AE-70

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An Experimental Study of Pressure Gradients for Flow of Boiling Water in a Vertical Round Duct. (Part 2)

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AN EXPERIMENTAL STUDY OF PRESSURE GRADIENTS FOR FLOW OF BOILING WATER IN VERTICAL ROUND DUCTS.

(Part 2)

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Summary:

The present report contains the results of the second phase of an experimental investigation concerning frictional pressure gradients for the flow of boiling water in vertical channels. The test section for this phase consisted of an electric heated stainless steel tube of 3120 mm length and 7.76 mm inner diameter.

Data were obtained for pressures between 6 and 41 ata, steam qualities between 0 and 70 per cent, flow rates between 0.025 and 0.210 Kg/sec and surface heat flux between 30 and 91 W/cm^2.

The results are in excellent agreement with our earlier data for flow in a 9.93 mm inner diameter duct, which were presented in report AE-69.

From the measurements we conclude that in the range investigated the non-dimensional pressure gradient ratio, φ^2, is independent of mass flow rate, inlet sub-cooling and surface heat flux.

On the basis of the measured pressure gradients, the following empirical equation has been established for engineering use.

$$\varphi^2 = 1 + 2400 \left(\frac{x}{p}\right)^{0.96}$$

This equation correlates our data (more than 1000 points) with a discrepancy of less than $\pm$ 15 per cent.

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This report is intended for publication in a periodical. References may not be published prior to such publication without the consent of the authors.

1.0. Introduction.

The present report contains recent data obtained as the second phase of an experimental study of pressure gradients for the flow of boiling water in vertical round ducts. For this phase the electrically heated test section consisted of a stainless steel tube with an inner diameter of 7.76 mm and a length of 3120 mm.

In an earlier report (AE-69) similar results were presented for a duct of the same length but with an inner diameter of 9.93 mm.

For both test sections the experimental techniques were almost identical, and with respect to descriptions of apparatus, experimental procedures and methods of computation these are therefore omitted in this report and the reader is referred to report AE-69.

One change, however, should be mentioned. This change was the introduction of a more accurate ($\pm$ 0.7 per cent) and reliable water flowmeter before the start of the 7.76 mm experiments. This flowmeter is described in a separate report (RPL-102).

The total experimental program also comprises test sections of 4 and 13 mm inner diameters. A detailed discussion of our data in relation to the literature on the subject will be given in a final report when the research program is fully completed. In this report only a brief discussion of the present data in relation to the 9.93 mm duct results will be presented.

2.0. Preliminary Runs

Before the ordinary two-phase flow runs were started the following measurements were made to check the accuracy of the experimental techniques.

1. One-phase flow friction coefficients
2. One-phase flow heat transfer
3. One-phase flow heat balances

2.1. One Phase Flow Friction Coefficients

The measured friction coefficients for flow of water are given in figure 2. The data are correlated by the equation

$$f = 0.229\ Re^{-0.219}$$

with a discrepancy of $\pm$ 1 per cent.

2.2. One Phase Flow Heat Transfer

Figure 3 shows the results of the heat transfer measurements for one phase flow of water. The solid line represents the well-known McAdams equation

$$Nu = 0.023\ Re^{0.8} \cdot Pr^{0.4}$$

Our data are on an average 7 per cent lower than this equation. The scatter around the mean value is for our data $\pm$ 5 per cent.

2.3. Heat Balance

The table below shows the results of the heat balance measurements, where the increase of water enthalpy through the test section was compared to the electric heat input.

Run	$\dot{m}$ kg/sec	Net Heat Input kJ/kg	Δi_{water} kJ/kg	Error %
1	0.2111	46.5	43.7	- 6.4
2	0.1429	44.0	43.8	- 0.5
3	0.1448	24.6	25.5	- 0.4
4	0.2440	25.6	25.5	- 0.4
5	0.2440	72.9	73.4	+ 0.7

Except for run 1 all heat balances were obtained with an accuracy better than 0.7 per cent.

2.4. Results of Preliminary Runs

The results of the one-phase flow measurements show a considerable improvement in accuracy compared to the one-phase flow data obtained with the 10 mm inner diameter duct. Especially the high accuracy of the measured heat balances should be pointed out, and we conclude that extremely accurate results may be expected for the two-phase flow measurements.

3.0. Range of Variables

Series of runs were made at pressures of approximately 6, 10, 20, 30 and 40 ata. At each of these pressures three series of runs were performed at heat fluxes of 30, 50 and 85 W/cm^2. One series was also obtained at 8 ata and 30 W/cm^2; and finally a few series were repeated in order to check the reproducibility of the results.

One test series normally consisted of 6 - 9 runs in the flow rate range from 0.016 kg/sec to 0.210 kg/sec. With these values of the mentioned parameters, exit steam qualities between 0 and 72 per cent were obtained.

The total experimental program for two-phase flow consisted of 154 runs.

4.0. Results and Discussions

In figures 4 to 8 the frictional pressure gradient ratio, φ^2, is plotted against the steam quality, x, with the pressure as parameter. The relatively large deviations from the mean pressures given in the diagrams are mainly caused by the axial pressure gradients in the test section. A substantial part of the observed discrepancies of the φ^2-values, especially at low static pressures, may be explained in the light of the difference in pressure for the different points.

Figure 9 shows a summary of the data presented in figures 4 to 8. φ^2-values for x = 1.0 computed by means of one-phase flow theory, are also given, as well as the results by Martinelli and Nelson (3).

In figures 10 to 24 our data are plotted according to the method by Lockhart and Martinelli, using φ^2 and X_{tt} as variables.

In our earlier report AE-69 we pointed out that some of the data, for example those in figures 28 and 29 of AE-69, indicated that the φ^2-values were possibly slightly dependent on the mass flow rate. The possibility that this indication was caused by small errors in flow-rate measurements was mentioned.

The present data, however, do not show this effect, and we conclude that the mass flow rate has no influence on the φ^2-values in the range investigated.

Figures 25, 26 and 27 show data obtained at constant heat flux but different pressures. Comparing these figures with figures 30, 31 and 32 of ref (1), almost identical pressure relationships are found for both ducts.

Figures 28 and 29 give the results from runs with different heat fluxes, but with almost constant pressure. As was the case for the 9.93 mm duct, no effect of surface heat flux has been observed in the range investigated.

5.0. Pressure Drop Correlation

Using the same method as for the 9.93 mm duct, a pressure drop correlation has been derived on the basis of the present data. Figure 30 shows a plot of the φ^2-values at X_{tt} = 1.0 against the static pressure. The solid line in the diagram is the correlation curve for these points and the dotted line the corresponding curve obtained for the 9.93 mm duct. From this figure and figure 39 in ref (1) the following empirical equation has been derived for the 7.76 mm duct.

$$\varphi^2 = 2400 \left(\frac{x}{p}\right)^{0.96} + 1 \qquad (1)$$

This equation corresponds to equation 16 in AE-69. It should be noticed that the present equation has been simplified as compared with equation 16 through the use of the same exponent for both pressure and quality.

