat fluxes and film-boiling heat-transfer coefficients were improved.

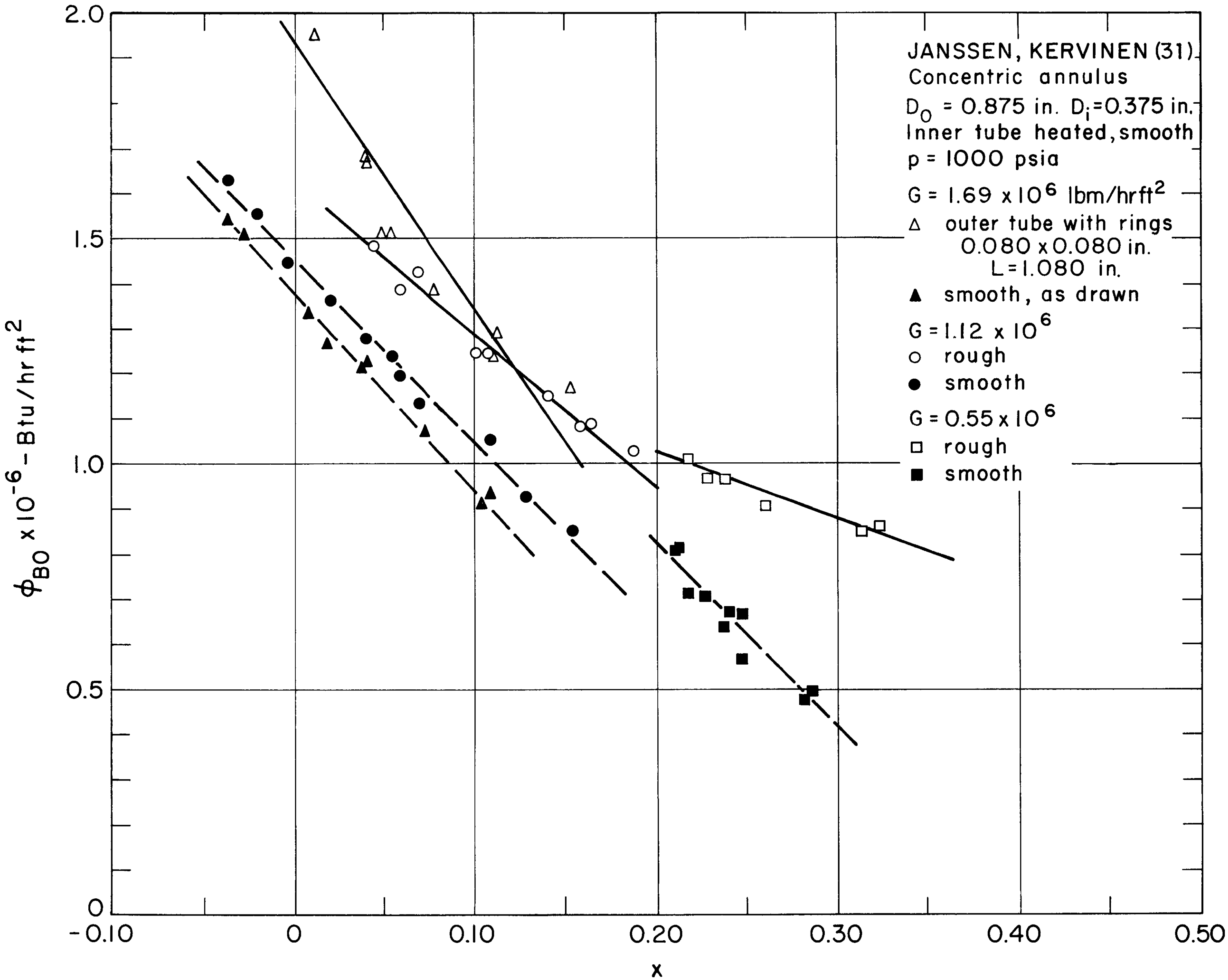


FIG. 20. EFFECT OF TURBULENCE PROMOTERS ON BULK-BOILING BURNOUT

4. VORTEX FLOW

It has been established for over forty years that swirling the flow will improve heat transfer in a forced-convective system; however, it is only in the past decade that extensive investigations of swirl flow have been reported. Generation of swirl flow has been accomplished by coiled wires, propellers, coiled tubes, inlet vortex generators, and twisted tapes. Virtually all of these arrangements have been shown to improve nonboiling and boiling heat transfer at the expense of increased pumping power. Heat-transfer coefficients are relatively high for vortex flow due to the enhanced radial turbulent fluctuations characteristic of flow past a concave surface. The radial body force produced by the swirling flow produces an additional secondary flow when favorable density gradients are present.

The considerable amount of literature on this subject, as well as the wide range of geometries and flow conditions, makes it impossible to present a complete survey. However, a comprehensive survey by Gambill and Bundy (46), which discusses most of the data taken before 1962, is quite adequate in this regard. As in the case of surface roughness and turbulence promoters, then, only representative data will be discussed and evaluated with the same performance criterion.

4.1 Coiled Wires

Coiled wires produce a certain amount of rotation in the flow; however, their primary effect would appear to be an increase of turbulence at the heated surface. Accordingly, the discussion of these augmentative devices has been included in the section on surface roughness.

4.2 Stationary Propellers

Propellers spaced along the flow channel have also been considered by several investigators. The performance of such intermittant-type vortex generators is not particularly outstanding as indicated by an evaluation of available data in Fig. 21.

4.3 Coiled Tubes

Heat transfer is definitely improved when the flow channel is formed into a helix. The correlation usually mentioned was obtained by Jeschke (48) for turbulent flow of air forty years ago, and apparently little has been done to extend the range of variables.

Coiled tubes were recently suggested as a means of improving boiler performance. Carver, Kakarala, and Slotnik (49) reported substantial improvements in bulk-boiling burnout with coils of 16 in. and 65 in. radii.

A variation on the coiled-tube technique was investigated with considerable success at Pratt & Whitney (29). Regular tubing was formed in a wave-shaped or serpentine pattern. In the high-quality region, heat-transfer coefficients were found to be much higher than those for straight tubes at comparable conditions, whereas pressure drop was not greatly increased. This, in effect, reflects a substantial improvement in the critical heat flux. Due to these excellent characteristics, this scheme was chosen for a zero-gravity boiler and tested successfully with bulk boiling of potassium (50).

4.4 Inlet Vortex Generators

In their early study on vortex flow, Gambill and Greene (51) demonstrated that tubes with spiral-ramp and tangential-slot vortex generators could handle extremely large heat fluxes. The now-classic

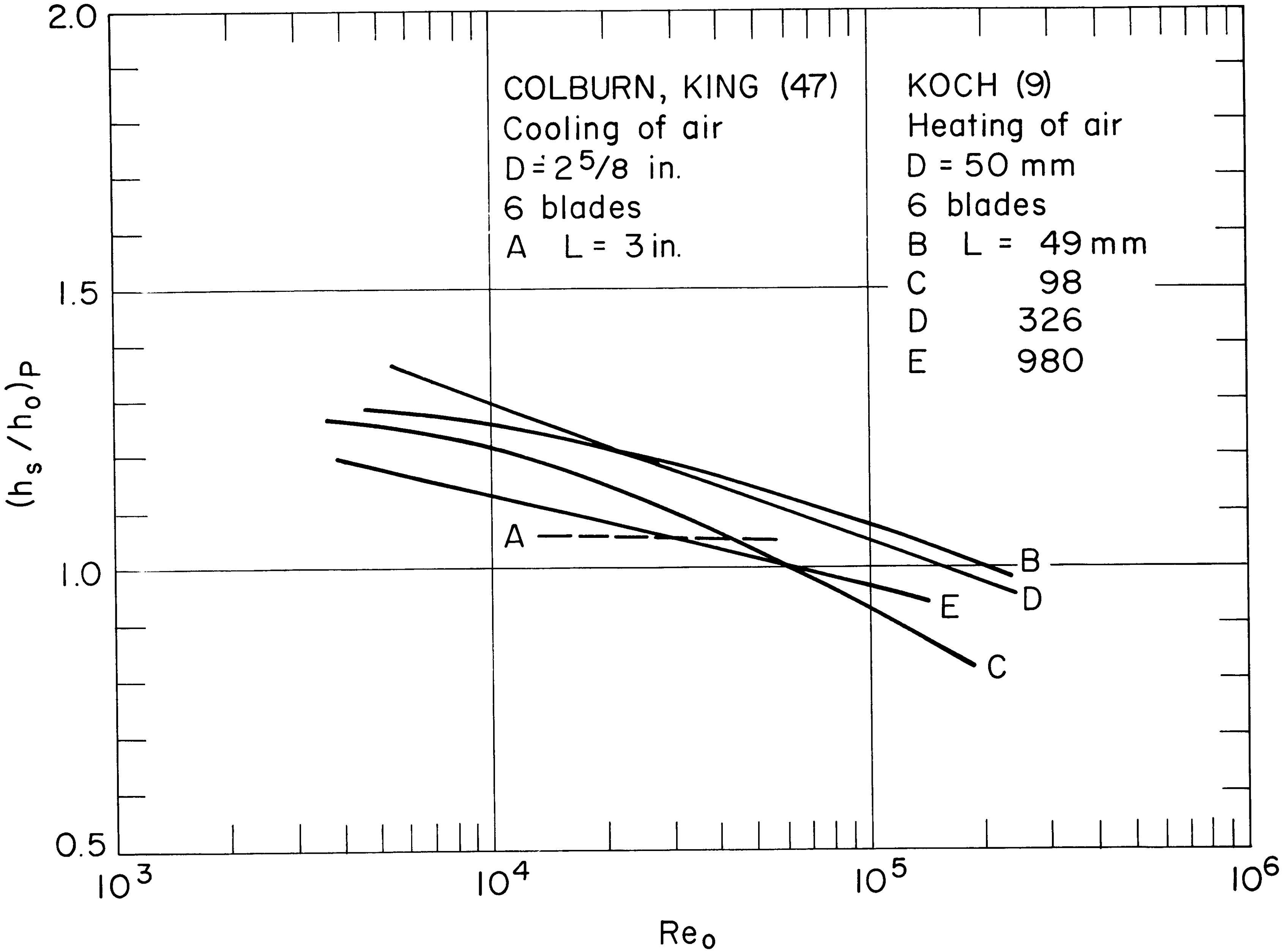


FIG. 21. PERFORMANCE OF PROPELLER-TYPE VORTEX GENERATORS

$(q/A)_{BO} \cong 55 \times 10^6$ Btu/hr ft^2 was obtained using a tangential-slot generator together with a short test section. It was concluded that this arrangement was superior to a smooth tube on the basis of equal pumping power although comparable smooth-tube data were not actually taken.

Since the vortex generator must be located at the inlet to the test section, there is a pronounced effect of heated length on the burnout flux due to the vortex decay. Application of this interesting scheme is, therefore, probably rather limited.

4.5 Twisted Tapes

4.5.1 Nonboiling

Twisted tapes are appropriate for a detailed evaluation since they have been quite extensively investigated, and the geometry is reasonably well defined. Fabrication is generally accomplished by twisting a metallic strip and inserting the uniformly deformed strip into a flow channel. It is mechanically impossible to achieve tight twist ratios with this technique; however, several investigators have obtained extremely tight twists by wrapping the strip around a small-diameter rod.

As noted by Gambill and Bundy (46), there is considerable disagreement among investigators regarding heat-transfer and friction data for twisted-tape assemblies. It is particularly disconcerting to see that both Nu vs Re and f vs. Re data have slopes ranging from highly negative to slightly positive. As a result of this disagreement it is reasonable to expect that the performance curves will be diverse.

Data of numerous studies are compared in Fig. 22 for air and in Fig. 23 for nonboiling water. The fact that the performance of twisted-tape systems is usually favorable with cooling attests to the effectiveness

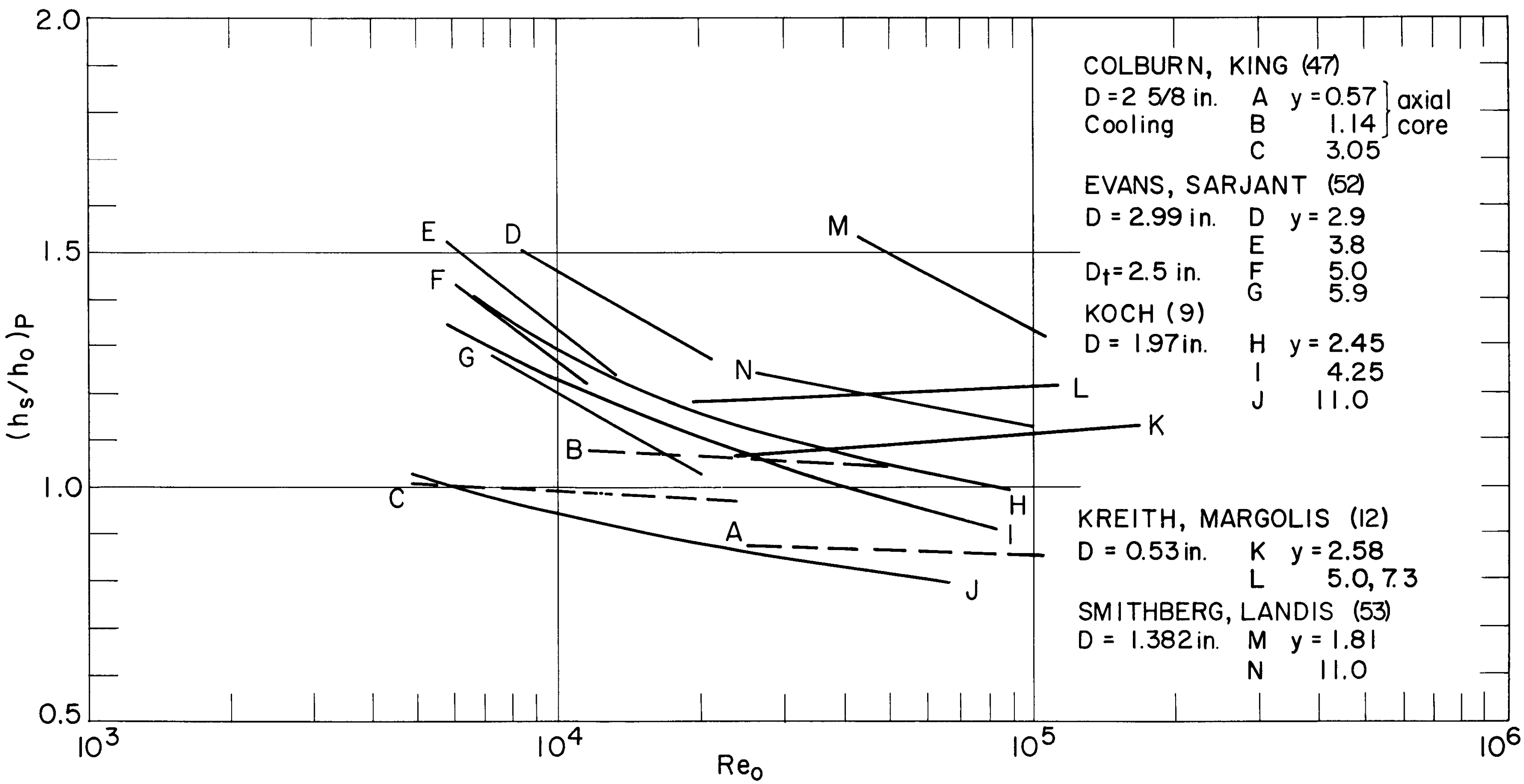


FIG. 22. PERFORMANCE OF TWISTED-TAPE VORTEX GENERATORS WITH AIR

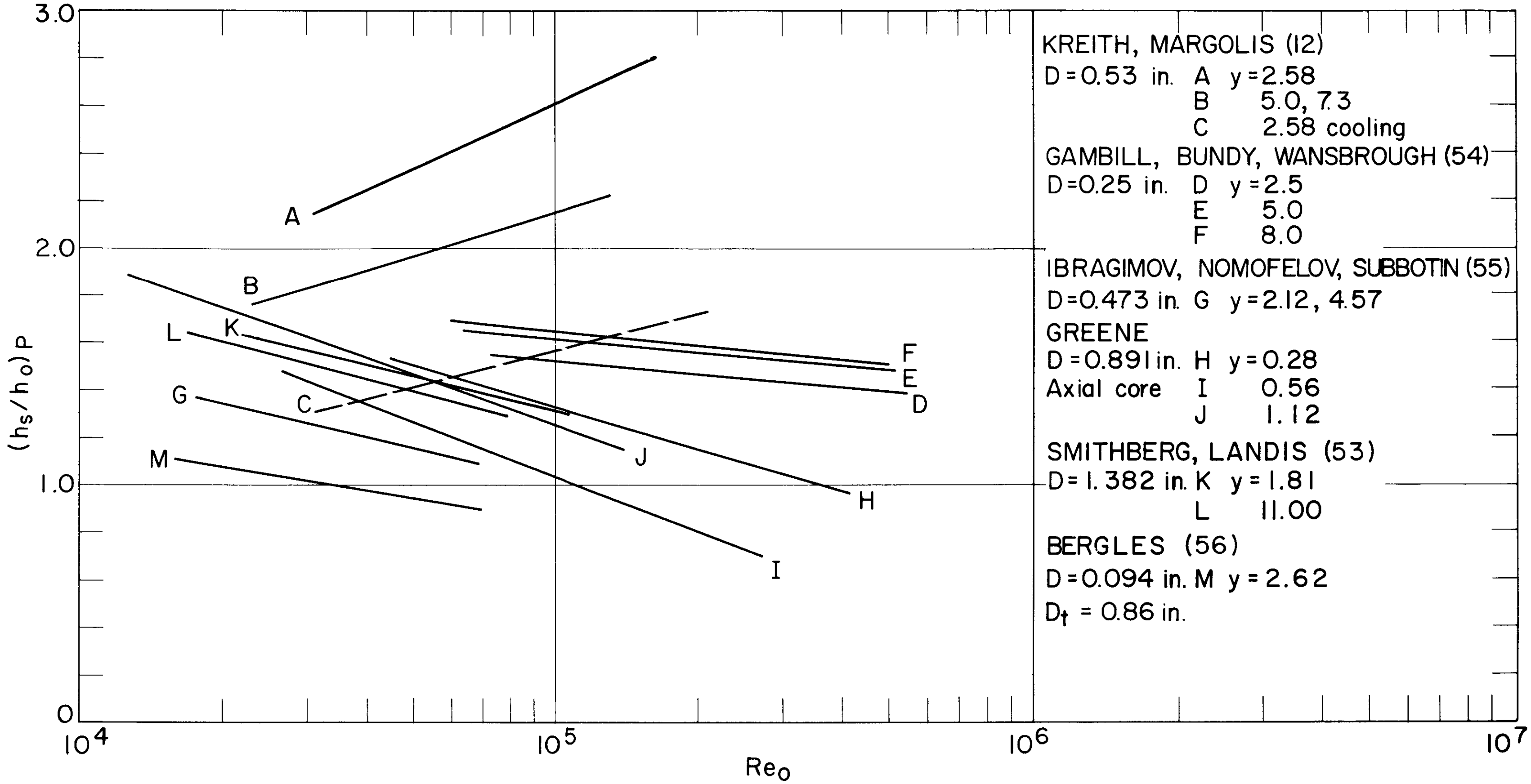


FIG. 23. PERFORMANCE OF TWISTED-TAPE VORTEX GENERATORS WITH NONBOILING WATER

of the basic curved-flow mechanism. It is evident, however, that the greatest benefit is obtained with heated systems. The performance data for air are generally lower than the water data. Since the centrifugal-force Grashof numbers for both fluids are of the same magnitude, there must be an additional mechanism which opposes the buoyant force in the case of air. It is quite probable that the radial pressure gradient increases the density of the more compressible air, thus counteracting the centrifugal free convection (12, 46). There must be additional factors besides the buoyancy considerations, however, since very careful measurements by Gambill, et al. (54) show that the heat-transfer data for water systems are only weakly dependent on the Grashof number.