In figure 31 some of the measured φ^2-values are plotted against the ratio x/p. The solid line represents equation 1, and the figure shows that equation 1 correlates our data within an accuracy of $\pm$ 15 per cent.

A plot of the measured φ^2-values against the values for φ_c^2 computed by means of equation 1 is given in figure 32.

Figure 33 shows a comparison between equation 1 and our data earlier presented in figures 4 to 8. The agreement is excellent in the total range investigated, except for the data at 6 and 10 ata and steam qualities greater than $\sim$ 0.50. Here the discrepancy increases from $\sim$ 20 per cent at x $\sim$ 0.50 to $\sim$ 35 per cent at x $\sim$ 0.70. Finally, the results of Sher and Green at 137 ata are also included in the diagram.

Figure 34 shows a comparison of equation 1 valid for the 7.76 mm duct and equation 16 of AE-69 valid for the 9.93 mm duct. The agreement is very good, and it is rather difficult to suggest whether the discrepancies that do exist are due to a diameter effect or to experimental errors.

A more detailed discussion of the data and the literature on the subject will be given in the final report covering all the diameters investigated.

6.0. Conclusions

Frictional pressure drops for the flow of boiling water in a round vertical duct of 7.76 mm inner diameter have been measured for a large range of variables.

In the limiting cases of $x = 0$ and $x = 1.0$, our data are in excellent agreement with one-phase flow theory.

In the two-phase flow region the present data are in excellent agreement with our earlier results obtained with a 9.93 mm duct, and our conclusions arrived at in AE-69 have been verified.

We conclude that in the range investigated inlet sub-cooling, surface heat flux and mass flow rate have no effect on the frictional pressure gradient ratio φ^2.

With regard to the diameter the difference in results for the two ducts is too small in order to conclude whether the difference is caused by a real diameter effect or by experimental errors.

Nomenclature

Symbol	Definition	Units
f	Friction coefficient	Dimensionless
Nu	Nusselt number	Dimensionless
$\dot{m}$	Mass flow rate	Kg/sec
p	Pressure	Kg/cm^2
Pr	Prandtl number	Dimensionless
Re	Reynolds number	Dimensionless
x	Steam quality	Kg/Kg
φ^2	Pressure gradient ratio	Dimensionless
X_{tt}	Parameter defined by eq. 12 ref. (1)	Dimensionless

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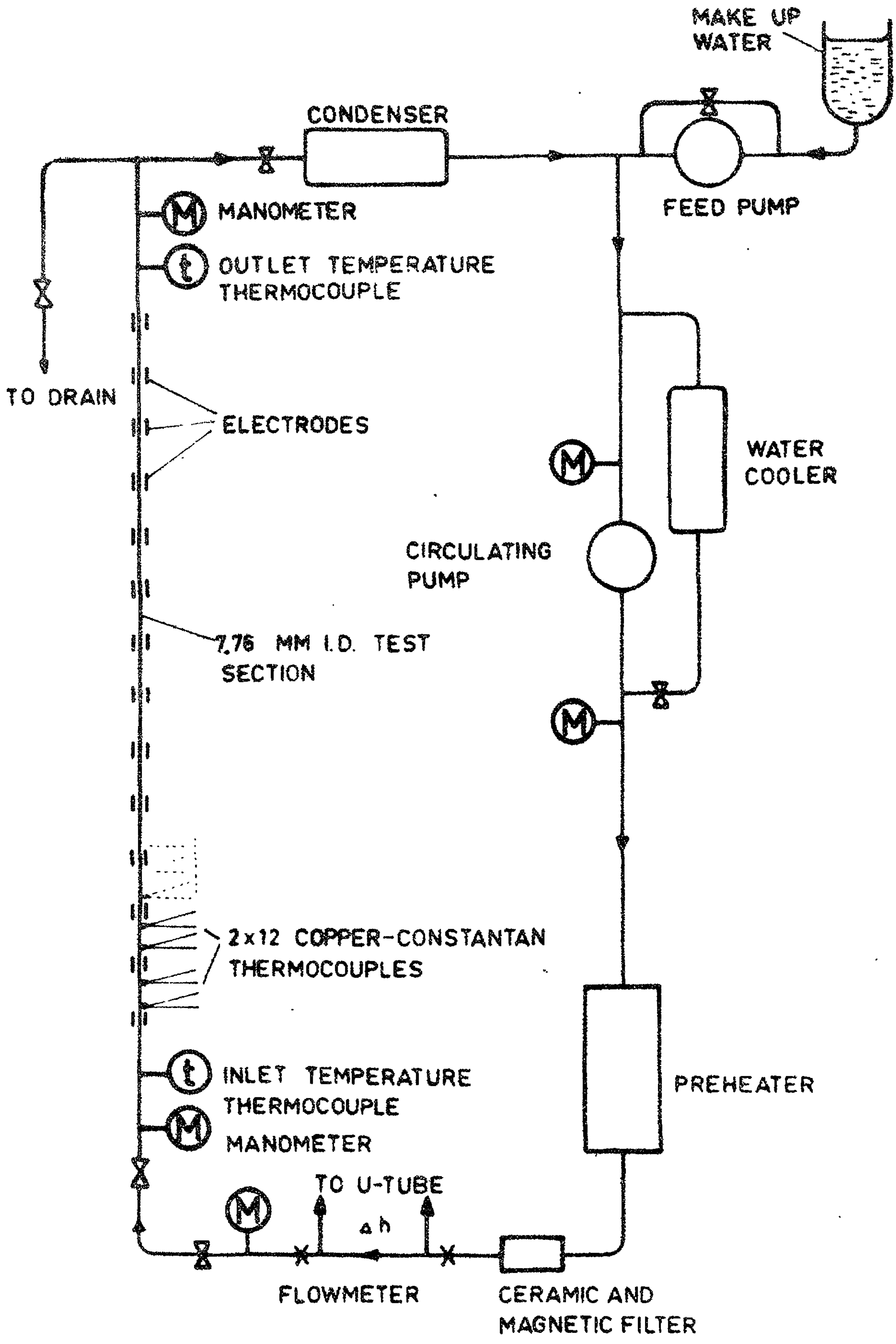

FIG. 1 FLOW DIAGRAM.

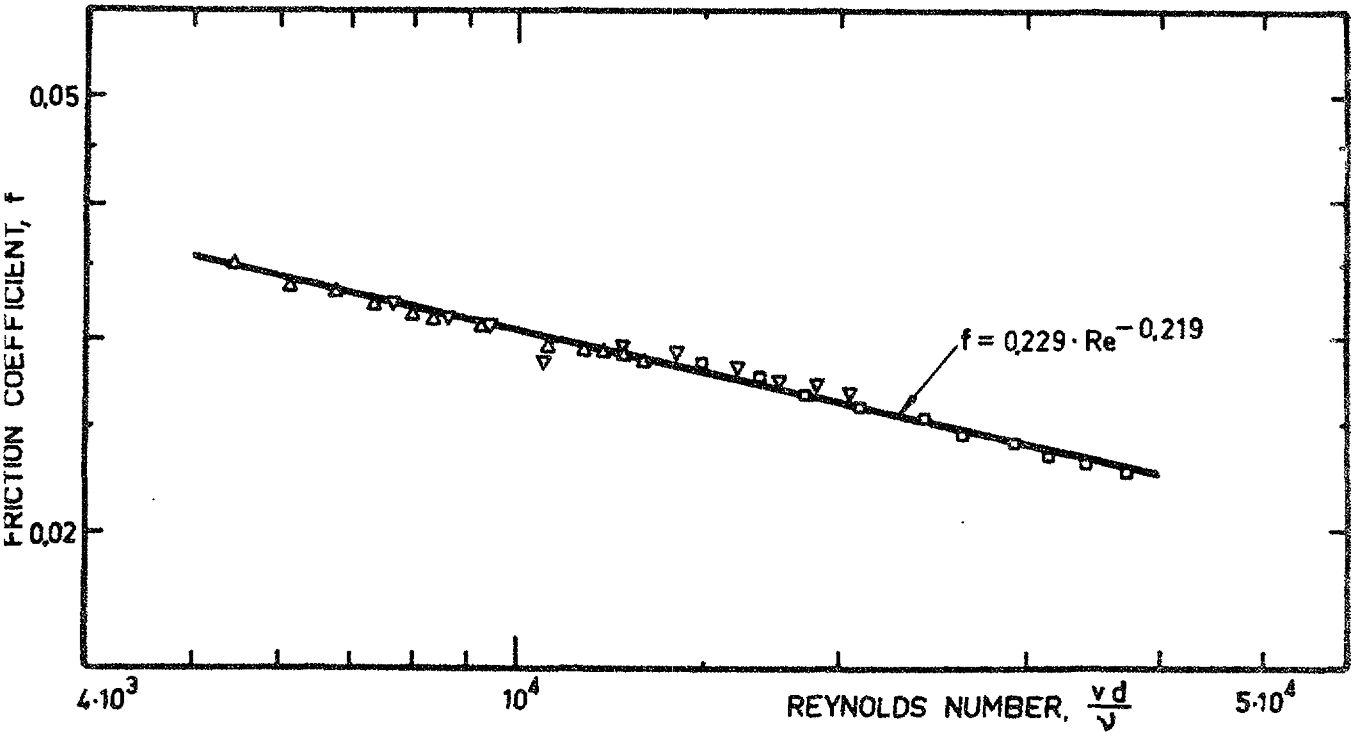

FIG. 2. FRICTION COEFFICIENTS FOR ISOTHERMAL FLOW OF WATER

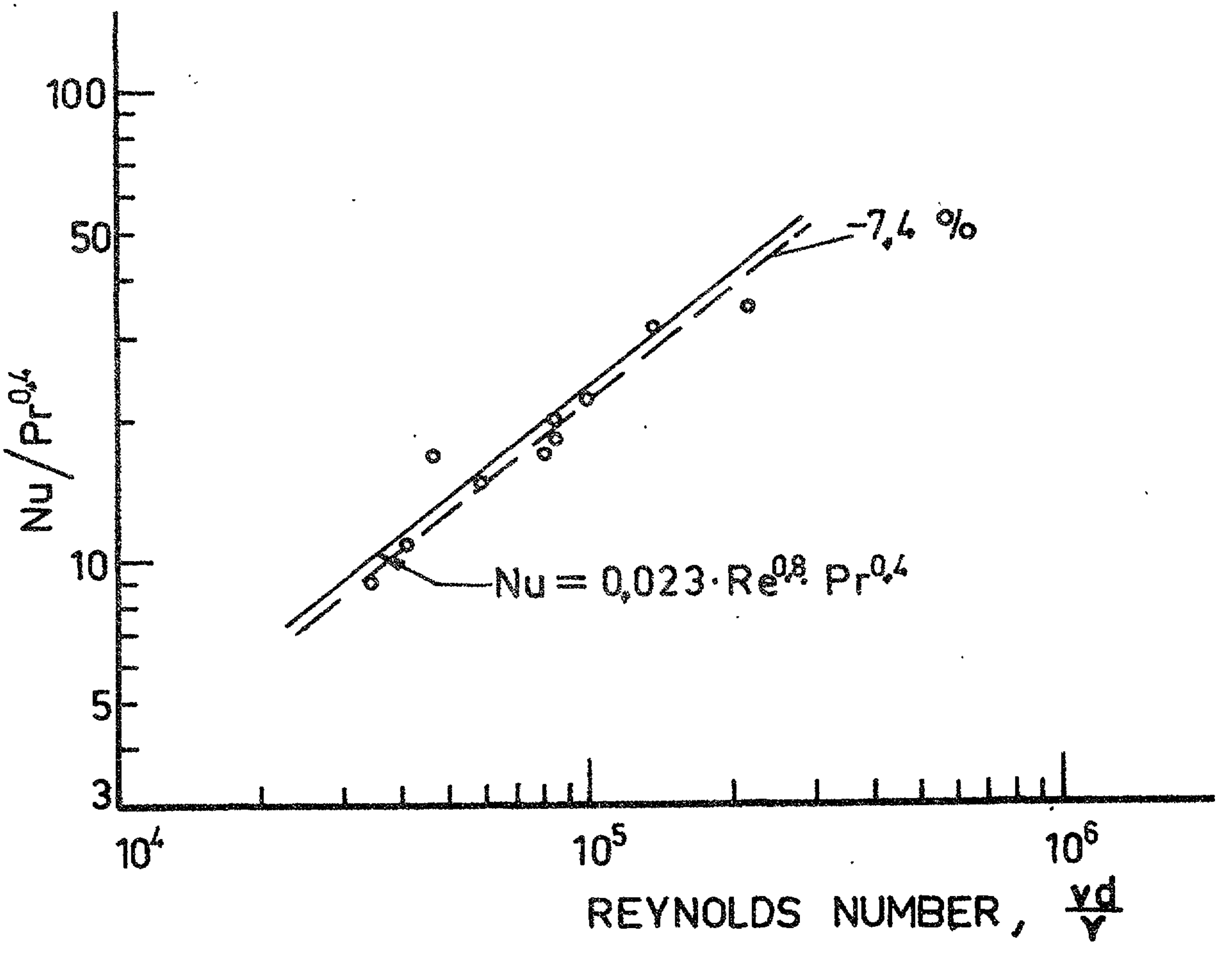