Tape roughness is certainly an important consideration. Rough tapes increase the hydraulic resistance without any appreciable improvement in the heat-transfer coefficient at the tape surface. In this regard, it is evident that good bonding between the tape and the tube is desirable in order to enhance the fin effect. The results of Smithberg and Landis (53), for example, indicate that improvements in heat transfer of over 25 percent can be attributed to the fin-effect alone. Under certain conditions, it may be desirable to have loose tapes so that they may be removed from the flow channel for cleaning, in which case the fin-effect will be negligible.

Seymour (57) has apparently performed the only study in which the tape twist was systematically varied. He found that the optimum was $y = 2.5$, independent of Re, for air flowing in a 0.87-in. i.d. tube.

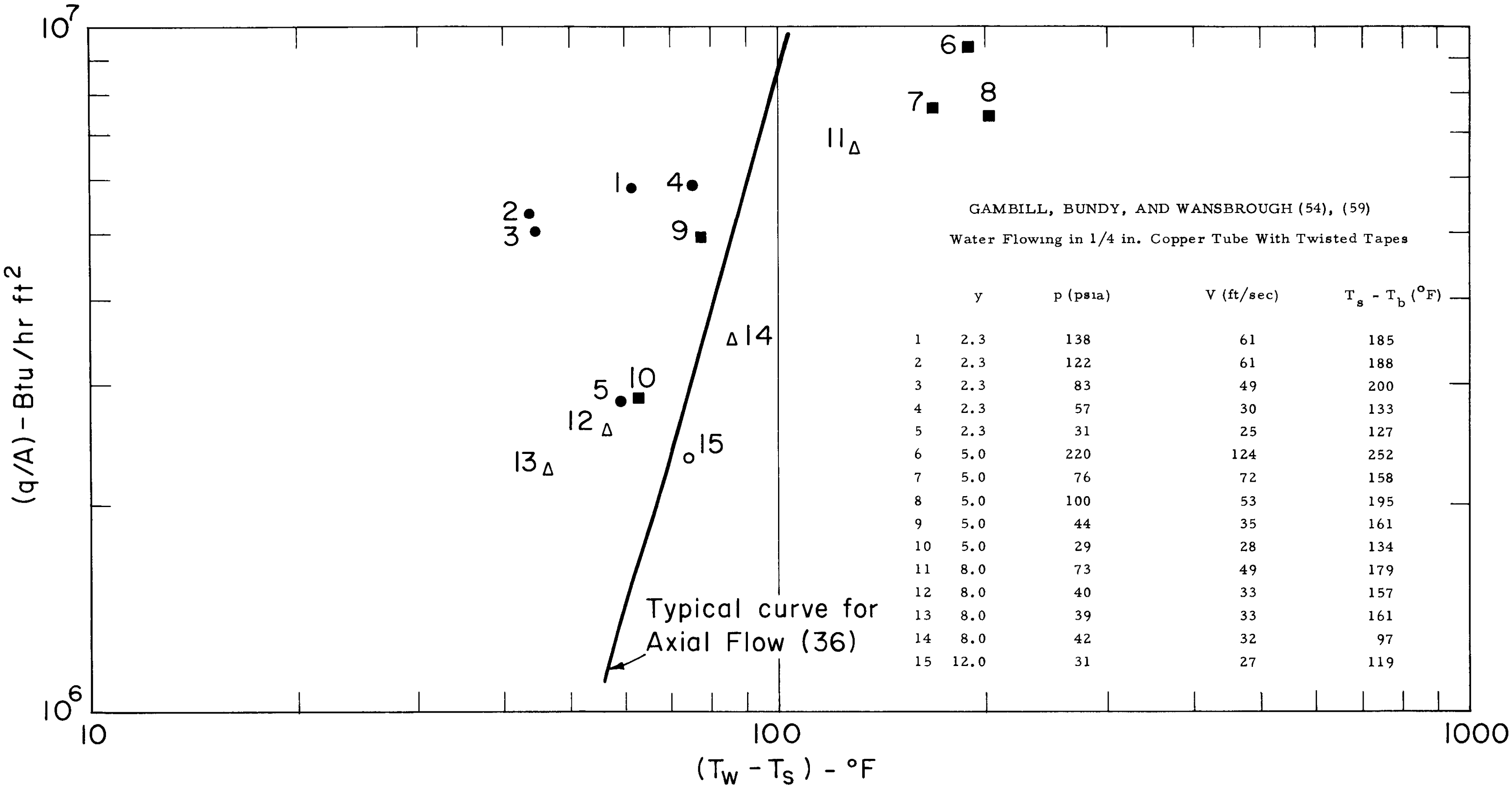
Several studies (47, 57) have considered twisted tapes which do not extend the length of the heated section. There would appear to be little advantage to this technique, except in systems where there is nonuniform heat generation.

In summary, it is noted that twisted tapes appear to be a very favorable augmentative technique. The performance factors for nonboiling forced convection are generally higher than those encountered with surface roughness elements and displaced turbulence promoters.

4.5.2 Surface Boiling

Gambill, Bundy, and Wansbrough (54) and Gambill and Bundy (58) are apparently the only investigators to report heat-transfer data for surface-boiling conditions. In order to examine the characteristics more closely, the water data of Ref.(54) were plotted and tabulated in Fig. 24. About the only conclusion which can be made on the basis of these limited data is that extremely high wall superheats are possible with swirl flow. There appears to be no centrifugal-acceleration correction which will bring the data into better perspective.

It is well established that burnout heat fluxes are higher for swirl flow than for axial flow. This is due primarily to the enhanced departure of the bubbles from the heated surface due to the radial force field. It is reasonable to speculate that the bubbles will tend to collapse away from the surface, with the result that they are less effective in increasing turbulence near the surface. In the region of fully-developed boiling, where convective effects no longer affect the boiling curve, higher wall superheats would be therefore expected for swirl flow. In any case, more experiments are necessary to clearly establish the effect of swirl flow on surface-boiling heat transfer.



	y	p (psia)	V (ft/sec)	$T_s - T_b$ (°F)
1	2.3	138	61	185
2	2.3	122	61	188
3	2.3	83	49	200
4	2.3	57	30	133
5	2.3	31	25	127
6	5.0	220	124	252
7	5.0	76	72	158
8	5.0	100	53	195
9	5.0	44	35	161
10	5.0	29	28	134
11	8.0	73	49	179
12	8.0	40	33	157
13	8.0	39	33	161
14	8.0	42	32	97
15	12.0	31	27	119

FIG. 24. SURFACE-BOILING HEAT TRANSFER WITH TWISTED-TAPE VORTEX GENERATOR

As usual the most important information needed for design is the burnout heat flux. Gambill, Bundy, and Wansbrough have presented adequate data for evaluation of the effect of swirl flow on surface-boiling burnout. It is generally most desirable to plot burnout heat flux versus exit subcooling with velocity, pressure, diameter, and length as parameters. The data of Gambill and co-workers have been presented in this form in Fig. 25. In order to permit clear visualization of the important trends, it was necessary to consider pressure and geometry as secondary variables and to designate only the various velocities. Certainly these variables contribute to the scatter of the data; however, as indicated in (60) they should not be of too great significance for the range of variables covered. The system stability, especially for the axial-flow tests, is of greater concern since considerable piping was installed between test section and flow-control valve. In any event the data do not appear to be unduly low, and since relatively few very high velocity data are available, these data should serve as an adequate reference. It is clearly evident that swirl flow produces a significant increase in the burnout heat flux.

These investigators have emphasized the insensitivity of burnout to subcooling. However, except for the lower velocities, the data do not bear this out. Even at high subcooling where the bubbles are small, the radial pressure gradient is effective in removing the vapor from the heated surface.

The most important feature of these data is that they enable a comparison on the basis of equal pumping power. Figure 26 essentially

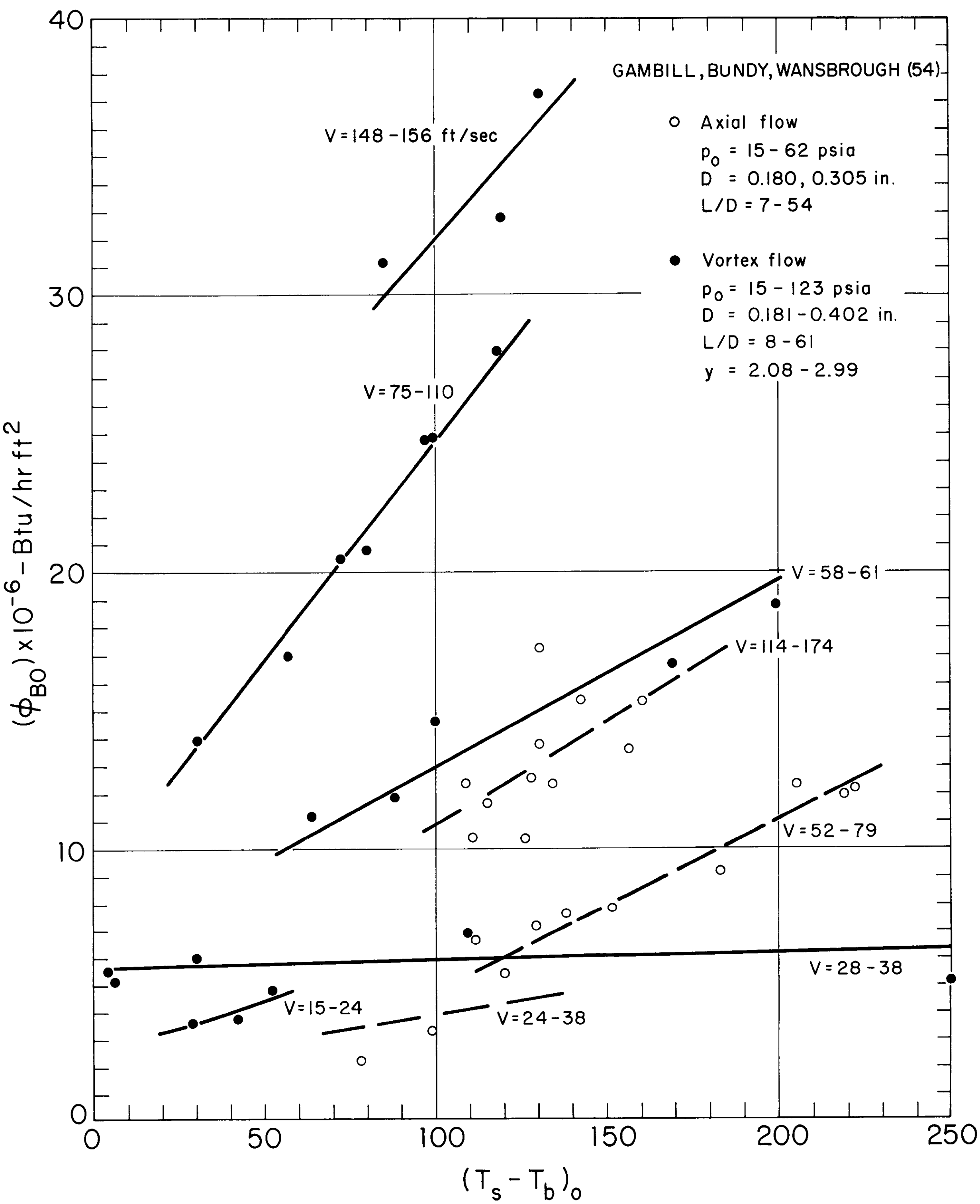


FIG. 25. INFLUENCE OF TWISTED-TAPE VORTEX GENERATOR ON SURFACE-BOILING BURNOUT

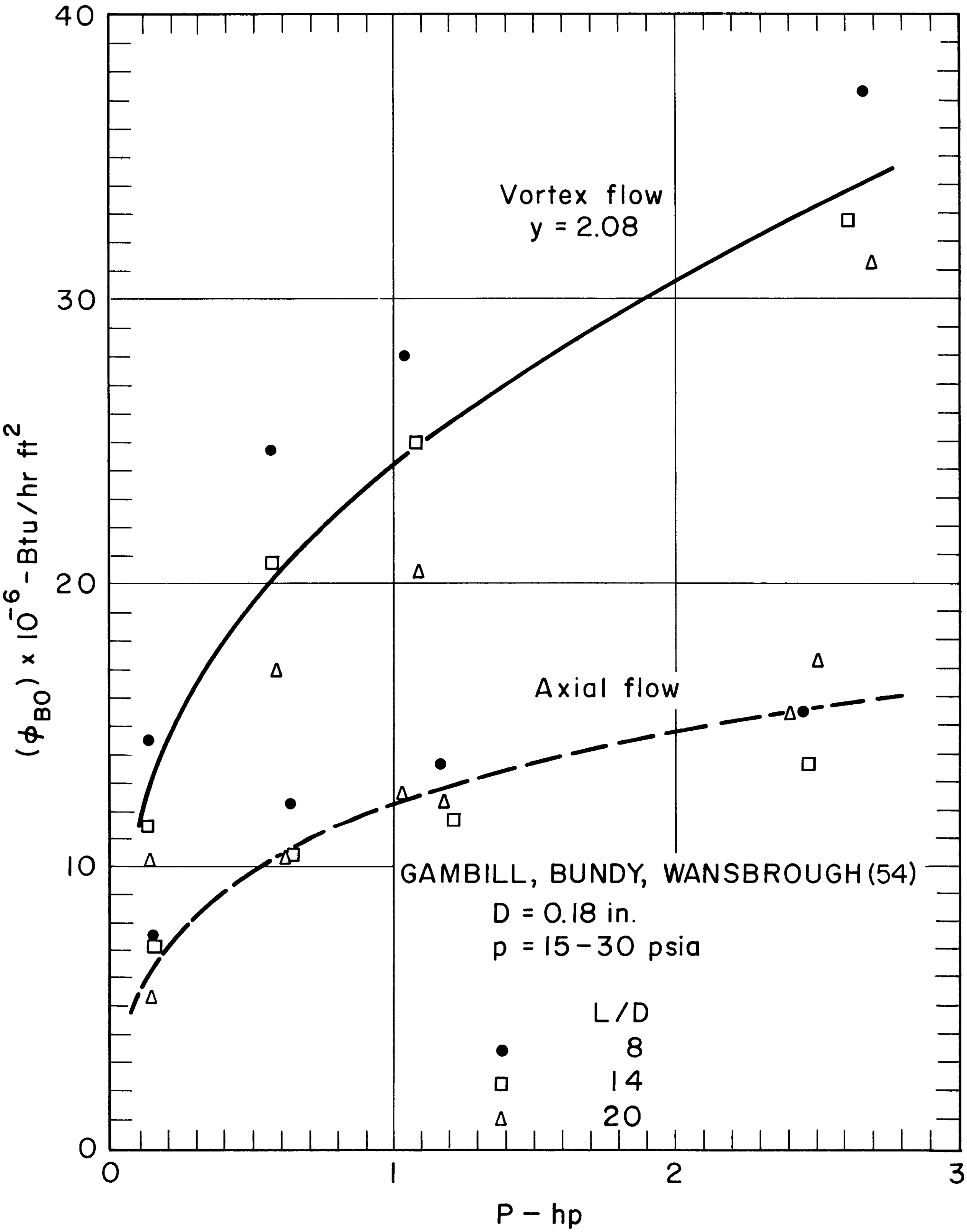


FIG. 26. DEPENDENCE OF SUBCOOLED BURNOUT ON PUMPING POWER FOR VORTEX- AND STRAIGHT-FLOW SYSTEMS

reproduces the comparison plot of these investigators. It is seen that burnout fluxes for swirl flow are approximately twice those for straight flow at the same test-section pumping power. The apparent trend with length, for the swirl-flow data at least, is not particularly significant since the exit subcooling was lower with the longer test sections.

Swirl flow has been used to advantage in situations requiring dissipation of extremely high heat fluxes. For example, integral twisted-tapes have been used in microwave power tubes (61). Feinstein and Lundberg (62) have reported a more recent study of swirl-flow burnout oriented toward this particular application.

An interesting variation of the usual twisted-tape system was recently considered by Gambill (63). Surface-boiling burnout data were taken for systems where all heat transfer took place from the tape surface. Burnout fluxes for the twisted tapes were slightly greater than those for flat tapes. It was postulated that the buoyant forces, which tend to hold the vapor on the heated surface, were overshadowed by complex secondary flows of the type described by Smithberg and Landis. Thus, it may be possible to further improve certain systems by generating heat in the twisted tapes as well as in the tube wall.

4.5.3 Bulk Boiling

Bulk-boiling heat-transfer data for water and freon 11 in swirl flow have been presented by Blatt and Adt (64). Average data for freon 11 in a twisted-tape system are presented in Fig. 27. There is some difficulty in interpreting these data as normal boiling curves since the exit conditions ranged from low quality to rather high superheat.