FIG. 3. HEAT TRANSFER RATES FOR ONE-PHASE FLOW OF WATER

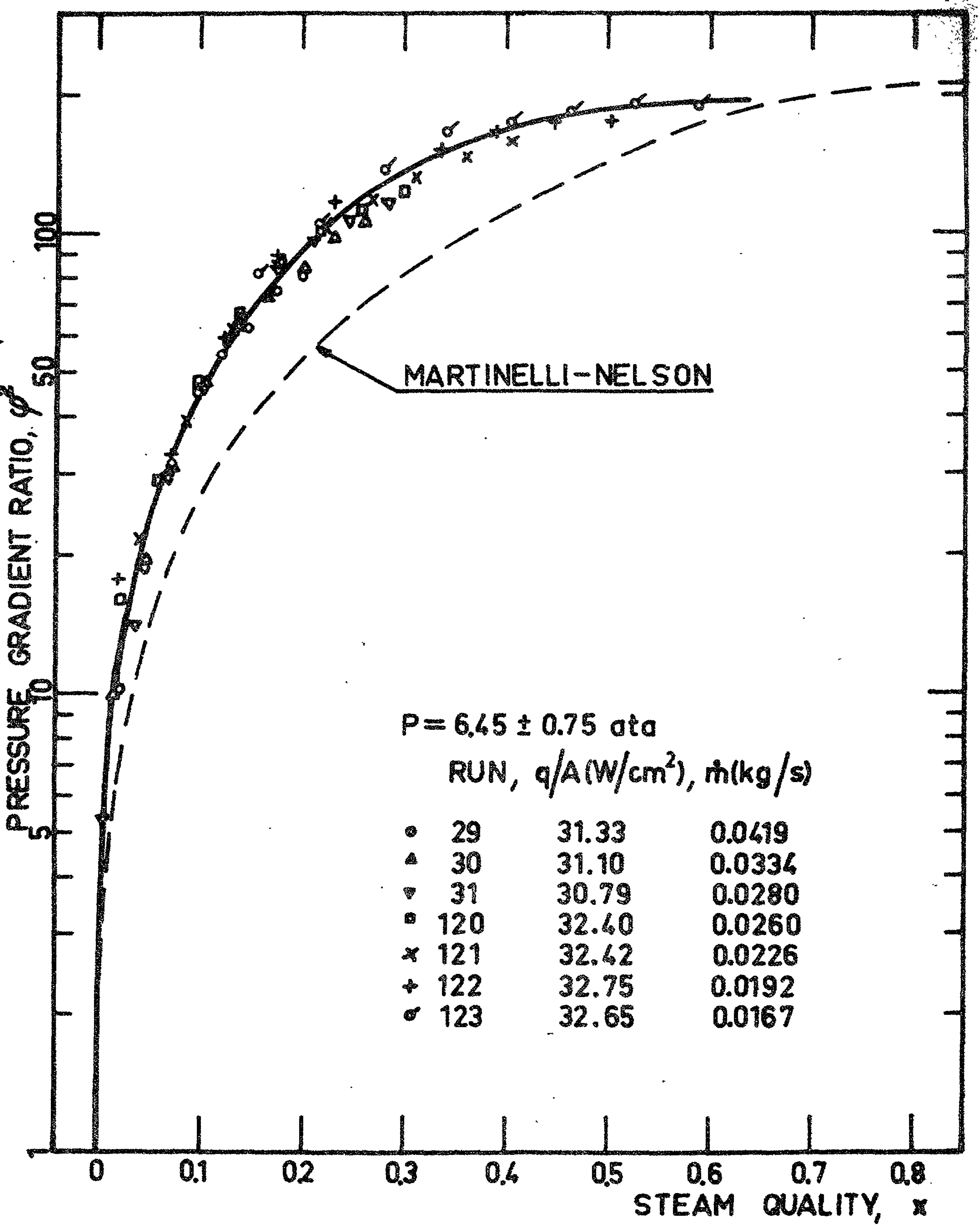

FIG. 4. MEASURED PRESSURE GRADIENT RATIOS

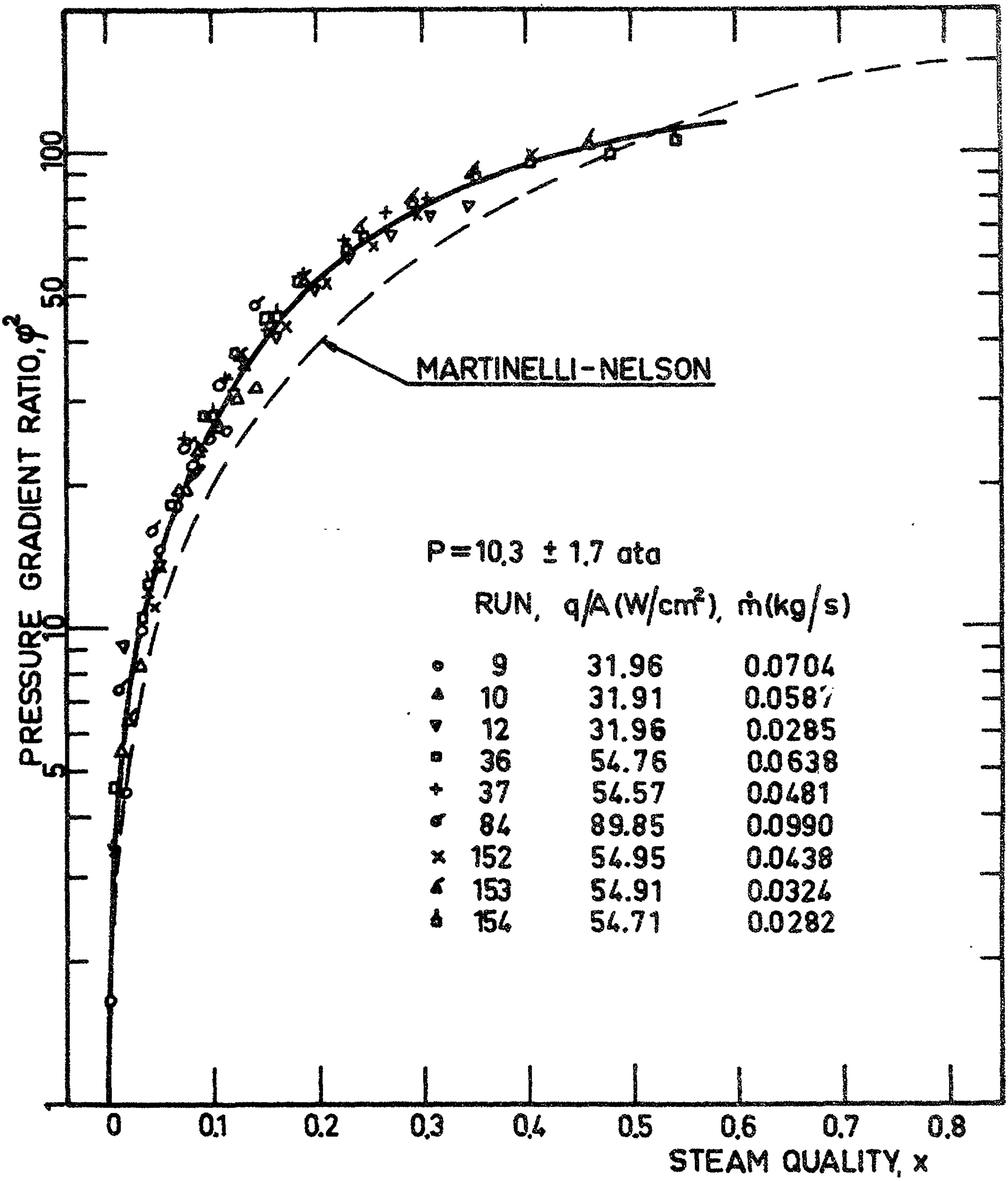

FIG. 5. MEASURED PRESSURE GRADIENT RATIOS

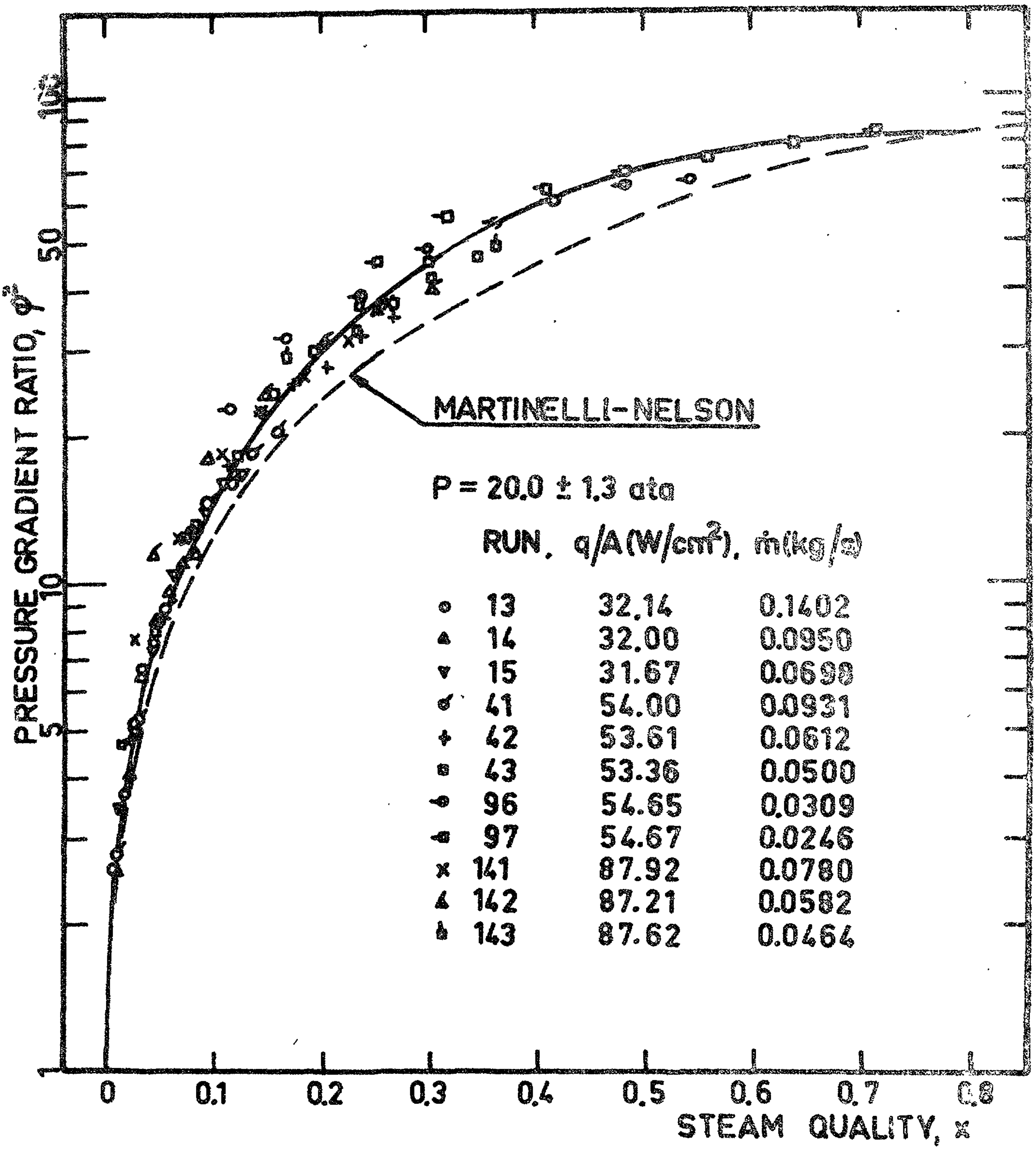

FIG. 6. MEASURED PRESSURE GRADIENT RATIOS