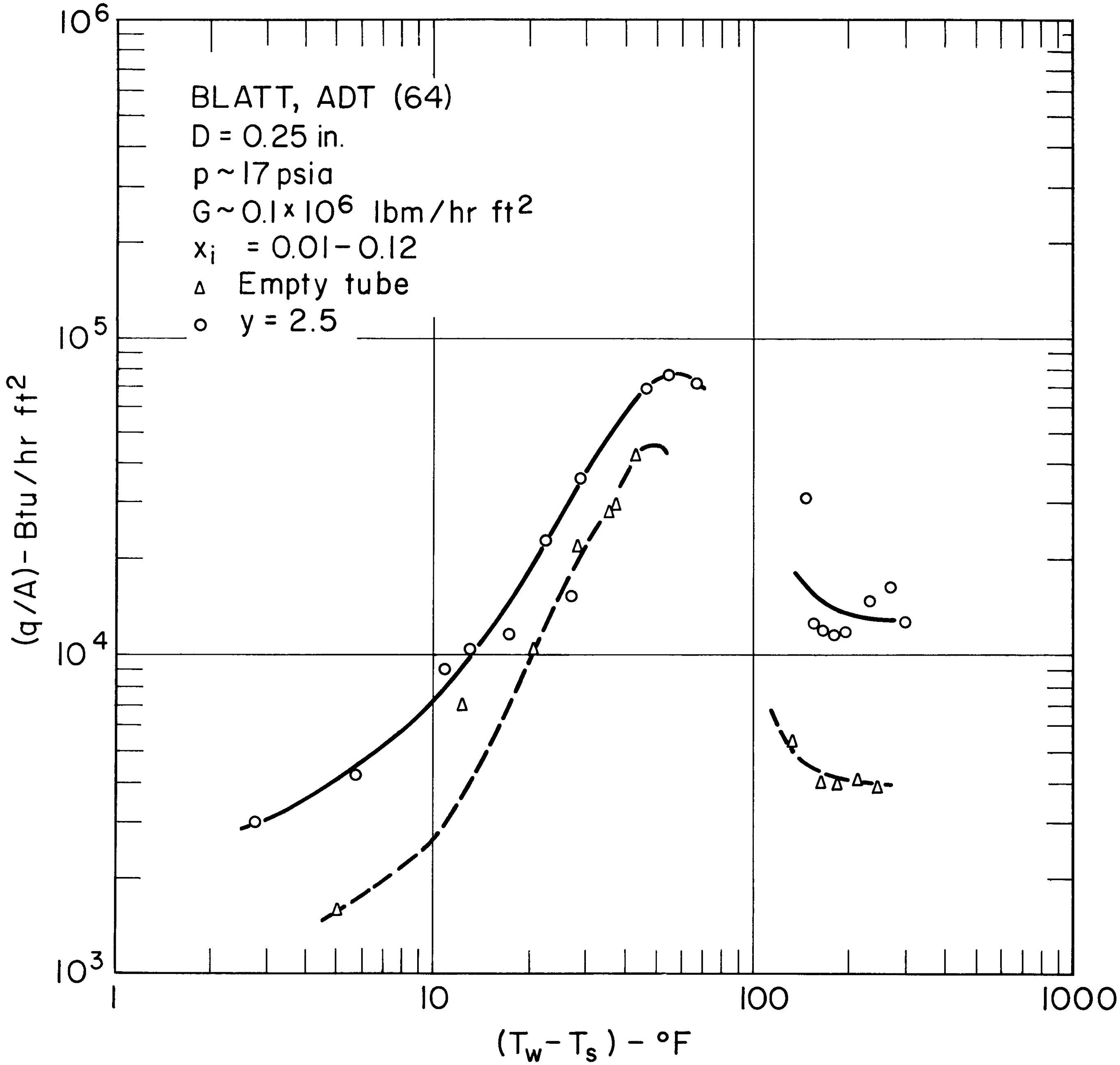


FIG. 27. INFLUENCE OF TWISTED-TAPE VORTEX GENERATOR ON BULK-BOILING HEAT TRANSFER

The twisted tape has a pronounced effect at low heat fluxes where a two-fold increase in heat-transfer coefficient can be noticed. There is less influence of the tape at higher heat fluxes; however, the peak heat flux appears to be raised significantly. The dramatic improvement in the lower film-boiling region could be partially due to the higher quality of the swirl data.

Pressure-drop data were also presented for low heat fluxes; however, the limited range of data did not permit the usual economic comparison. Extensive data for bulk-boiling of water were also presented which showed improved heat-transfer coefficients only at low heat fluxes. There was still a substantial increase in the pressure drop with swirl. These tests, then, indicate that the effectiveness of the twisted tapes for bulk boiling is dependent on the fluid as well as the flow conditions.

On the other hand, tests by Bernstein, et al. (29) with water indicated that twisted tapes of both plain and perforated types were effective in increasing heat-transfer coefficients (or delaying burnout) at high vapor qualities. Pressure drops were extremely high, however, especially at small twist ratios. Twisted tapes have been used in once-through boilers to reduce tube-wall temperatures in the high quality region (65). In this installation, a gap was maintained between the wall and the tape to avoid collection of impurities which might cause corrosion.

High-pressure burnout data for bulk boiling of water in swirl flow were reported by Viskanta (66). In Fig. 28 these data are compared with straight-flow predictions of Macbeth (67), which were chosen in preference to the ANL data due to the wider range of variables covered by the

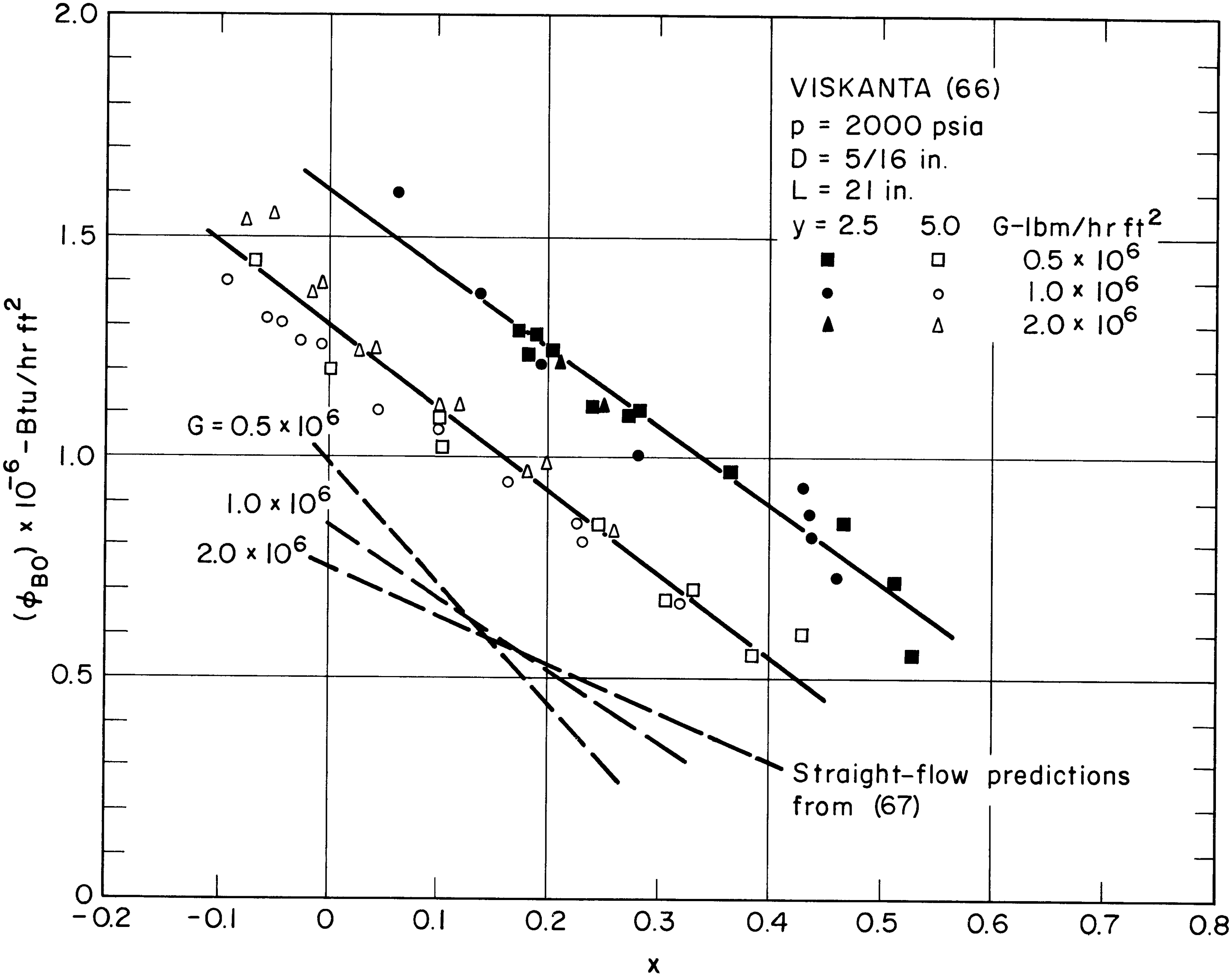


FIG. 28. INFLUENCE OF TWISTED-TAPE VORTEX GENERATOR ON BULK-BOILING BURNOUT

correlation. It is seen that substantial increases in the critical heat flux are produced by the twisted tapes. It is also interesting to note that the G-effect is considerably reduced with swirl flow.

Viskanta also presented a comparison in terms of pumping power which indicates that critical heat fluxes are as much as two times higher for swirl flow than for straight flow at the same pumping power. It is not clear whether actual boiling pressure-drop data were used for this comparison; however, the conclusion should be generally valid.

It is reasonable to assume that annular flow is the predominant flow regime when swirl is employed. This is due to the radial body force which tends to keep the liquid against the heated surface. The normally unstable slug flow would then be prevented. However, since the higher quality data would be in annular flow anyway, the swirl must also act to stabilize the film. Perhaps it does this by effectively preventing fog flow since any liquid will tend to remain at the wall due to the centrifuging.

Extensive experimental work in two-phase swirl flow has also been performed at SNECMA. Bulk-boiling burnout data have been reported for several types of channels with twisted-tape inserts. The various test-section geometries studied were round tubes, annuli (six tapes), and rod clusters (four heated rods with nine tapes). The preliminary data have been reported by Moussez and co-workers in numerous reports including (68, 69, 70). Typical increases in burnout of 30 percent have been reported, although improvements of over 60 percent were obtained with the annular geometry. A summary report on this work is to be issued shortly.

Swirl flow has also been applied to bulk boiling of liquid metals, including mercury and sodium, with favorable results. A discussion of some of the current work is included in the recent survey by Poppendiek, Gambill, and Greene (71).

5. VIBRATION

In recent years vibration has been seriously considered as a method of augmenting heat transfer. Some investigators have reported decreases in heat transfer while others have obtained over 600 percent increase in heat-transfer coefficients with vibration. In discussing the interactions between vibrations and heat transfer, it is appropriate to distinguish between two different techniques of applying the vibrations. The most direct approach is to vibrate the heated surface mechanically. In many applications, however, the large mass of the heat-transfer apparatus makes it difficult to employ this type of vibration. The second technique, then, has vibrations applied to the fluid and focussed towards the heated surface. Under certain conditions, though, the same improvement in heat transfer will result from either surface or fluid vibration. Numerous investigations have been reported for both heated surface and fluid vibration. Geometries have ranged from the simple single cylinder to a complex heat-exchanger core. In general, a fairly wide range of vibrational variables has been covered, and experiments have been run with both gases and liquids. As usual the liquid data are more complex due to the possibility of cavitation and boiling.

Since the interactions between vibrations and heat transfer are extremely complex, no comprehensive analytical treatment of the phenomenon has been formulated. Because of this lack of understanding and the discrepancy in empirical results, it is not yet possible to incorporate with confidence vibrational techniques in the design of heat-transfer equipment. In addition, data for economic evaluation are usually lacking.

The present discussion will thus be devoted to summarizing the various experiments and the conditions under which vibrations have been found to improve heat transfer.

5.1 Heated-Surface Vibration

The accompanying table briefly summarizes major investigations involving vibration of the heated surface. It is seen that increases in heat-transfer coefficients up to 600 percent have been reported.

5.1.1 Nonboiling

5.1.1.1 Horizontal Cylinders in Stagnant Fluid

The predominant geometry employed in these investigations has been the horizontal, heated cylinder vibrating either horizontally or vertically. It is appropriate to first compare data for this rather well-defined system.

It has been generally recognized that a certain critical intensity of vibration is required before h is affected. The vibrational intensity has been variously characterized as the amplitude-frequency product, af, or as the vibrational Reynolds number in terms of the average velocity of the heated surface, $4afD/\nu$. If the vibrational Reynolds number completely describes the effects of vibration, the following relation would be expected to apply

$$Nu = f(Gr, Pr, Re_v) . \tag{6}$$

This formulation was used effectively by Deaver, Penney, and Jefferson (76) to describe their water data. As indicated in Fig. 29, their averaged data fall into three rather distinct regions: the region of low Re_v where free convection dominates, a transition region where free convection

TABLE III

INVESTIGATIONS OF HEATED-SURFACE VIBRATION

Investigators	Test Arrangement	Results
Martinelli, Boelter (72)	Natural convection, water Horizontal tube, vibrating vertically	Up to 500% increase in h
Lemlich (73)	Natural convection, air Horizontal cylinder, vibrating vertically and horizontally	Up to 400% increase in h, independent of vibrational direction
Teleki, Fand, Kaye (74)	Natural convection, air Horizontal cylinder vibrating vertically	Up to 100% increase in h above critical intensity
Fand, Peebles (75)	Natural convection, air Horizontal cylinder, vibrating horizontally	Increase in h above critical intensity--similar to acoustic vibration
Deaver, Penney, Jefferson (76)	Natural convection, water Horizontal wire, vibrating vertically	Substantial increases in h; regions of vibrational influence delineated
Shine (77)	Natural and forced convection, air Horizontal tubes, vertical and horizontal vibrations	Up to 600% increase in h
Tsui (78)	Natural convection, air (analytical also) Vertical plate vibrating transversely	Up to 25% increase in h
Shine (79)	Natural convection, air Vertical plate vibrated transversely	Up to 50% increase in h above critical intensity
Schoenhals, Clark (80)	Natural convection (analytical also) Vertical plate, vibrating transversely	No increase in h predicted or observed for small vibrational amplitudes

Investigators	Test Arrangement	Results
Blankenship, Clark (81, 82)	Natural convection (analytical also) Vertical plate, vibrating transversely	Laminar h decreased as predicted; turbulent h increased by 50%
Scanlan (83)	Forced convection, water Heated surface in channel vibrating transversely	Increase in h up to 180% below certain intensity
Anantanarayanan, Ramachandran (84)	Forced convection axial flow, air Wire vibrating transversely	Up to 130% increase in h
Sreenivasan, Ramachandran (85)	Forced convection, air Horizontal cylinder, vibrated vertically	No effect of vibration
Raben (86, 87)	Forced convection including surface boiling, water Annuli, inner surfaces heated and vibrating	Up to 500% increase in h, improvement in boiling at low q/A
Palyeyev, Kachnelson, Tarakanovskii (88)	Forced convection normal to test sections, water and kerosene Cylinder vibrating axially	Up to 440% increase in h
Ogle, Engel (89)	Forced convection, water Annulus, inner surface heated and vibrating	No significant increase in h
Carr (90)	Saturated pool boiling, water Horizontal cylinder vibrating vertically	10% average increase in q/A at constant ΔT
Kovalenko (91)	Saturated pool boiling, water Horizontal cylinder vibrating vertically	Adverse effect on boiling except at very low q/A

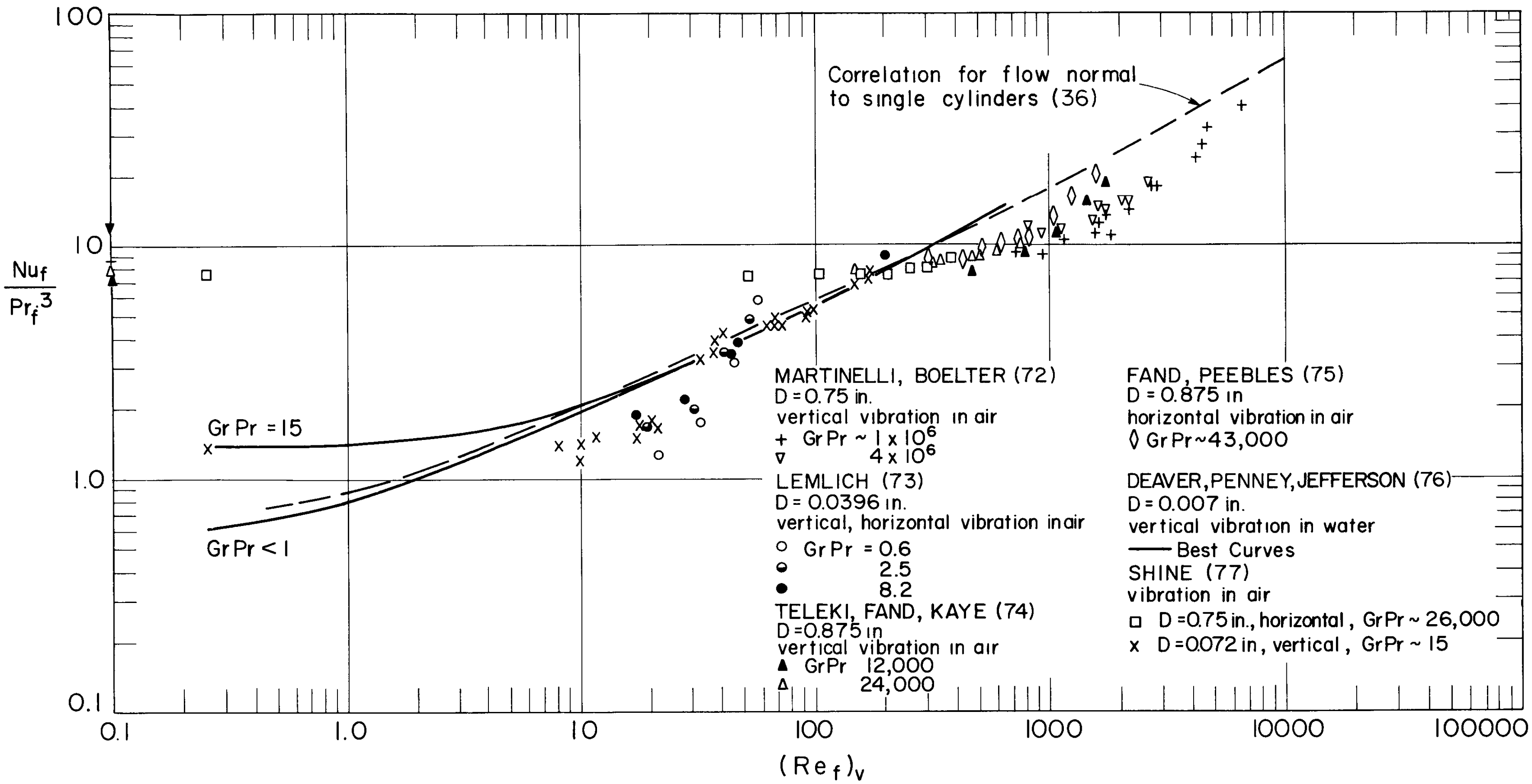


FIG. 29. INFLUENCE OF MECHANICAL VIBRATION ON HEAT TRANSFER FROM HORIZONTAL CYLINDERS IN STAGNANT FLUID

and the "forced" convection due to vibration interact, and finally the region of dominant forced convection. A significant contribution was made by these authors when they showed that this last region of vibrational effects was reasonably correlated by a standard correlation for forced flow normal to a cylinder. In commentary on that paper, Shine indicated that his data were in substantial agreement. Shine further indicated that the transition region was not of great significance for most systems. Following this proposal it is simple to evaluate the effect of vibrations on heat transfer for this geometry. The forced-convection correlation is equated to the natural-convection correlation to obtain the critical Re_v, and above this critical value the forced-convection correlation applies.