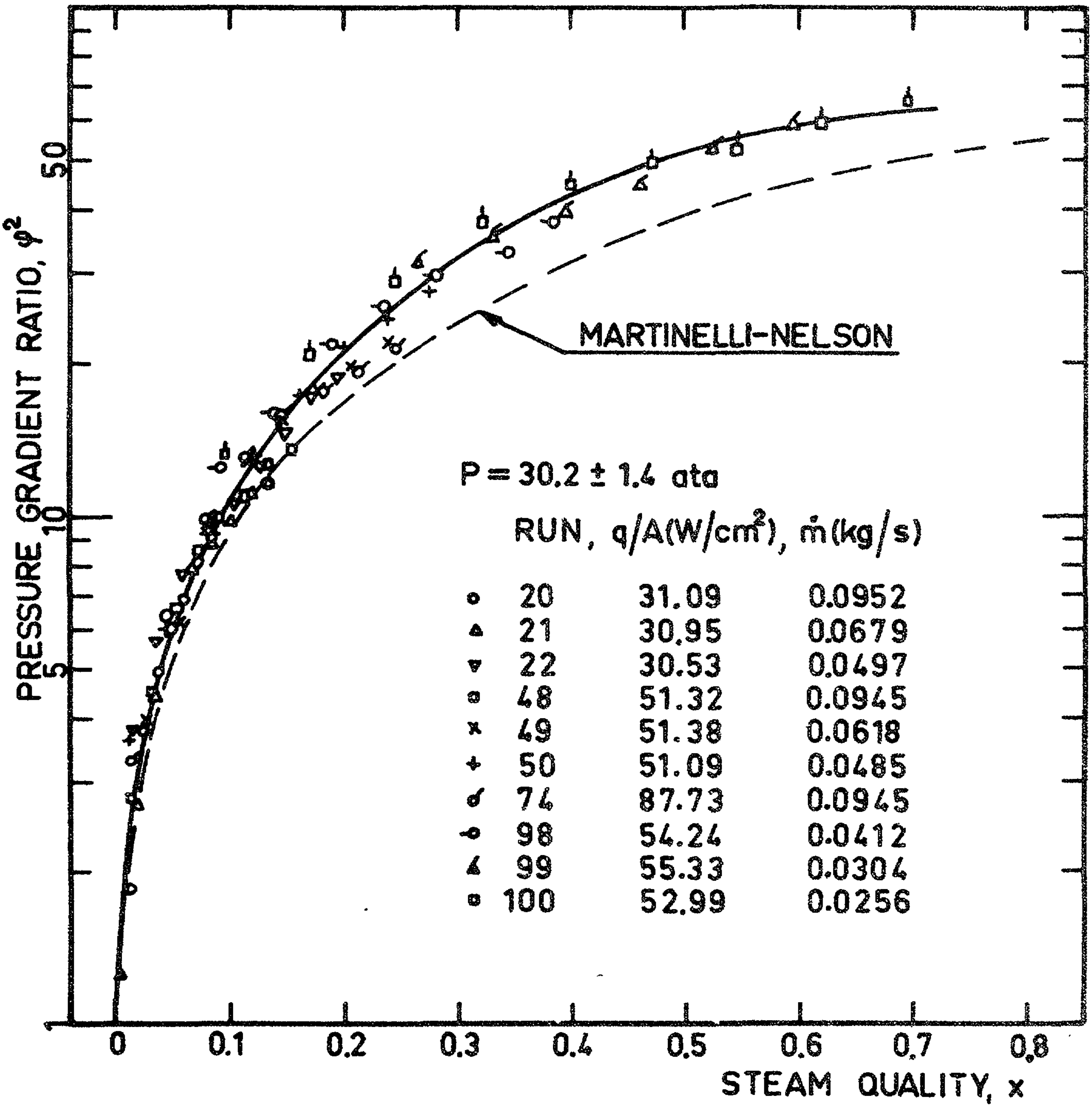

FIG. 7. MEASURED PRESSURE GRADIENT RATIOS

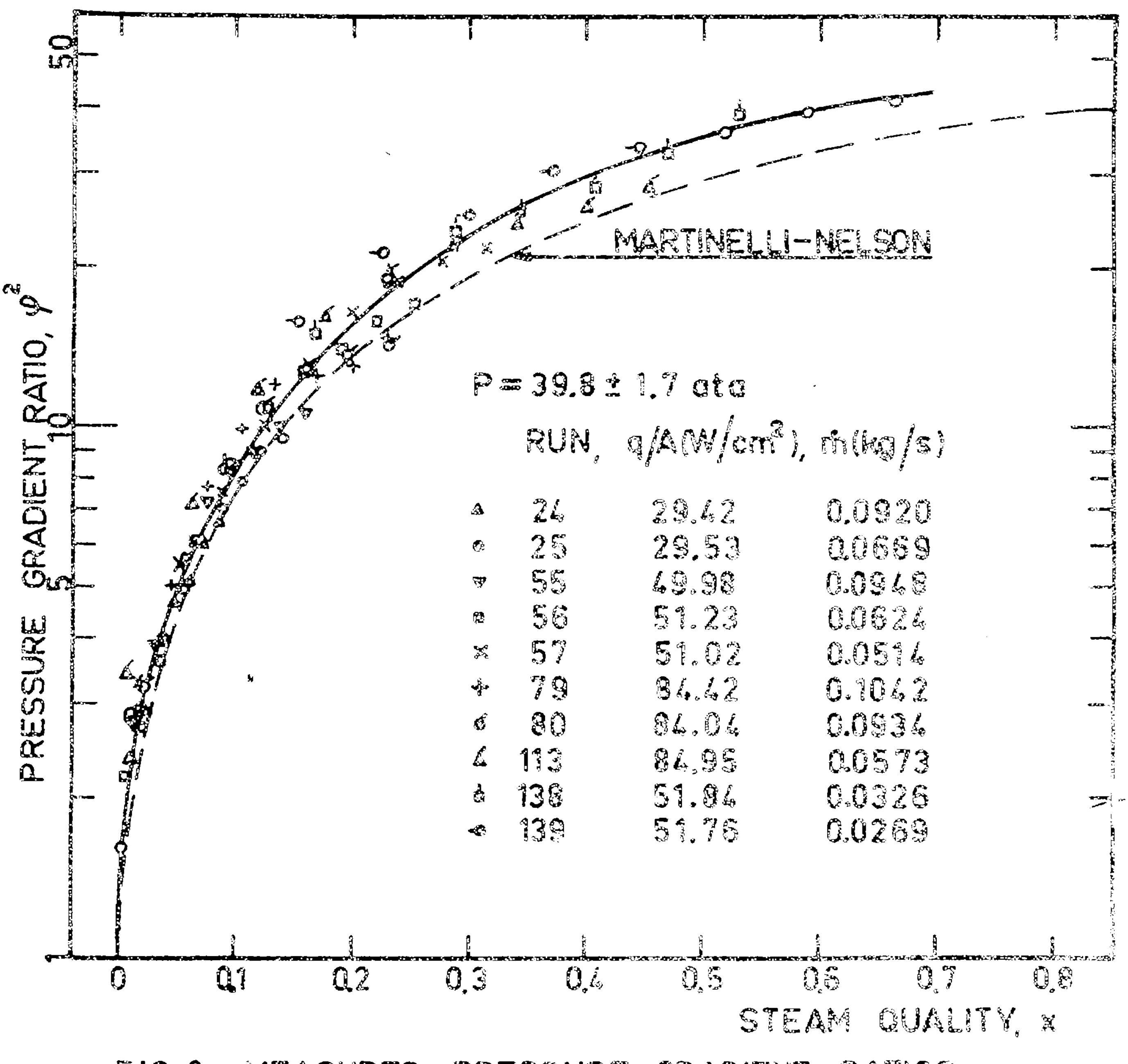

FIG. 8. MEASURED PRESSURE GRADIENT RATIOS