This simple description no doubt oversimplifies the complex interaction of the vibrations and heat transfer; however, its utility is confirmed by the diverse data which are approximately correlated. Although there is considerable scatter exhibited in Fig. 29, most of the data are within the scatter of the data of Deaver, et al. None of the results for liquids presented in Fig. 29 appear to have been taken for cavitating conditions.

5.1.1.2 Forced-Flow Systems

Substantial improvements in heat transfer have also been recorded when vibration of the heated surface is used in forced-flow systems. For this case, however, the geometrical arrangements and flow conditions are so varied that a simple correlation does not appear to be possible.

Figure 30 summarizes six investigations of heated-surface vibration with forced flow. The effect on heat transfer varies from slight

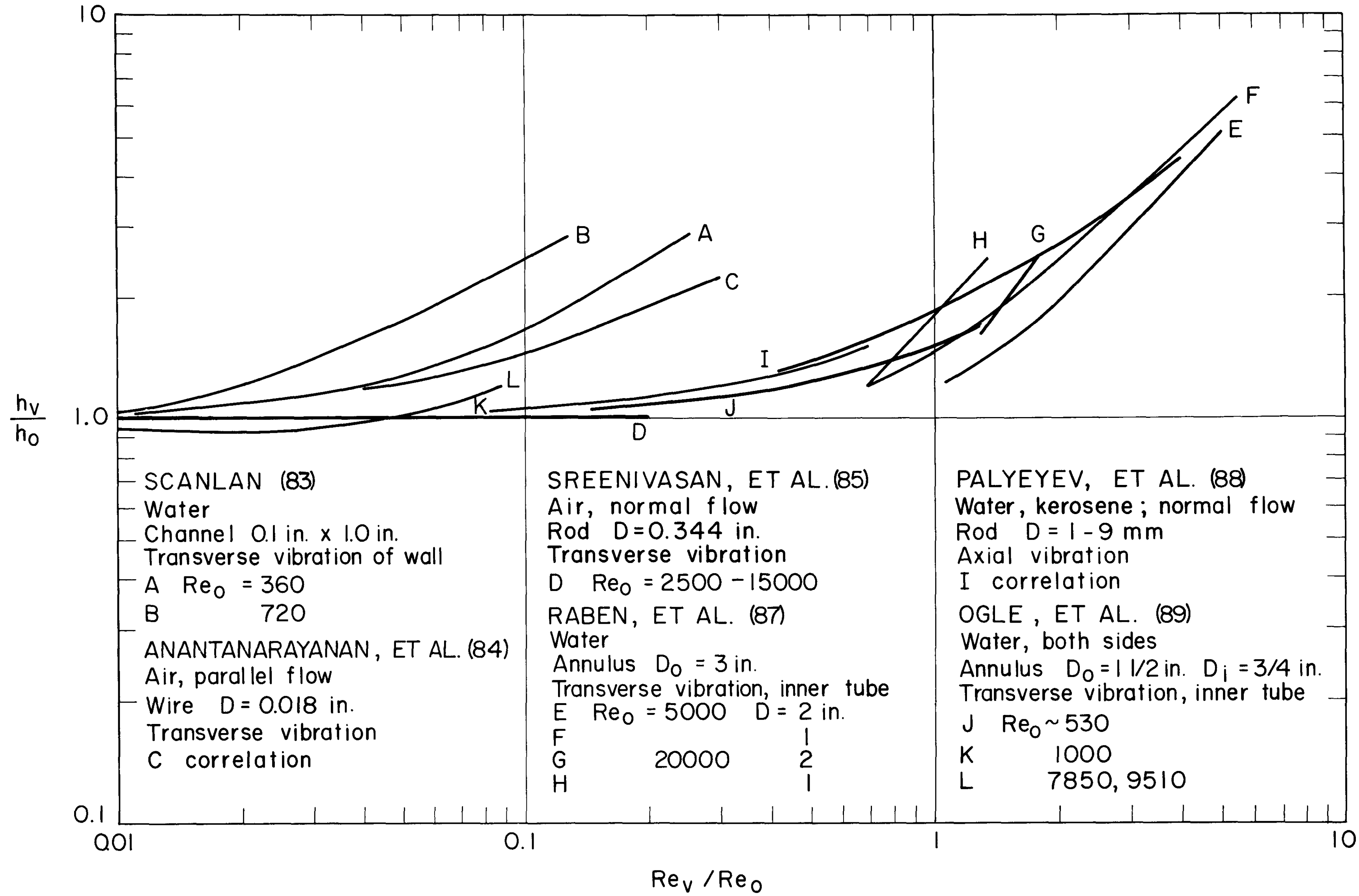


FIG. 30. EFFECT OF SURFACE VIBRATION ON HEAT TRANSFER TO FLUIDS IN FORCED FLOW

degradation to over 500 percent improvement depending on the system and the vibrational intensity. The parameters chosen for Fig. 30 were adequate to correlate data of several of the investigations; however, it is evident that they are not sufficient for general correlation.

The data of (84) for parallel flow appear to be successfully correlated in terms of a Reynolds number based on diameter rather than length. As pointed out in the discussion, however, the relatively large thermocouples attached to the wire could have introduced error in the evaluation of h. In the subsequent study (85), an analysis was made to show that the vibrational disturbances were too small to have any effect on heat transfer. Air has also been used as the working fluid in the studies noted by Shine (77).

Tests with liquids appear to be more difficult to interpret due to the possibility of cavitation at high intensities. Scanlan (83) found that the improvement in heat transfer was essentially dependent only on amplitude and frequency over a wide range of laminar flow Reynolds numbers. Above a certain combination of f and a, there was a sharp decrease in the improvement, which was attributed to the onset of cavitation and subsequent partial insulation of the heated surface. This behavior has not been confirmed by other investigators--if anything, there is a sharp increase in h_v/h_o when cavitation occurs. Deaver, et al., (76) noticed this in their experiments with an oscillating wire in a pool of water. The same sharp increase in improvement was recorded by Palyeyev, et al. (88) for water flowing normal to an oscillating cylinder; however, this was not apparent with kerosene.

The double-pipe heat-exchanger experiments of Ogle and Engel (89) show moderate increases in U_v/U_o except for turbulent flow at low vibrational intensities. The authors attempt to explain the decrease by postulating a suppression of the turbulence on the shell-side. This explanation does not concur with the tests of Raben, et al., (87) who recorded substantial increases with a similar annular geometry which was electrically heated. The various curves representing the data of Raben, et al., indicate that D_e and Re_f effects are important, at least for the more complex geometries.

5.1.2 Boiling

No conclusive work appears to have been performed with surface vibration where boiling is involved. Carr (90) made some measurements with saturated pool boiling of water using a steam-heated tubular test section. Low-frequency, high-amplitude vibration increased the heat transfer rate at constant $(T_w - T_s)$ by about 10 percent for both nucleate boiling and film boiling. Kovalenko (91) reported data for saturated pool boiling of water on a vibrating cylinder. At very low heat flux a small improvement in heat transfer was noted; at higher heat flux the heat transfer was adversely affected; and when the boiling became fully developed, there appeared to be no effect. Apparently the vibration disrupted nucleation in the region of less intense boiling. Since the maximum heat flux for this investigation was only $q/A = 8800$ Btu/hr ft^2, there is actually little that can be concluded regarding the effect of vibration on nucleate boiling.

Raben, et al., (87), have reported what appears to be the only study of forced-convection surface boiling with heated-surface vibration. For low heat fluxes, increases in heat transfer up to 130 percent were noted, as might be expected from the dramatic increases in nonboiling heat transfer. Since the improvement was reduced as the boiling became fully developed, it was concluded that little improvement in local boiling heat transfer would be expected in general from heated surface vibration.

Economic evaluation is difficult since sufficient data are not available. No comparative pressure drop data are reported for forced flow. However, the overriding consideration would be the cost of the vibrational equipment and the power required to run it. Ogle and Engel (89) found for one of their runs that about twenty times as much energy was supplied to the vibrator as was gained in improved heat transfer. Even though the vibrator mechanism was not optimized in this particular investigation, it is difficult to see how heated-surface vibration will be practical.

5.2 Fluid Vibration

Major investigations involving fluid vibrations are summarized below in Table IV. Substantial improvements in heat transfer have been reported; however, the description of the phenomenon is even more difficult than that of surface vibration. In particular, the vibrational variables are more difficult to define due to the remote placement of the transducer. In general a wider range of frequency and amplitude is possible with this indirect method. With the use of higher intensities, cavitation will occur frequently in experiments with liquids.

5.2.1 Gases

There has been a great deal of research effort directed to studying the interaction of acoustic fields and heat transfer for the case of single horizontal cylinders in air. As in other augmentative areas this

TABLE IV

INVESTIGATIONS OF ACOUSTIC VIBRATIONS - GASES

Investigators	Experimental System	Results
Kubanskii (92)	Free convection, air Horizontal cylinder Axial acoustic field	Up to 100% increase in h
Holman, Mott-Smith (93)	Free convection, air Horizontal cylinder Transverse sound field	Up to 105% increase in h
Fand, Kaye (94)	Free convection, air Horizontal cylinder Transverse sound field	Up to 160% increase in h
Sprott, Holman, Durand (95)	Free convection, air Horizontal cylinder Transverse sound field	Up to 200% increase in h
June, Baker (96)	Free convection, air Vertical plate Transverse sound field	Up to 220% increase in h
Kubanskii (97)	Forced convection, air Horizontal cylinder Sound wave ⊥ flow, ⊥ axis of cylinder, also parallel to axis	Up to 50% increase in h
Fand, Cheng (98)	Forced convection, air Horizontal cylinder Acoustic vibrations ⊥ cylinder and flow	Up to 25% increase in h
Fussell, Tao (99)	Forced convection, air Horizontal cylinder Transverse sound field	Up to 40% increase in h at low Re, large distance
Jackson, et al (100, 101, 102)	Forced convection, air Tube Speaker at inlet	Up to 26% increase in h at low Re, decrease at high Re

Investigators	Experimental System	Results
Lemlich, Hwu (103)	Forced convection, air Tube Speaker at inlet	Up to 51% increase in laminar h
Lemlich (104)	Forced convection, air Tube Musical reed at inlet	Up to 35% increase in h
Mathewson, Smith (105)	Forced convection, air and isopropanol (condensing) Tube Siren at inlet	Up to 44% increase in h Up to 60% increase in condensing h
Moissis, Maroti (106)	Forced convection, air Automotive-type radiator section Siren at inlet	Increases in U up to 30%

geometry has proved to be convenient to study and analyze. The listing in Table IV includes representative studies chosen from the rather extensive literature, in particular those which present experimental data.

Kubanskii (92) obtained improvements up to 100 percent when a speaker was oriented along the axis of a heated cylinder. Holman and Mott-Smith (93) reported increases in heat transfer of over 100 percent for a constant-pressure sound field (directed normal to a heated cylinder). In an extensive research program in this area, Fand and Kaye (94) demonstrated improvements of up to 160 percent with intense acoustic vibrations directed normal to cylinders. Local heat-transfer coefficients were later measured to help clarify the augmentative mechanism (107).

It is well established that the improvements in heat transfer are due to an acoustically induced vortex flow, or thermoacoustic streaming, near the heated surface. A critical sound intensity is required for the inception of this streaming. The transition region, which occurs before the vortices become fully developed, cannot be described as a simple superposition since the governing equations are nonlinear.

The effect of vibrating the heating surface on convective heat transfer appears to be comparable to the effect of vibrating the fluid with acoustical vibration in a similar convective case if the wavelength of the sound is large and the amplitude of vibration of the vibrating heated surface is small compared to a characteristic length of the system (75, 108, 98).

Superposition of axial cross flow upon this simple system greatly reduces the effect of the acoustic field. As shown in Fig. 31 there is a decrease followed by a modest increase as the cross-flow velocity is

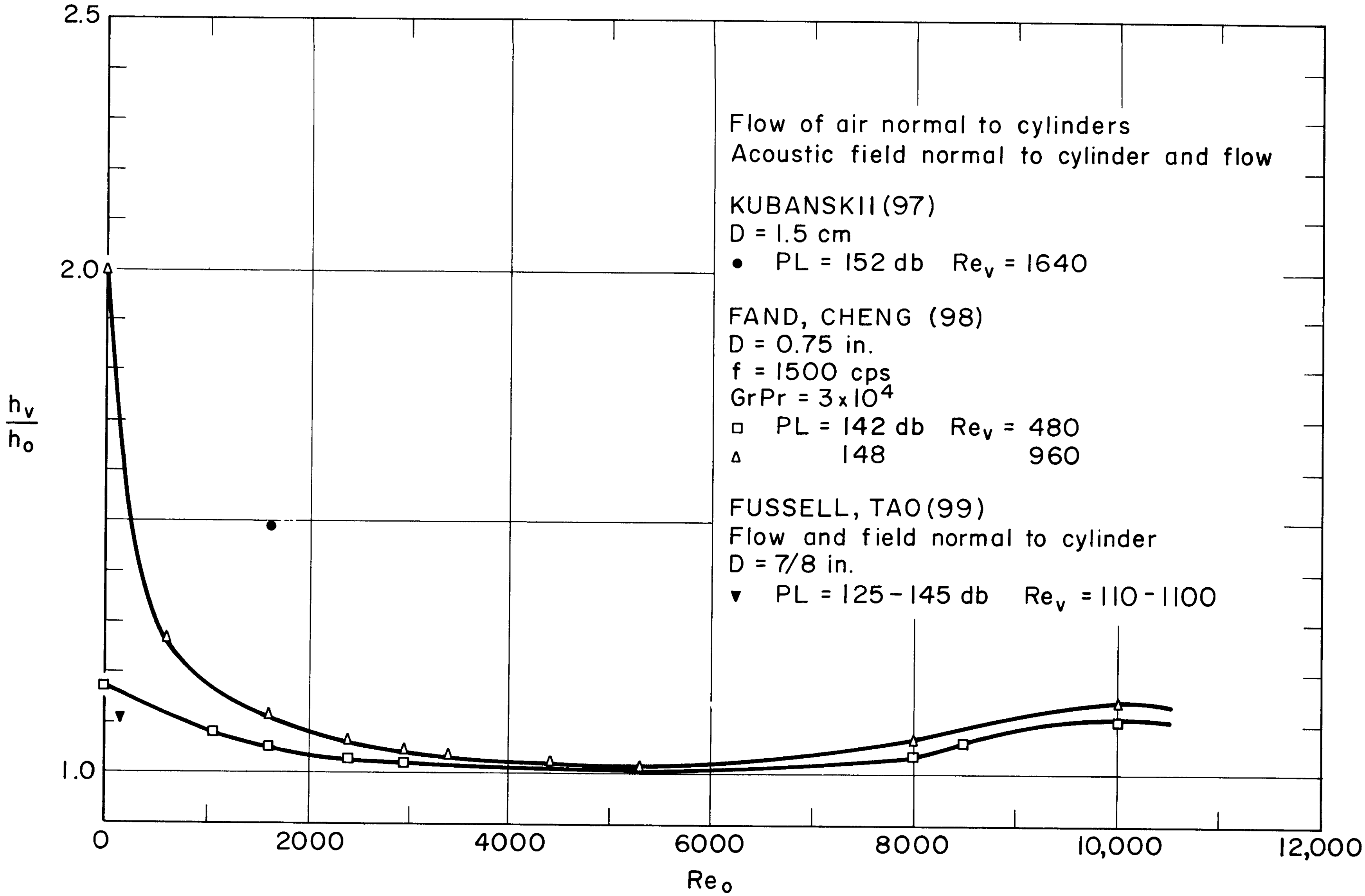


FIG. 31. EFFECT OF ACOUSTIC VIBRATIONS ON HEAT TRANSFER TO AIR FLOWING OVER CYLINDERS

increased. Fand and Cheng (98) interpret the mechanism at low velocities to be still thermoacoustic streaming. The increase at high velocity is conjectured to be due to complicated flow interactions. The very limited data of Kubanskii (97) for a similar arrangement show that the heat transfer with cross flow can still be materially improved if a very intense sound field is employed. Fussell and Tao (99) are roughly in agreement with the lower intensity results of Fand and Cheng.

Numerous attempts have been made to apply acoustic vibrations to the more practical case of flow in channels. Results of four investigations are summarized with average heat-transfer coefficients in Fig. 32. In all cases loudspeakers were installed at the inlet to the test section and were carefully tuned to operate at resonance. Jackson, Harrison, and Boteler (100) made local measurements and recorded periodic variation of the local heat-transfer coefficient. The maximum h occurred at the half wave length position, which was the antinode of the sound field. Later visual studies showed the existence of thermoacoustic streaming (102). In a subsequent study, Jackson, Purdy, and Oliver (101) recorded a decrease in the improvement as the Reynolds number increased. The entrance effect was very pronounced at low Re, and local improvements of over 130 percent were achieved. This entrance effect was less pronounced as Re increased.

Lemlich and Hwu (103) reported measurements of over-all coefficients for a similar geometry. They found that the improvement was greatest near the transition range of Re. This led to the conclusion that the vibratory motion, in addition to acting as a disturbance in itself, was also a turbulence trigger.

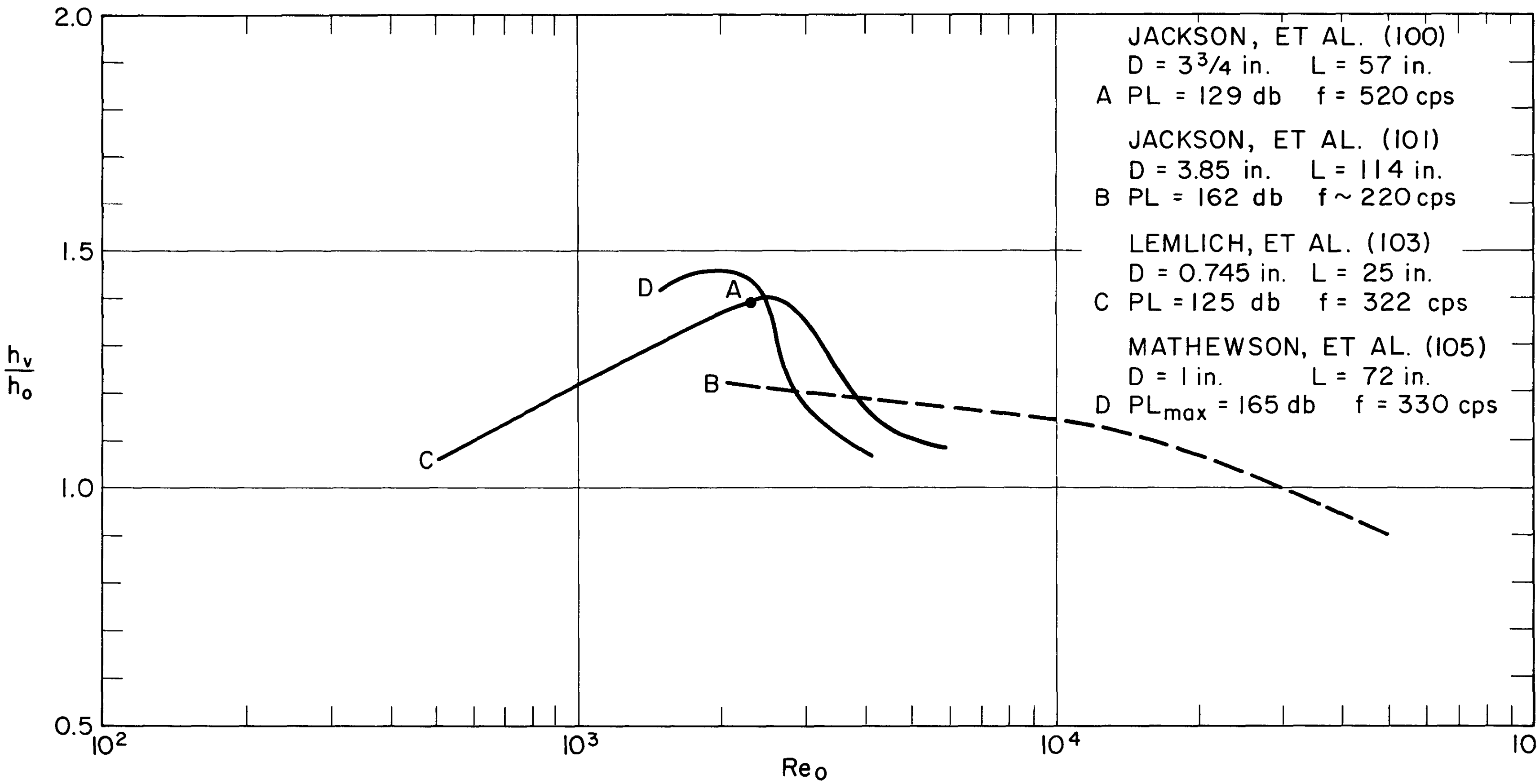


FIG. 32. INFLUENCE OF ACOUSTIC VIBRATIONS ON HEAT TRANSFER TO AIR FLOWING IN TUBES

Mathewson and Smith (105) reported similar data for air. They used the same system to demonstrate that turbulent film condensation of isopropanol was improved by as much as 60 percent when the sound was applied.

More complex geometries have also been considered. Moissis and Maroti (106) applied acoustic vibrations at the inlet of an automotive-type radiator section. When a standing wave was obtained by means of a reflector, improvements in over-all coefficients up to 30 percent were obtained.

5.2.2 Liquids

An outline of experiments where vibrations have been applied to liquids is given in Table V.

5.2.2.1 Pool Experiments

Several investigators have considered the effect of vibration on heat transfer from wires to water in natural convection. Gibbons and Houghton (109) recorded the effects of frequency on nonboiling, nucleate-boiling, and film-boiling heat transfer. Nonboiling coefficients were found to increase as much as 500 percent at 20 cps. The effect was considerably less at higher frequencies, even though the intensity was maintained relatively constant. Nucleate-boiling heat-transfer coefficients were increased over a certain range of frequencies, but film boiling was unaffected. These investigators noted increased nucleation and substantial bubble break-up with vibration. Isakoff (110) found that the burnout heat flux in saturated pool boiling of water was raised 60 percent by application of intense sonic vibrations to the pool. No discernable shifting of the boiling curve was noted, although higher heat fluxes were required for nucleation. Ornatskii and Shcherbakov (111) applied

TABLE V

INVESTIGATIONS OF ACOUSTIC VIBRATIONS - LIQUIDS

Investigators	Experimental System	Results
Gibbons, Houghton (109)	Natural convection; nucleate and film pool boiling; water Horizontal wire Vibrating piston at bottom of pool	Up to 500% increase in natural convection, less effect on nucleate and film boiling
Isakoff (110)	Saturated pool boiling, water Horizontal wire Vibrating diaphragm at bottom of pool	60% increase in burnout
Ornatskii, Shcherbakov (111)	Pool boiling, water Horizontal wire Ultrasonic transducer at bottom of pool	80% increase in burnout at high subcooling
DiCicco, Schoenhals (112)	Saturated pool film boiling, Refrigerant 11 Horizontal wire, fluid pulsed	Up to 100% increase in q/A at constant ΔT
Zhukauskas, et al. (113)	Free and forced convection, water and oil Tubes and plates Ultrasonic transducer at bottom of tank	180% increase in h at low Gr, little increase at moderate velocity
Larson, London (114)	Free and forced convection, water and toluene Sphere Ultrasonic transducer at bottom of channel	300% increase in free-convection h, no increase at moderate velocity
Martinelli, Boelter, Weinberg, Takahi (115)	Forced convection, water Tube Pump pulsations	Negligible increase in h for laminar and turbulent region
Marchant (116)	Forced convection, water Tube Pump pulsations	Up to 40% increase in h at low Re

Investigators	Experimental System	Results
West, Taylor (117)	Forced convection, water Tube Pulsations generated by pump	Up to 70% increase in h
Shirotsuka, Honda Shima (118)	Forced convection, water Tube Pulsation generator at inlet	Over 100% increase in turbulent h
Linke, Hufschmidt (119)	Forced convection, oil Single and multiple tubes Flow pulsation at inlet	Up to 380% increase in h in laminar range, 30% increase in turbulent range
Darling (120)	Forced convection, water and glycerol solution Tube Flow interrupter upstream and downstream	Up to 70% increase in turbulent h with interrupter upstream
Lemlich, Armour (121)	Forced convection, water Annulus, inner tube heated Flow interrupter upstream and downstream	Up to 50% increase in h with interrupter upstream
Bergles (122)	Forced convection including surface boiling, water Tube Piston-type transducer at test section exit	Up to 50% increase in h at high ΔT Little effect on boiling
Romie, Aronson (123)	Forced-convection Surface boiling, water Annulus, inner tube heated Ultrasonic transducer upstream	Burnout unaffected by vibration
Bergles, Newell (124)	Forced convection including surface boiling, water Annulus, inner tube heated Outer tube vibrated ultrasonically	Local increases in h up to 40% at high ΔT Little effect on boiling

one-megacycle vibrations to a pool-boiling system. Improvements in burn-out varied from 30 to 80 percent over the subcooling range of 6 to 146 ^{o}F.

DiCicco and Schoenhals (112) noticed up to 100 percent improvement in heat transfer when fluid pulsations were applied to a film-boiling system. Zhukauskas and co-workers (113) carried out ultrasonic vibration studies with water and transformer oil under conditions of free and forced convection. Vibrations increased heat-transfer coefficients by as much as 180 percent at low Grashof numbers, but at higher Grashof numbers the increase was substantially less. Crystal wind and cavitation were noted and suggested as probable mechanisms. However, since these effects were much less pronounced with oil, it was suggested that microflow near the walls was also important. Forced-convection tests were made with electrically heated tubes and plates. An increase in heat-transfer coefficient of 81 percent was observed when the plates formed narrow channels and the velocity was 0.1 m/sec. However, as the velocity was increased to 1 m/sec, there was virtually no increase in the heat-transfer coefficient, even at intensities of 2 w/cm^2.

Larson and London (114) reported extensive studies of heat transfer from a sphere to water and toluene where the liquid was subjected to ultrasonic agitation. As much as a 300 percent increase was observed in the heat-transfer coefficient in natural convection. At low frequencies this increase was attributed to cavitation, while at higher frequencies the quartz wind streaming appeared to be important. The tests were run with higher Grashof numbers than those used by Zhukauskas, et al.; however, no pronounced decrease in the effect of vibration was noted as the Grashof number was increased. The increase in film coefficient was found to be negligible as the flow velocity was increased above 0.2 ft/sec.

5.2.2.2 Channel Experiments

The investigations mentioned above were primarily concerned with heat transfer from test sections located in a large body of fluid. The case of unbounded convection is, however, of limited practical interest since most heat-transfer equipment involves flow in ducts. Investigations of heat transfer where vibration has been applied to the fluid have been run with test arrangements where the vibrator was located upstream or downstream of the heated section.

A popular approach has utilized low-frequency vibrations or pulsation, either naturally occurring or produced by relatively simple flow interrupters. The results of these investigations are summarized in Fig. 33.

Martinelli, et al. (115) detected no change in heat transfer in a pulsating system. In the discussion of that paper, Marchant presented limited data which indicated up to 40 percent increase in h at low Re. West and Taylor (117) achieved increases in heat-transfer coefficients with water by only partially damping the pulsating flow from a piston-type pump. At optimum amplitudes of pulsation, turbulent coefficients were increased by 70 percent. Linke and Hufschmidt (119) reported Aachen data which show a striking improvement at Re = 1250. At a higher Re, however, the improvement is much less. One would suspect turbulence triggering at the lower Reynolds number. Darling (120) obtained similar improvements by employing an interrupter valve upstream of the heated section. No increase in heat transfer was noted when the valve was placed

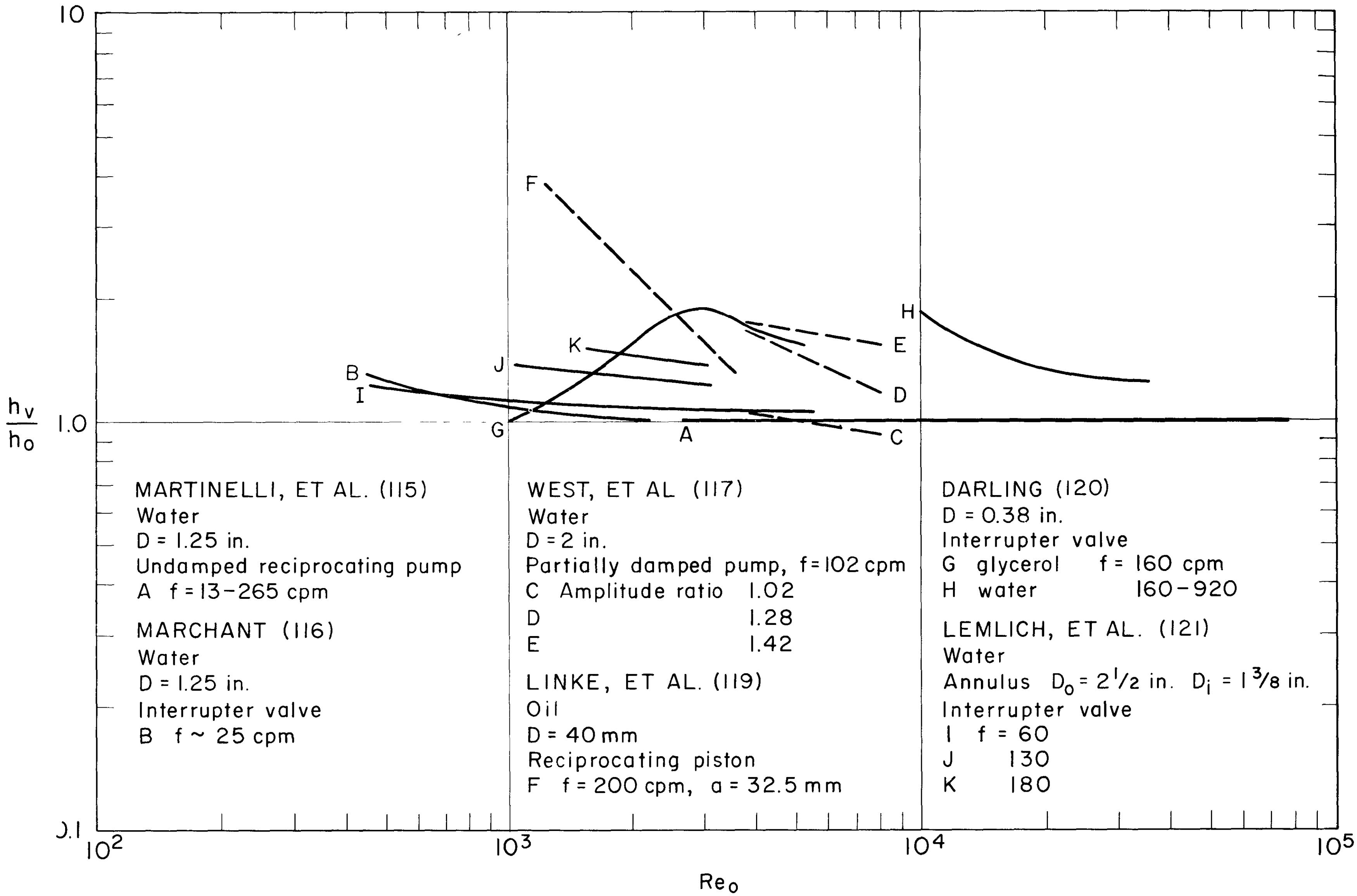


FIG. 33. EFFECT OF INLET PULSATIONS ON HEAT TRANSFER TO LIQUIDS FLOWING IN PIPES

downstream, so it was speculated that cavitation was the primary mechanism of improvement. Lack of cavitation at low Re was suggested to explain the strange behavior of the glycol data. Lemlich and Armour (121) used a similar apparatus to demonstrate that the interrupter should be located as close as possible to the upstream end of the heated section. Visual studies confirmed that cavitation at the heated surface was indeed the cause of the increased heat transfer. All the experiments show that the effect of vibration is reduced as the Reynolds number increases.

The data of Shirotsuka, et al. (118) were also obtained with a system with inlet pulsations. Coefficients were improved by over 100 percent at Re = 3000, but the improvement was negligible at Re = 10,000. Friction data were included and would have been useful in evaluating the performance if they had not been taken for different pulsation conditions.

In a recent investigation (122), 80-cps vibrations were applied to the turbulent water flow at the exit section of an electrically heated tube. The actual vibrational intensity in the heated section was estimated to be low; however, increases in heat-transfer coefficients up to 50 percent were recorded. As shown in Fig. 34, the vibration had an effect on heat transfer only at relatively high surface-minus-fluid temperature differences. This effect became less as surface boiling was initiated, and with fully developed boiling, including burnout, there was little effect of vibration.

Subcooled burnout of water in an annulus under the influence of ultrasonic vibrations was investigated by Romie and Aronson (123). The transducer was located upstream of the test section. Visual inspection indicated that bubble size was reduced and frequency of bubble formation

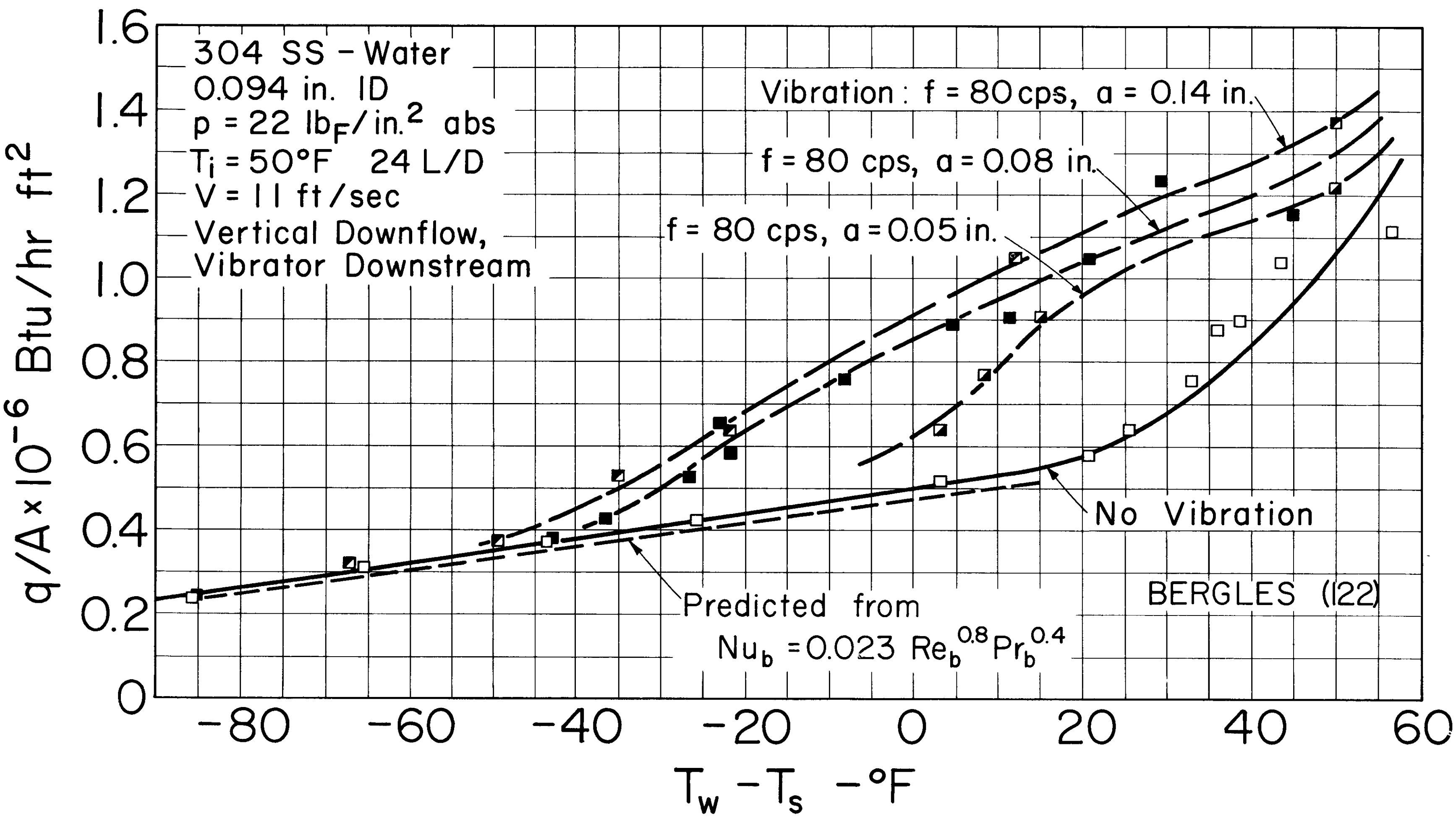


FIG. 34. INFLUENCE OF LOW-FREQUENCY VIBRATIONS ON NONBOILING AND BOILING HEAT TRANSFER

increased at moderate boiling fluxes. However, no such effect was noted near burnout, and burnout fluxes were essentially unaffected by the ultrasonics. Attenuation of the ultrasonic energy by the vapor was noted to be the probable cause of the weak influence of the ultrasonics on vigorous boiling.