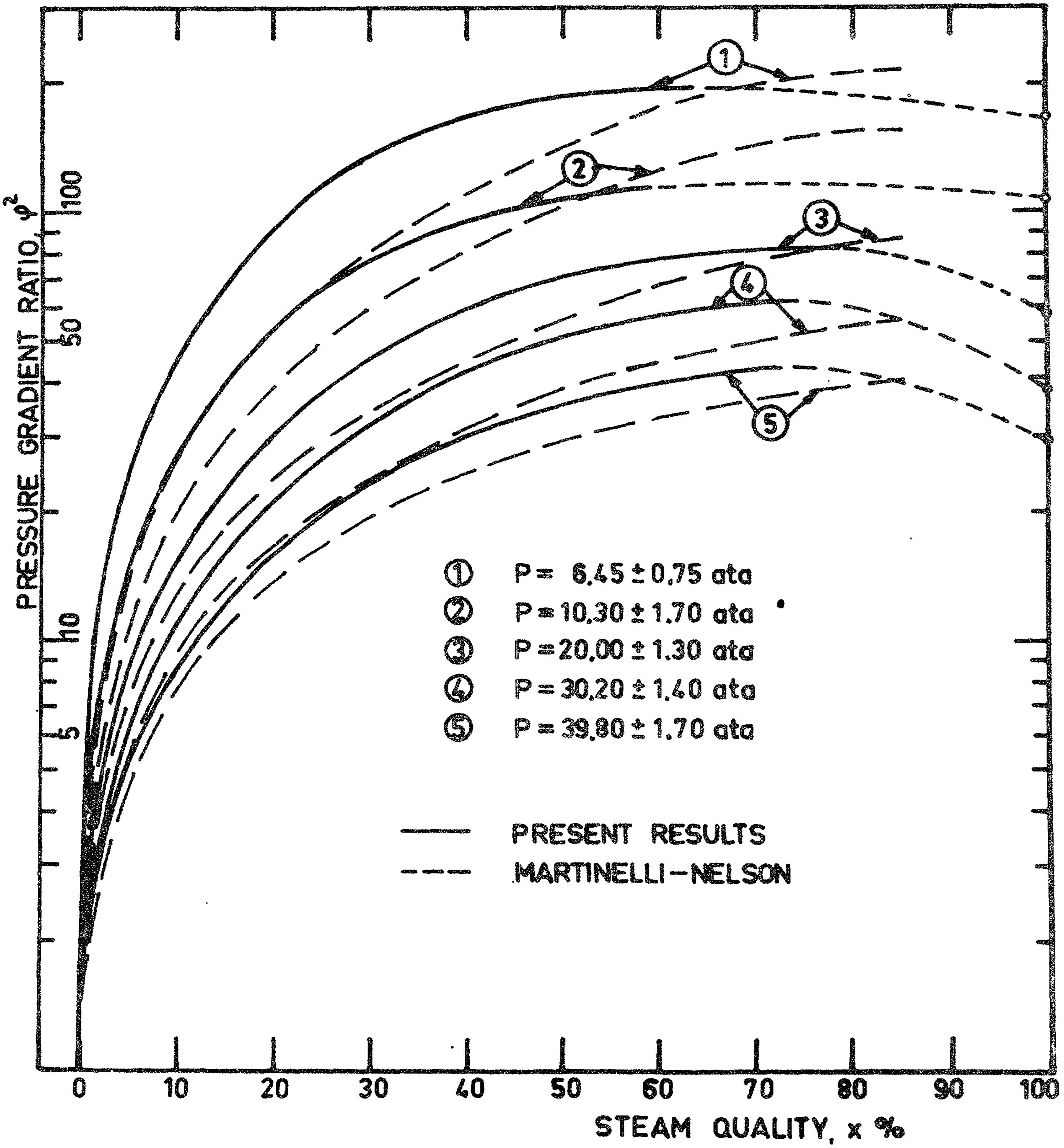

FIG. 9. COMPARISON WITH THE MARTINELLI AND NELSON METHOD

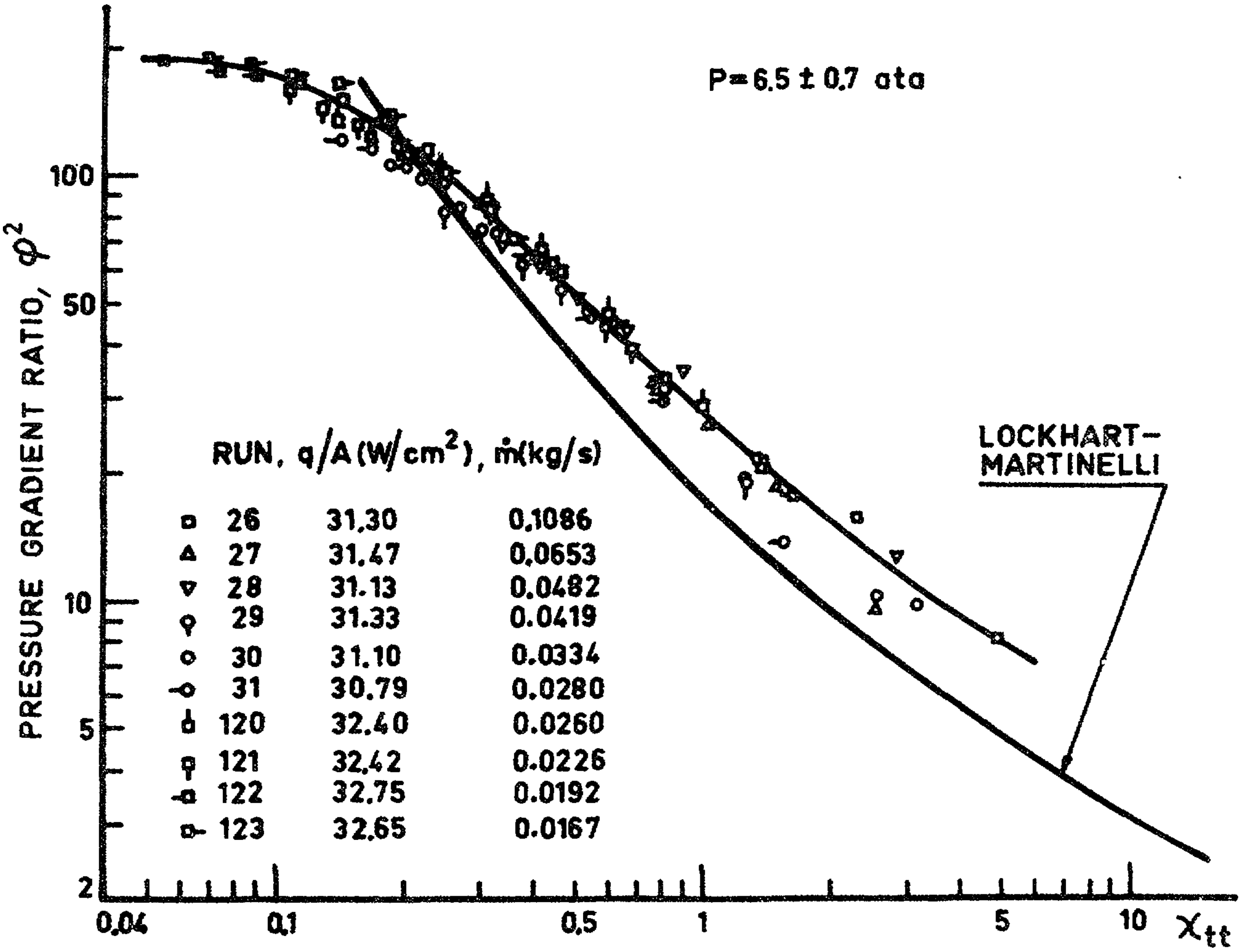

FIG. 10.