As noted in these two preceding investigations, there is considerable attenuation of the vibrational intensity when the transducer is located upstream or downstream of the test channel. An apparatus was designed by Bergles and Newell (124) which permitted the application of intense ultrasonic vibrations to the fluid in the immediate vicinity of a heated surface. As indicated in Fig. 35 heat transfer was improved with vibration at low velocities and high nonboiling heat fluxes. The high heat fluxes were conducive to cavitation, which was effective so long as it occurred at the heated surface. There was less effect of vibration when cavitation occurred at the transducer since the resulting vapor attenuated the vibrational intensity. Fully developed surface boiling was unaffected due to the attentuation by the vapor and the dominance of the bubble agitation.

These experiments, then, indicate that under certain conditions, significant increases in heat-transfer coefficients can be achieved when vibrations are applied to liquids. The most dramatic increases have occurred in cases of unbounded natural convection, although some improvement has been noted with flow in channels. There is usually considerable attenuation of the sound field when the transducer is located upstream or downstream of the test channel.

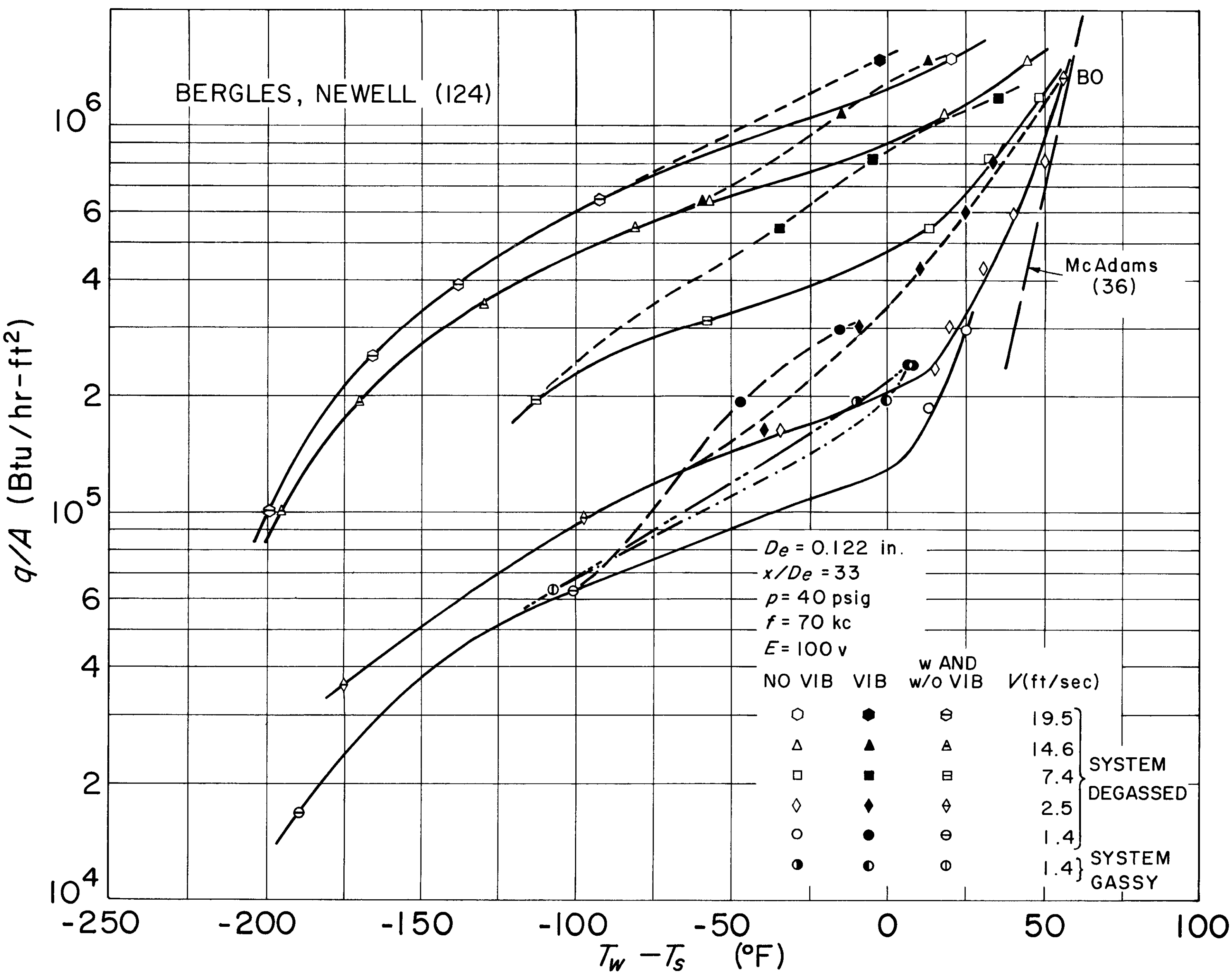


FIG. 35. INFLUENCE OF ULTRASONIC VIBRATIONS ON HEAT TRANSFER UNDER NONBOILING AND BOILING CONDITIONS

6. ELECTROSTATIC FIELDS

The most fascinating augmentative technique is electrohydrodynamics or EHD, which utilizes electrostatic fields to influence heat transfer. Electrostatic fields augment convective heat transfer by means of an electrostatic body force, which can be directed to cause greater bulk mixing of the fluid due to density differences. Since electrostatic or Coulomb forces are proportional to electric field strength, it is economically appropriate to consider EHD only for use with poorly conducting (dielectric) fluids.* Although the beneficial effects of EHD were reported almost thirty years ago, it is only in recent years that full-scale research has been initiated in this area. The results of this work are encouraging, and practical applications are being seriously considered.

6.1 Nonboiling

Senftleben and Braun (126) inaugurated work in this area by studying the influence of a radial electric field on free convection from a heated horizontal wire. The 0.03-mm wire was located in a concentric tube which was then maintained at a high voltage relative to the wire. This produced a highly divergent electrical field. Up to 50 percent improvement in heat transfer was recorded with gases, including air, oxygen, and C_2H_5Cl.

This effect has been explained in terms of the dielectrophoretic force which causes the fluid to move toward the region of highest field

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*An electric field can also be used with a magnetic field to exert a force on an electrically conducting fluid. This electromagnetic pumping has been proposed as a means of increasing condensation heat-transfer rates (125).

strength. When two phases are involved, the phase with the higher dielectric constant will move in this direction. Since the dielectric constant increases with increasing fluid density, the hot fluid near the surface will tend to be displaced by the colder fluid from the free stream. The increased mixing is responsible for the improvement in heat transfer.

The electrostatic forces are generally very small for gases, and as a result the improvements in heat transfer are modest. A discussion of the effects of a radial electrostatic field on heat transfer to gases, including additional references, has been presented by Motulevich, et al. (127). A gas-cooled reactor configuration with EHD was tested by Berger and Derian (128). Heat transfer was generally improved depending on the flow conditions.

Marco and Velkoff (129) studied the interactions between an electrostatic field and the natural-convection boundary layer where a fine-wire electrode and a flat plate were used. Coefficients in the vicinity of the wire were improved by a factor of four. These investigators concluded that corona wind, caused by ionization of the air near the wire, was the primary augmentative mechanism for this case. It is expected that the corona-wind phenomenon would be overshadowed by any appreciable convection.

Investigations with liquids using an annular geometry have been reported by Ashmann and Kronig (130), De Haan (131), Allen (132), and Choi (133). The improvements are generally larger than obtained with gases. Choi's data as shown in Fig. 36 indicate a 200 percent improvement in natural convection with freon. Bonjour and co-workers (134) utilized a parallel-wire geometry which also gave a nonuniform

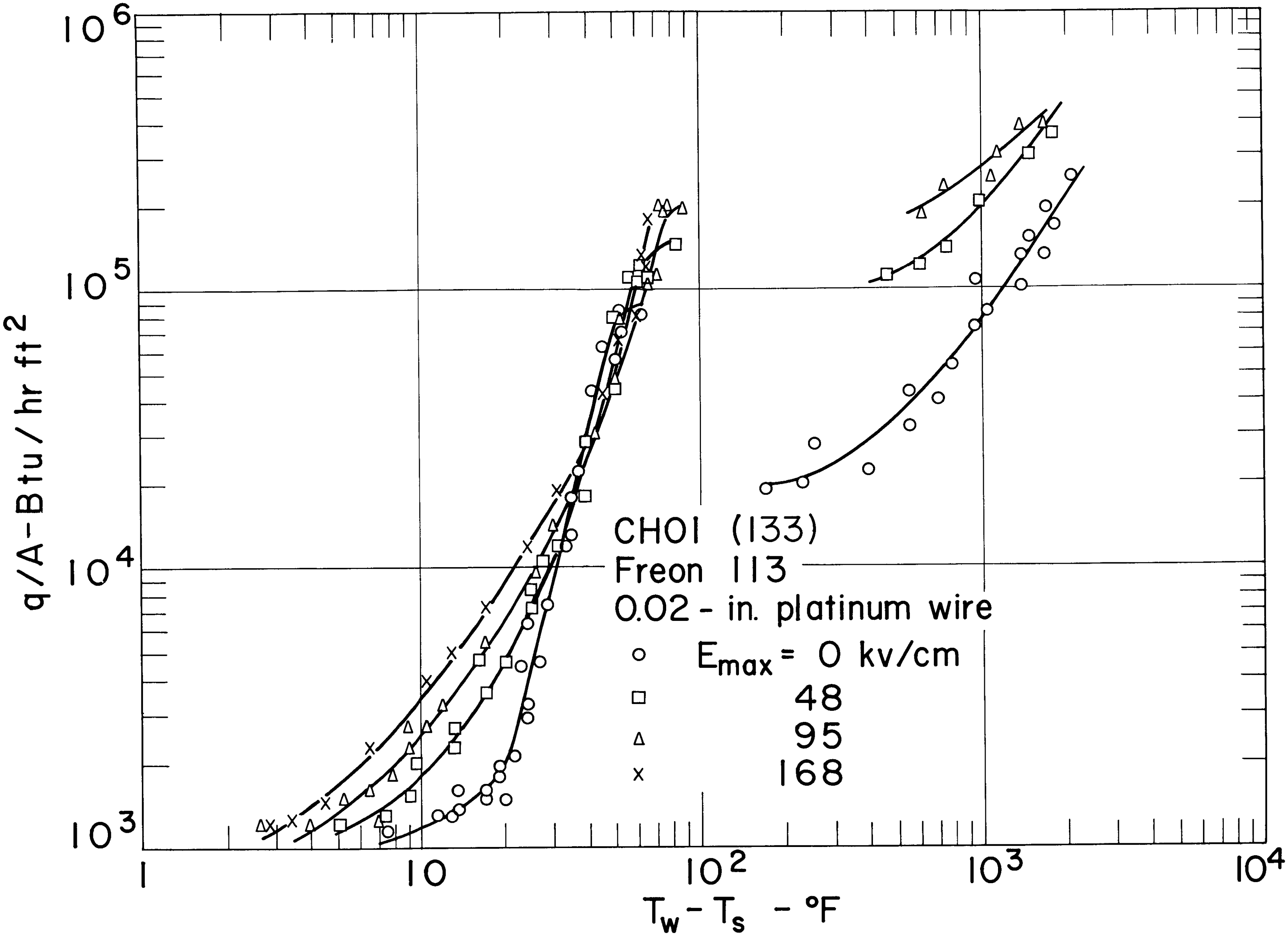

FIG. 36. INFLUENCE OF ELECTROSTATIC FIELDS ON POOL BOILING HEAT TRANSFER

electrostatic field. Their data indicate, for example, that heat-transfer coefficients can be increased by 400 percent for ethyl ether in natural convection with a field strength of 160 kv/cm.

Attempts have been made to correlate the natural convection annular data in terms of the usual Nu, Pr, and Gr numbers plus an electrostatic Grashof number, generally referred to as the Senftleben number. This correlating parameter was suggested by Kronig and Schwarz (135) and used in modified form with success for both gas and liquid systems by several investigators (130, 127, 133, 136).

Schmidt and Leidenfrost (137) applied a radial electric field to the fully-developed laminar flow of transformer oil in a horizontal annulus with the inner surface heated. Improvements in heat transfer of over 400 percent were recorded. Some increase in pressure drop was also noted. This more practical forced-flow situation was also considered in a recent study by Levy (138). As shown in Fig. 37 application of the field in the annular gap produced improvements of some 140 percent in the heat transfer. An electrostatic parameter, El, representing the ratio of electrical body force to inertial force, was chosen for preliminary correlation of these data. The dimensional parameter, El/Γ, was actually used since Γ was unknown for the silicon oil.

6.2 Boiling and Condensation

The natural-convection studies with liquids have been extended to boiling with excellent results. As seen by the typical data of Choi in Fig. 36, the substantial effect of the field is reduced once boiling is initiated. There is no significant effect of the field on fully-developed pool boiling; however, the critical heat flux for freon is increased by

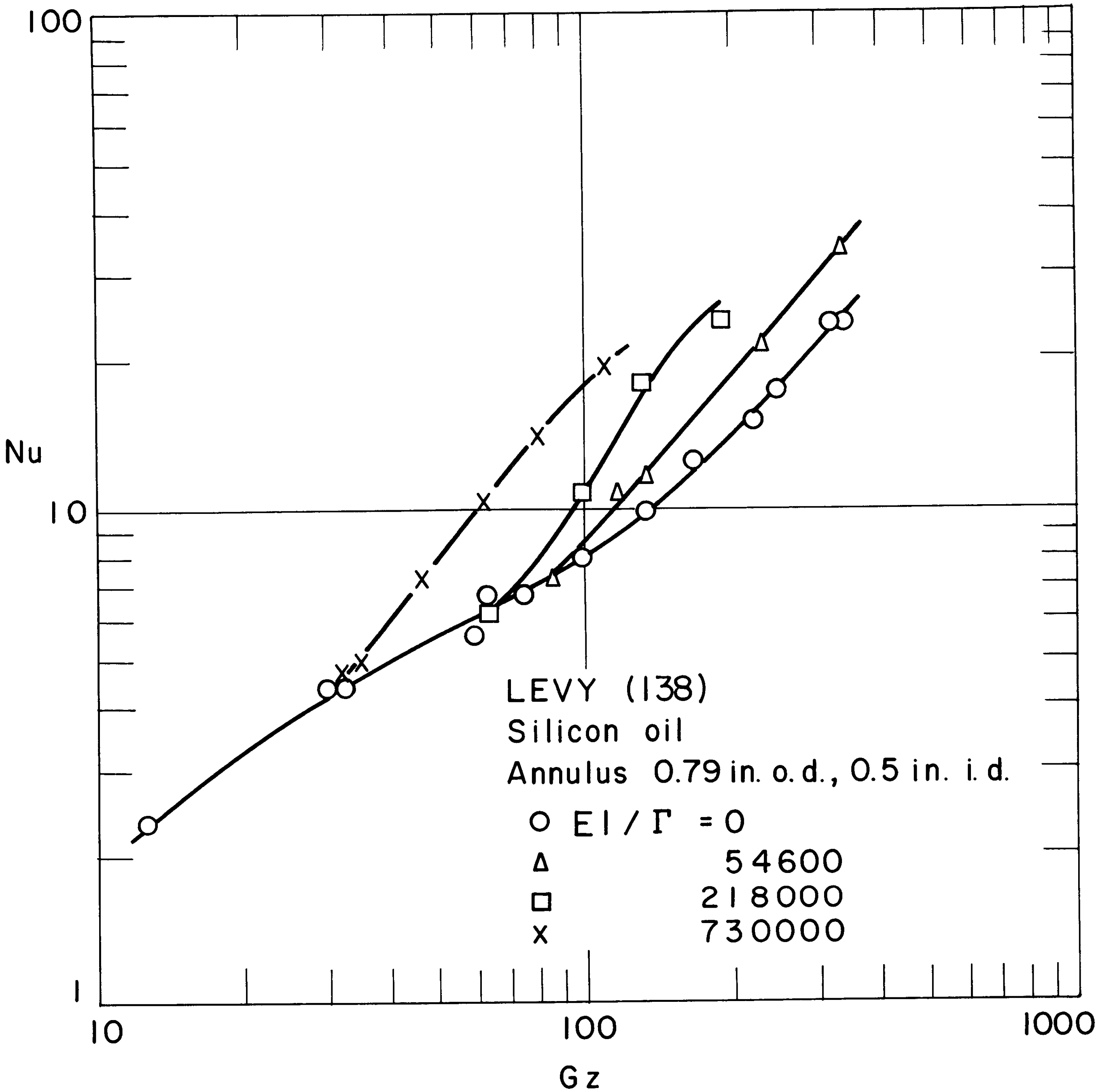

FIG. 37. EFFECT OF ELECTROSTATIC FIELDS ON LAMINAR FLOW HEAT TRANSFER

a factor of two. Durfee and Markels (139) utilized a 3/8-in-diameter heated section and charged (up to 10 kv) the can-like container. The critical heat flux for saturated pool boiling of isopropanol was increased by a factor of 6 when the maximum field was applied.

As noticed in Choi's results, film boiling is also greatly improved by application of the electrostatic field. Similar results were obtained by Bonjour, et al. (134) and Durfee and Markels (139). Since this latter study employed steam heating, the transition boiling region could be covered. It is interesting to note that this transition region virtually disappears at high values of the field strength.

In the three geometries used in these boiling studies, the electrostatic field strength increases as the heated surface is approached. Under these conditions the liquid tends to displace the vapor from the heated surface, thus accounting for the dramatic increases in pool-boiling burnout. The effective artificial gravity or buoyancy produced by the field has been considered for space applications. In addition to this dielectrophoretic-force effect, the condenser effect has been proposed to account for the vapor-film destabilization and attendant improvement in film boiling. This is reviewed in (139).

The usefulness of electrical fields has been greatly extended by the findings at Atlantic Research. Durfee and Markels (139, 140) reported that nucleate boiling, critical heat flux, and film boiling were all improved with deionized water in forced flow. As shown in Fig. 38, the critical heat flux was increased by over 100 percent for low-velocity flow in an annulus. The tests were restricted to low exit-quality conditions, presumably bubbly flow. One would expect that higher quality

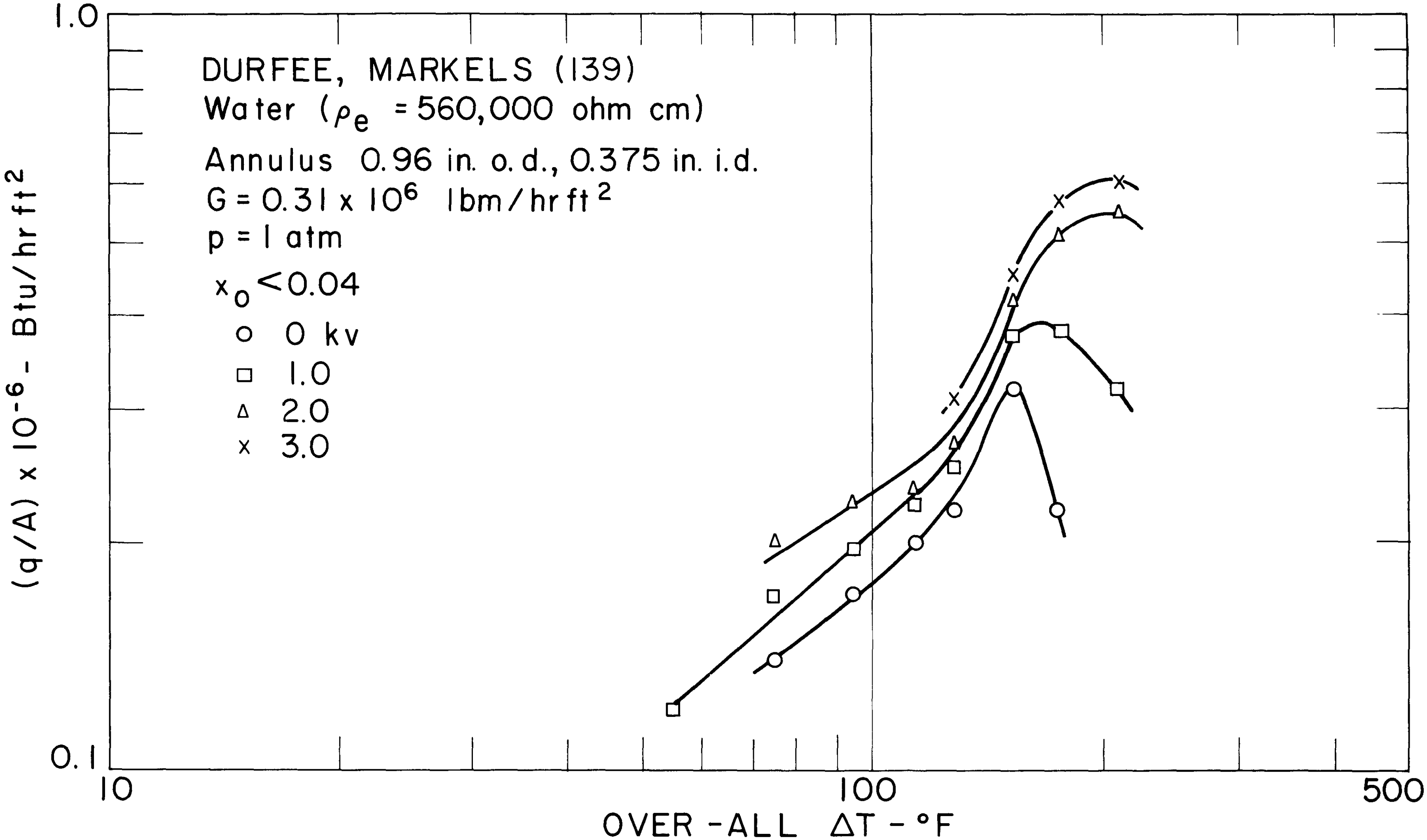

FIG. 38. INFLUENCE OF ELECTROSTATIC FIELDS ON FORCED-CONVECTION BOILING HEAT TRANSFER

conditions would also be improved since the electrostatic field would tend to stabilize the annular liquid film on the heated wall.

Preliminary results for burnout at higher pressures have been summarized in a recent report on this program (141). Improvements of about 20 percent in quality burnout were recorded. An operating-cost economic comparison, which included pumping power data, showed that the EHD system was slightly superior to a conventional system.

In addition to these investigations of heating, condensation experiments were reported by Velkoff and Miller (142). The electrostatic field was shown to increase laminar film condensation of freon on a vertical plane by as much as 300 percent. Experiments with condensing freon in an annular test section were reported by Choi (143). Condensation took place on the inner wall of a vertical tube with an inner, concentric electrode. Condensing coefficients were increased over 200 percent at field strengths over 30 kv/cm. The improvement was due largely to a film instability induced by the field. Certain aspects of adiabatic two-phase EHD flows were discussed by Reynolds (144). The investigations by both Choi and Reynolds will be summarized in a forthcoming ASD-TDR.

Several investigations of EHD phenomena are currently in progress at universities and industrial laboratories. There is still much work to be done before the effects of system geometry, type and strength of applied voltage, and fluid properties are well established. One can foresee applications of this augmentative scheme in specialized types of convective heat-transfer systems.

ADDITIVES

7.1 Liquid Systems

Under certain circumstances, small amounts of an addition agent have produced substantial improvements in heat transfer. The working fluid for a heat-transfer system is usually specified by the process or chosen on the basis of its desirable properties. An additive is then desired which will essentially preserve the desirable properties of the working fluid while still improving the heat transfer. There is also the possibility, of course, of being in a position to choose an optimum binary mixture, although this aspect will not be stressed here. The simplicity of this augmentative scheme has stimulated considerable research.

7.1.1 Boiling

Boiling appears to be the only area where addition agents are useful. As noted in the summary of Table VI, a great many additives have been investigated, and some have been found to exert a substantial influence on boiling heat transfer.

With the proper concentration of certain additives, increases of about 40 percent in the heat-transfer coefficient for saturated pool boiling can be realized. Specifically, increases in h of this order have been reported in (145-149). However, decreases in h for all concentrations studied were noted in (150-155).

The most important characteristic of the additives is their influence on saturated pool boiling burnout. Increases in burnout at certain additive concentrations are reported in (149-151, 153, 155-160). The

TABLE VI

EFFECT OF ADDITION AGENTS ON BOILING HEAT TRANSFER

Investigators	System	Results
Jakob, Linke (145)	Pool boiling Water/Nekal BX (0.5%)	Increase in h of 23%
Insinger, Bliss (146)	Pool boiling Water/Triton W-30 (0.2%)	Increase in h of 20%
Morgan, Bromley, Wilke (147)	Pool boiling Water/wetting agents	Maximum increase in h of 40%
Averin, Kruzhilin (148)	Pool boiling Water/isoalcohol (2.5%) isoamyl	Increase in h of 28%. Burnout decreased by 18%
Lowery, Westwater (149)	Pool boiling Methanol/nonionic, cationic, anionic agents ($<$ 10%)	General increase in h for nucleate and film boiling, increases in burnout
Bonilla, Perry (150)	Pool boiling Water/ethanol, n-butanol, acetone (various %)	Decreases in h, but slight increase in burnout at low %
Vos, van Stralen (151)	Pool boiling Water/methylethylketone (various %)	Decrease in h, but 150% increase in burnout at low %
Benjamin, Westwater (152)	Pool boiling Water/ethylene glycol (various %)	Reduction in h at all %
Westwater, Dunskus (153)	Pool boiling Isopropanol/organic additives ($\leq$ 0.5%)	General decrease in nucleate-boiling h, increase in burnout and film boiling h
Sternling, Tichacek (154)	Pool boiling 14 binary mixtures (various %)	Nucleate-boiling h and burnout decreased at all %

Investigators	System	Results
Huber, Hoehne (155)	Pool boiling Diphenyl/benzine (various %)	At low %, reduction in h but increase in burnout of 100%
van Wijk, Vos, van Stralen (156)	Pool boiling Water/acetone, MEK, alcohols, ethylene glycol (various %); organic binaries	Increase in burnout with all mixtures at low concentrations
van Stralen (157)	Pool boiling Water/alcohols, acetone MEK, ammonia (various %)	Increase in burnout with all mixtures at low % Pressure effect shown also
Kutateladze (158)	Pool boiling Water/ethanol (various %)	Increase in burnout at moderate %
Carne (159)	Pool boiling Water/organics (various %)	Moderate increases in burnout at low % with certain additives
Scarola (160)	Pool and forced-convection (tube) boiling/1-pentanol (2.2%)	Increases in saturated pool-boiling burnout, but decreases in subcooled burnout
Kreith, Summerfield (161)	Forced-convection surface boiling (tube) aniline/water (5%)	Improvements in boiling h
Noel (162)	Forced-convection surface boiling (tube) Hydrazine/ethylenediamine (10%)	Decrease in boiling h
Rose, Gilles, Uhl (163)	Forced convection surface boiling (annulus) Water/alcohols (high %)	Decrease in both h and burnout
Leppert, Costello, Hoglund (164)	Forced-convection surface boiling (rod) Water/propanol, methanol (low %)	Increases in h at certain %

data of (152, 154) indicate reduction in the critical heat flux for all mixtures and concentrations. Most additives, then, improve burnout, but the concentration of the additive is extremely important. Typical results of van Stralen, et al. (156, 157) as shown in Fig. 39 indicate a sharp increase in the critical heat flux at some low concentration and rather rapid decrease as the concentration is increased. The optimum concentration varies with the mixture and to some extent with the pressure.

Additional references on additive investigations are noted in the extensive report by van Wijk, et al. (156). In a more recent survey report, Metzler (165) comments on the diverse effects of additives and the rather conflicting theories which have been advanced to explain these effects.

The rather spectacular increases (over 200 percent with 1-pentanol) reported by van Wijk and co-workers (156) have been shown to be largely a function of heater size. Both Bernath (20) and Kutateladze (158) noted that the maximum improvement was less pronounced for large heaters, and in a recent study, Carne (159) clearly showed the importance of geometry. For a similar water-pentanol system, Carne obtained an increase of only 25 percent in burnout with a 1/8-in. heater as opposed to the 240 percent increase that van Stralen got with a 0.008-in. heater (Fig. 39). With practical size heaters, certain additives offer no improvement at all.

Subcooled pool boiling was considered by Scarola (160). As shown in Fig. 40 at low subcooling there was a considerable increase in burnout with 2.2 percent by weight of 1-pentanol (approximate optimum concentration of (156)). At higher subcooling, however, the additive

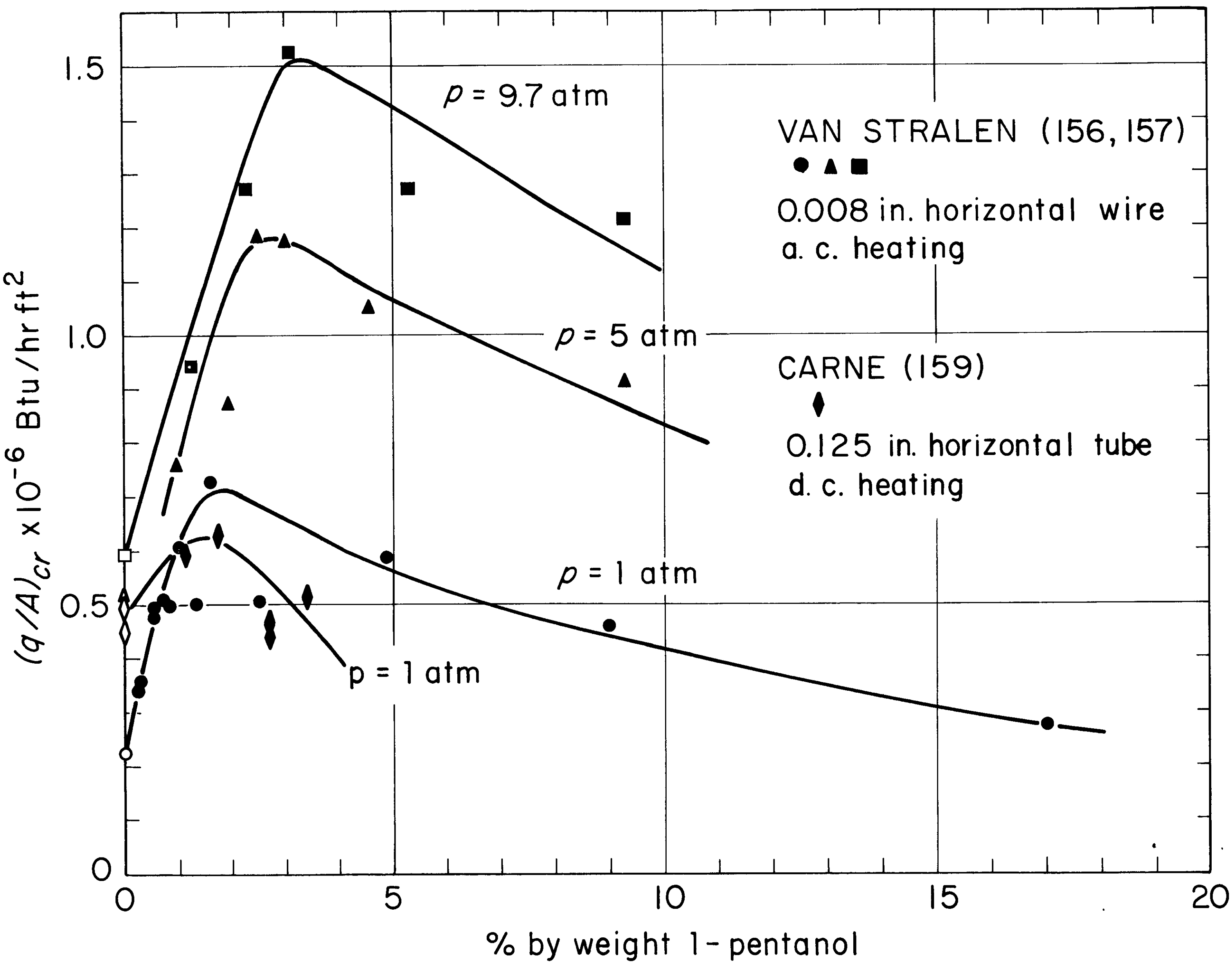

FIG. 39. CRITICAL HEAT FLUX VARIATION WITH ADDITIVE CONCENTRATION

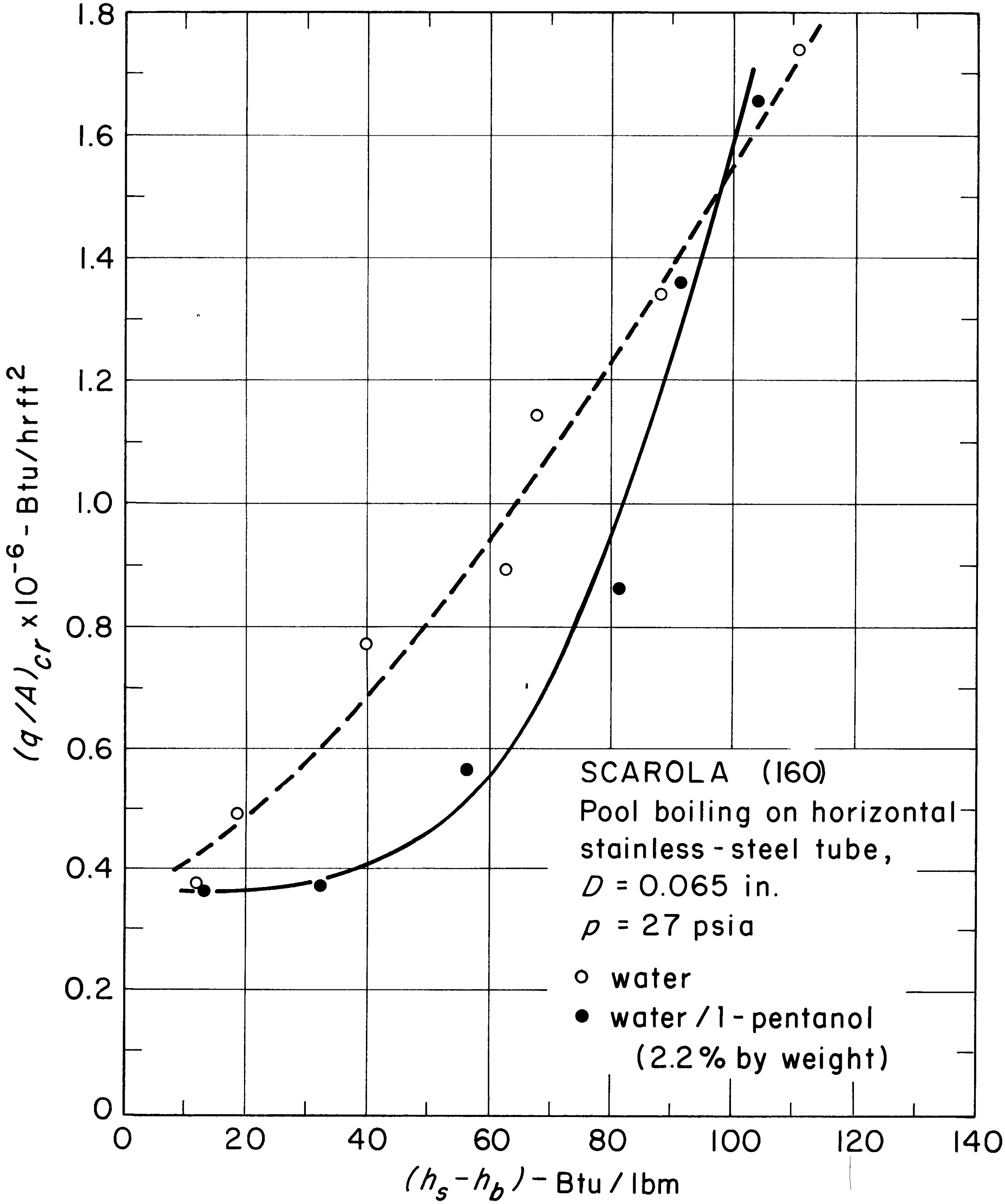

FIG. 40. INFLUENCE OF ADDITION OF 1-PENTANOL ON CRITICAL HEAT FLUX FOR SUBCOOLED POOL BOILING

produced a decrease in burnout. At very high subcooling the curves for normal and treated water appear to come together.

Several investigators have considered the more practical case of forced-convection surface boiling. In the course of their experiments with commercial-grade aniline, Kreith and Summerfield (161) noted incipient boiling, and even fully developed boiling, at wall temperatures well below the saturation temperature of pure aniline. This was attributed to the small percentage of water in the commercial product. Noel (162) found that h decreased when ethylene-diamine was added to hydrazine. Rose, Gilles, and Uhl (163) reported decreases in both heat-transfer coefficient and burnout when relatively large percentages of alcohol were added to water. Leppert, Costello, and Hoglund (164) took considerable heat-transfer data for water with small percentages of propanol or methanol. They found that the curve for fully developed surface boiling shifted to the right or to the left depending on the alcohol concentration. The maximum improvement in heat transfer was modest, however, and furthermore, the burnout flux was essentially constant. The main advantage of their binary systems appeared to be the improvement in smoothness in boiling. The reduced surface tension of the mixture caused an appreciable decrease in bubble size, thus the vapor formation was quite steady and smooth.

Burnout with forced-convection surface boiling at low pressures has also been investigated by Scarola. Typical results are given in Fig. 41. At low subcooling there is a distinct reduction in the burnout heat flux with the addition of 1-pentanol. It appears that the smaller bubbles are a disadvantage under conditions of low pressure and subcooling.

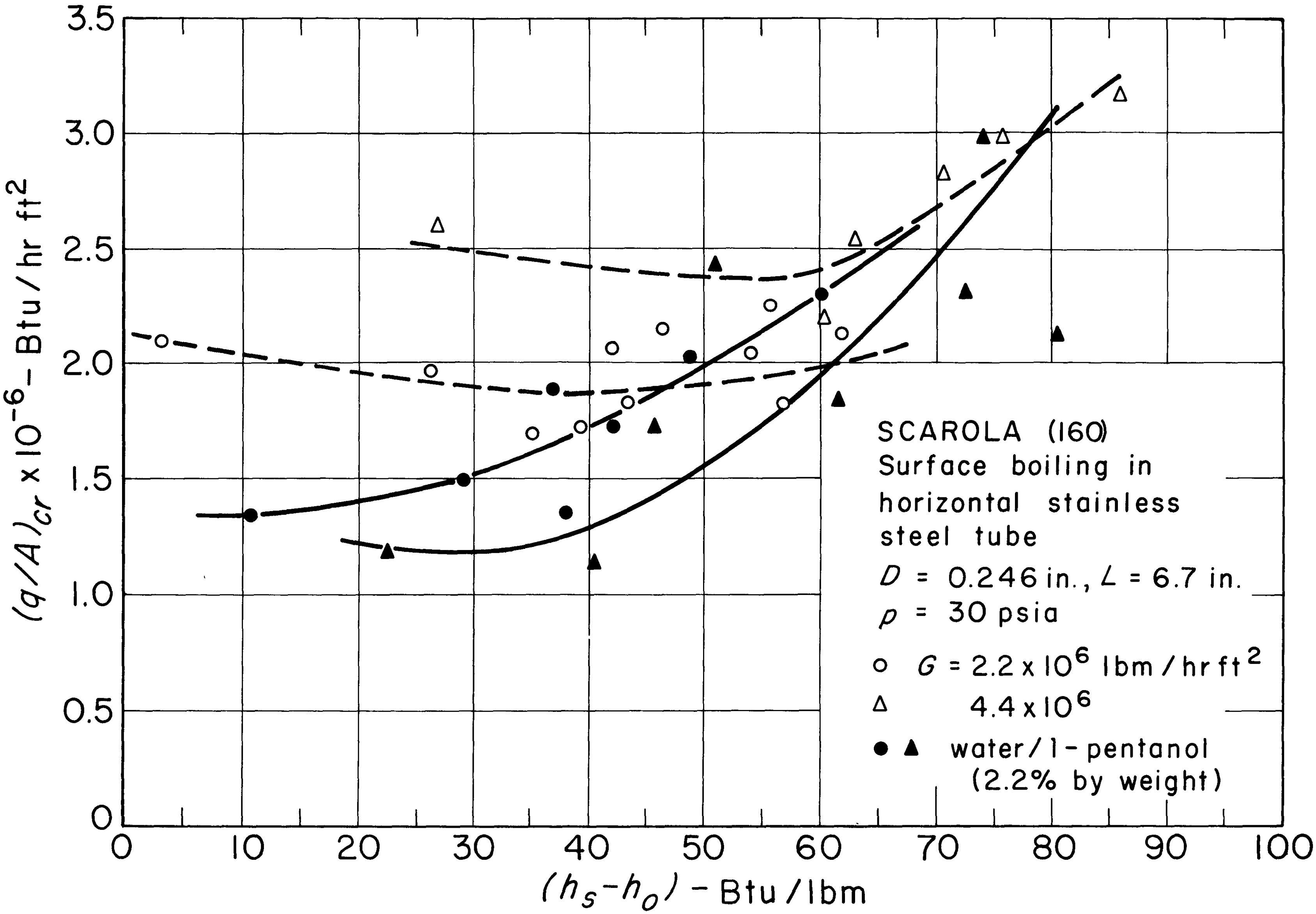

FIG. 41. INFLUENCE OF ADDITION OF 1-PENTANOL ON BURNOUT WITH FORCED-CONVECTION SURFACE BOILING

Normally the large nonequilibrium vapor volume increases the velocity, thereby preventing vapor blanketing at the surface. When the voids are reduced, this beneficial effect is less and burnout is lower. At higher subcooling, however, there is consistent indication that burnout may be slightly improved by addition of the volatile additive. At high subcooling, the void fraction is low, and the smoother boiling would be expected to be of some benefit.

In general the improvements in heat transfer and burnout offered by additives are not sufficient to make them useful for practical systems. There are difficulties involved in maintaining the desired concentration, particularly when the additive is volatile. In many cases the additives, even in small concentrations, are somewhat corrosive and require special piping or seals.

7.1.2 Condensation

Dropwise condensation can be enhanced by the addition of promoters to the vapor. The promoter condenses out and forms a non-wetting film on the surface. As in the case of promoters which are applied directly to the surface, there is a tendency for the layer to wash off, and the injection must be repeated at regular intervals. An extensive survey and investigation of this subject has been reported by Osmet and Tanner (166) and Osmet (167).

7.2 Gas-Solid Suspensions

The discussion of the additive technique would be incomplete without mention of solids added to a gas stream. During the past twenty years there has been a great deal of research in the area of two-phase gas-solid flow. The combined flow of gas and solids is important in such industrial processes as gas cleaning, pneumatic transport, combustion, and operation of fluidized beds. Momentum transfer is of particular interest in the first two processes, whereas heat transfer is the primary concern for the latter operations. However, when gas-solid suspensions were suggested as coolants for gas turbine and nuclear reactor systems some ten years ago, there was relatively little design information which could be obtained from these studies. Zenz and Othmer (168) indicate how this dilute-phase transport relates to relatively dense phase fluidized beds.

Solid particles in the micron to millimeter size range are dispersed in the gas stream at loading ratios w_s/w_g ranging from 1 to 15. The solid particles, in addition to giving the mixture a higher heat capacity, are highly effective in promoting enthalpy transport near the heat-exchange surface. Heat transfer is further enhanced at high temperatures by means of the particle-surface radiation.

The first experimental work appears to have been carried out by Schluderberg (169) who demonstrated the effectiveness and feasibility of nitrogen-graphite suspensions as a coolant. An extensive investigation was subsequently undertaken at Babcock and Wilcox to obtain detailed heat-transfer and pressure-drop information as well as operating experience for the suspensions. Heat-transfer coefficients for heating were improved by as much as a factor of ten through the addition of graphite. The suspensions were also shown to be far superior to gas coolants on the basis of pumping power requirements, especially when twisted tape inserts were used. There was relatively little settling, plugging, or erosion in the system. With helium suspensions, however, there was serious fouling of the loop coolers which was attributed to Brownian particle motion due to the temperature gradient. This effect is characteristic only of microscopic particles; however, graphite is generally reduced to micron-size particles after circulating for a short time.

The results of this investigation were reported in a series of reports, the final ones being (170) and (171). Summary articles by Rhode, et al. (172) and Schluderberg, et al. (173) elaborate on the conclusions of this work.

Soviet researchers also reported improvements of up to 400 percent in heat transfer with suspensions of 0.15 to 2.08 mm graphite particles (174). They found that the best heat transfer was obtained with the smallest particles although it is not clear how they were able to keep the particles from pulverizing to micron size.

In a more recent study, Abel and co-workers (175) demonstrated that the cold-surface deposition is a very serious problem with micronized graphite. This occurred with both helium and nitrogen suspensions and could be alleviated only with very high gas velocities. An economic comparison was presented in terms of a system pumping power - heat transfer rate ratio versus gas flow rate. This comparison indicated that the pure gas was generally more effective than the suspension at both low and high gas flow rates. In all probability the loop heater is very effective; however, this gain is offset by the low performance of the cooler. Due to the great difference in properties of the suspension and pure gas coolants, it is necessary to consider entire systems when evaluating the performance.

Graphite has been the most popular material for suspensions due to possible reactor application; however, ceramic particles have also been studied at some length. Farber and Morley (176) reported improvements of 300 percent in heat-transfer coefficient with air and particles of aluminasilica catalyst where $w_s/w_g = 7$. Similar experiments with once-through heated systems were reported for an air-lead and an air-glass system by Tien and Quan (177) and for an air-glass system by Depew and Farber (178, 179). The size of these particles can be optimized (approximately 30μ according to (178)) and maintained due to their hardness. No deposition has been reported; however, there is usually severe erosion of system components.

An analysis performed by Tien (180) was successful in predicting the heat-transfer characteristics of suspensions at loading ratios less than one. However, the model was shown to be inaccurate for the higher loading ratios of practical interest.

Work is continuing on the practical application of gas-solid suspensions. A basic investigation of the fluid mechanics and heat transfer of gas-solid suspensions is currently underway at Rutgers University. This work has been presented in quarterly reports as well as the topical reports of Peskin and Dwyer (181), Peskin and Rin (182), and Chen (183). Investigations are also being carried out at laboratories in France and England. It appears probable that improved pumping systems will be developed, and the deposition problem will be solved in the near future.

8. SUMMARY

This study has presented a survey and evaluation of the numerous techniques which have been shown to augment convective heat transfer. It is believed that the survey of experimental investigations is reasonably complete through 1964. The augmentative area is, however, of such current importance that no survey can hope to be complete, especially since a great deal of report literature is involved. The survey has included a brief description of each experiment as well as a summary of the results. The equal-pumping-power performance criterion has also been applied to representative data for turbulence promoters.

Surface roughness elements of both the integral and attached types are effective in improving nonboiling heat transfer inside tubes of various cross section. For many arrangements, the performance factor, $(h_r/h_o)_P$, approaches 2 for a spacing ratio, L/e, of the order of 10. With turbulence promoters in general, the best performance is obtained when operating in the transitional range of Reynolds numbers. Since the heat transfer and friction are strong functions of the type and size of roughness, the analogy results are of limited usefulness.

Surface material and finish can be varied to appreciably improve saturated pool boiling. Nucleate boiling coefficients can be increased by as much as a factor of four by providing artificial nucleation sites. However, surface condition does not appreciably affect the critical heat flux for pool boiling. Surface-boiling heat transfer can be improved slightly by varying surface material and finish. Machined roughness is effective in improving surface-boiling burnout, especially at low subcooling.

Sandblasting or machining the surface can raise the critical heat flux for bulk boiling by as much as 50-100 percent. Chemical treatment of the surface promotes the highly effective dropwise condensation.

It is generally desirable to take advantage of the fin-effect when installing surface promoters. The subject of extended surfaces was not covered in detail; however, typical internally finned tubes are shown to have favorable performance factors.

Displaced promoters consisting of axially located bluff bodies are not particularly effective in improving nonboiling heat transfer. However, bulk-boiling burnout can be improved by over 50 percent when protuberences are located on the unheated wall of an annulus.

Heat transfer can be significantly improved by any of the techniques which produce vortex flow in the heated section. Twisted-tape vortex generators appear to be best suited for most practical applications and result in significant improvement in nonboiling and boiling situations. The twisted tapes are much more effective for heating than for cooling. A comparison of numerous investigations indicates that the performance factors for nonboiling water are higher than those for air; however, there is considerable disagreement among investigations which cannot be explained simply in terms of geometrical and flow considerations. Burnout fluxes for forced-convection surface boiling with twisted tapes can be increased 100 percent over empty-tube values at comparable pumping power. Similar results have been obtained for bulk-boiling burnout.

Substantial improvement can be realized when vibration is applied either directly to the heated surface or to the fluid near the heated surface. With natural convection, numerous investigators have obtained improvements of several hundred percent with surface vibration. At high vibrational intensities the data can be described quite effectively in terms of a vibrational Reynolds number. Surface vibration is rather difficult to apply to forced-flow systems; however, with certain arrangements improvements of similar order can be obtained. The available data are inconclusive regarding the effect of surface vibration on boiling heat transfer.

Acoustic vibrations are very effective in augmenting natural-convection heat transfer to gases. When any appreciable forced flow is present, however, the influence of vibration is quite small. When loudspeakers are installed at the inlet to tubes, moderate improvements in heat transfer appear to be due to turbulence triggering. The results for vibration with liquid systems are more complex due to the frequent occurrence of cavitation. Both nonboiling and boiling heat transfer can be improved as much as 100 percent with pool systems. Little improvement appears possible with forced-convection systems because of the attenuation of the vibrational intensity due to remote transducer placement.

Vibrational techniques appear to have limited practical application. Elaborate equipment is usually required to produce the vibration, and in most cases excessive vibrator power is required to obtain a relatively small improvement in heat transfer.

When electrostatic fields are applied to dielectric fluids, considerable improvement in heat transfer can be realized. Natural-convection heat transfer to liquids can be increased by several hundred percent with suitable field orientation. Forced-convection data have been taken only for laminar flow where improvements of over a hundred percent have been recorded. The critical heat flux for saturated pool boiling can be elevated by as much as 600 percent with EHD. Preliminary experiments indicate that improvements in bulk-boiling burnout at higher qualities appear to be too small to be economically justified. Laminar film condensation can be substantially improved with the proper field orientation. Further tests will be required to establish the applicability of EHD. In particular, the high voltages which are necessary will be a serious problem for many applications.

Small amounts of certain addition agents, particularly when added to water, can produce some improvement in nucleate boiling heat transfer. As a result of recent experiments it appears that the improvements in saturated pool boiling burnout are small for practical-size geometries, whereas burnout in forced-convection surface boiling is adversely affected by additives. Dropwise condensation can be promoted by agents which condense out and form a nonwetting film on the surface. Suspended solid particles greatly improve the heat capacity and heat-transfer coefficient of a gaseous coolant. However, improved pumping systems and a solution of the deposition problem will be required before this technique can be successfully applied.

It is evident, then, that most types of convective heat transfer can be improved by a variety of augmentative techniques. The present report will serve as a guide to these techniques and the experimental evidence. Augmentative techniques have now been tried on enough cases of practical interest that it should be possible to decide whether a particular system can be economically improved. It appears that many of these schemes are very impressive on an experimental basis and will be eventually considered seriously for practical applications.

APPENDIX

COMPUTATION OF COMPARISON FOR NONBOILING FORCED CONVECTION

Data for friction factor and Nusselt numbers as a function of Reynolds number, Prandtl number, and promoter geometry are given in graphical or tabular form.

A value of Re_a is chosen and f_a and h_a (or Nu_a) are noted. Assuming no change in fluid properties and constant channel geometry, the heat-transfer performance for either equal pressure drop or equal pumping power can be calculated. For equal pressure drop, since

$$\Delta p \sim f\, V^2 , \tag{A-1}$$

$$Re_o/Re_a = V_o/V_a = (f_a/f_o)^{0.5} . \tag{A-2}$$

Assuming, for example,

$$f_o = 0.184/Re_o^{\,0.2} , \tag{A-3}$$

$$Re_o = (Re_a^{\,2}\, f_a\, 5.37)^{0.556} . \tag{A-4}$$

Using, for example, the standard Dittus and Boelter-McAdams relation,

$$Nu_o = 0.023\, Re_o^{\,0.8}\, Pr^{0.4} , \tag{A-5}$$

the desired ratio

$$(h_a/h_o)_{\Delta p} = (Nu_a/Nu_o)_{\Delta p} \tag{A-6}$$

can be calculated.

Similarly for equal pumping power,

$$P \sim fV^3 , \tag{A-7}$$

$$Re_o = (Re_a^{\,3}\, f_a\, 5.37)^{0.358} , \tag{A-8}$$

and $(h_a/h_o)_P$ can be calculated.

For a given data point, it is noted that $(Re_o)_{\Delta p} > (Re_o)_P$ so that $(Nu_o)_{\Delta p} > (Nu_o)_P$ and $(Nu_a/Nu_o)_P > (Nu_a/Nu_o)_{\Delta p}$.

It is necessary to examine the empty or smooth-tube data for each study to determine the proper correlations for f_o and Nu_o. Different fluids and different heat-flux conditions usually require some corrections to the conventional correlations. It is noted, however, that friction data are often taken under isothermal conditions, so it is only consistent to treat the smooth-tube friction data on the same basis.

Since $V_a < V_o$, it is evident that for the same q the fluid temperature for the augmented case will be less than that for the unaugmented case. The effect on the comparison is usually small, however, and has been neglected in the present analysis.

Most experimental data are reported for test sections of considerable length so that entrance effects are not important. It is noted, however, that length has not been considered as a variable in the present comparison. The results would thus be in error for very short test sections where the smooth tube, more so than the augmented tube, would have a higher heat-transfer coefficient. It is noted that swirl-flow entrance and exit losses, in particular, can be quite large; however, few data are available for the estimation of these losses.

In order to limit the computation time, only 5 or 6 values of Re_a, covering the experimental range, were chosen. Unless tabular data were available, convenient values of Re_a were used, and the experimental data were interpolated. These results were plotted as $(h_a/h_o)_P$ vs Re_o, although Re_a could have equally well been chosen. Smooth curves were drawn through the calculated results.

REPORT CODE FOR REFERENCES

A/CONF	United Nations Geneva Conference Proceedings
AAEC	Australian Atomic Energy Commission
AEEW	Atomic Energy Establishment Winfrith (United Kingdom)
AERE	Atomic Energy Research Establishment (United Kingdom)
ARL	Aeronautical Research Laboratories
ASD	Aeronautical Systems Division
ATL	Advanced Technology Laboratories
DP	E. I. Dupont de Nemours & Co.
EURAEC	European Atomic Energy Community
GEAP	General Electric Atomic Power (Equipment Department)
JPL	Jet Propulsion Laboratory, California Institute of Technology
MIT	Massachusetts Institute of Technology
NAA	North American Aviation
NACA	National Advisory Committee for Aeronautics
NYO	New York Operations Office, AEC
ORNL	Oak Ridge National Laboratory
PWAC	Pratt & Whitney Aircraft
RADC	Rome Air Development Center
RTD	Research and Technical Division (Air Force)
SNECMA	Societe National d'Etude et de Construction de Moteurs d'Aviation (France)
TID	Technical Information Service Extension, AEC
WADC	Wright Air Development Center

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