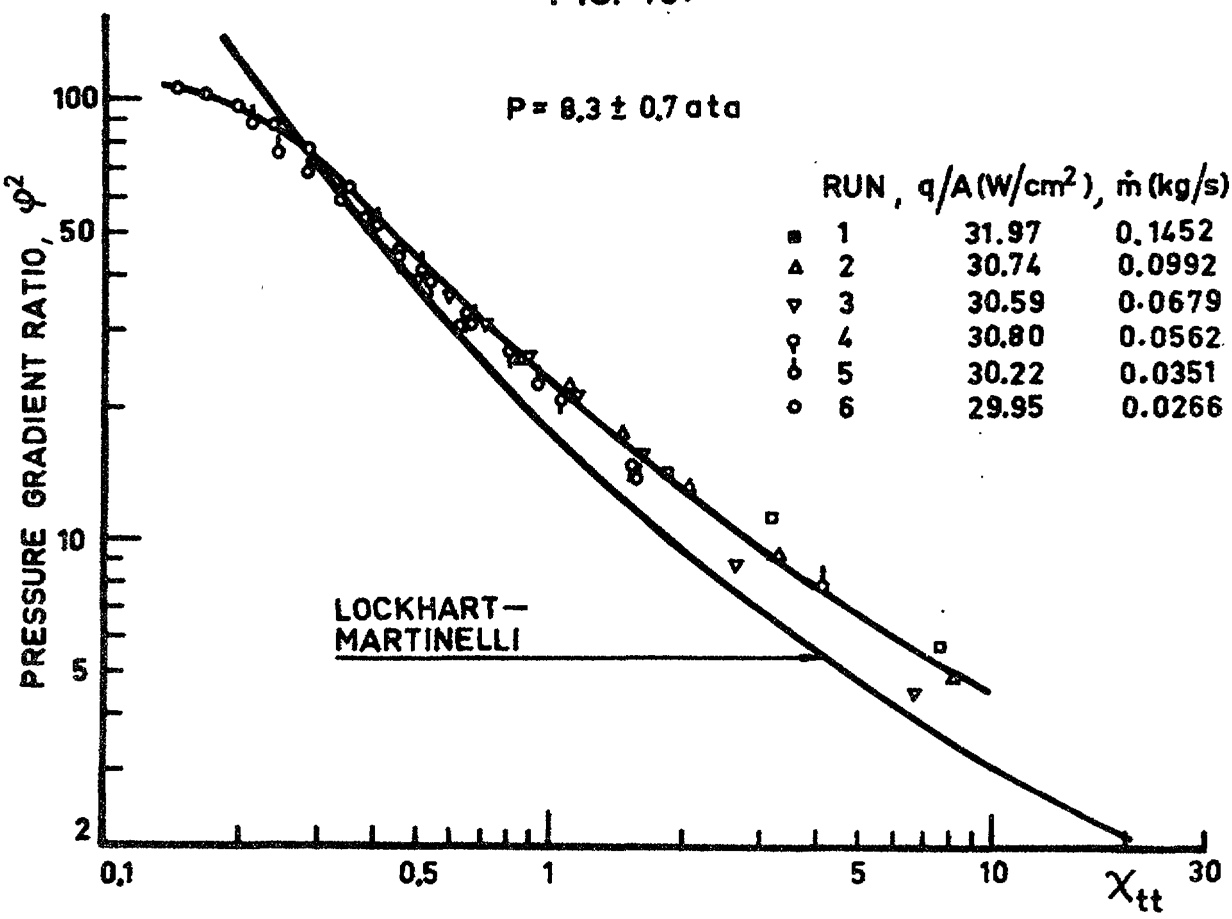

FIG. 11. MEASURED PRESSURE GRADIENT RATIOS

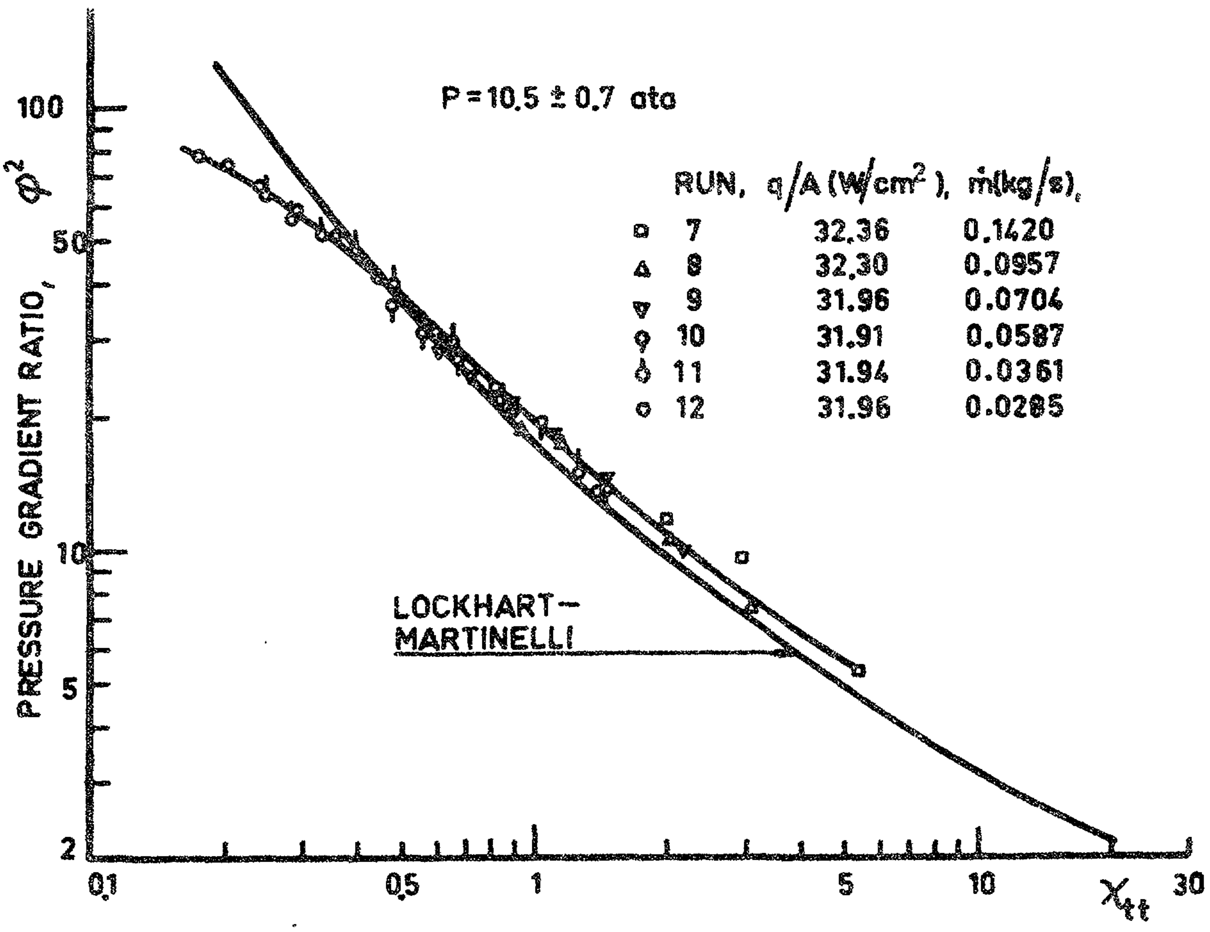

FIG. 12.

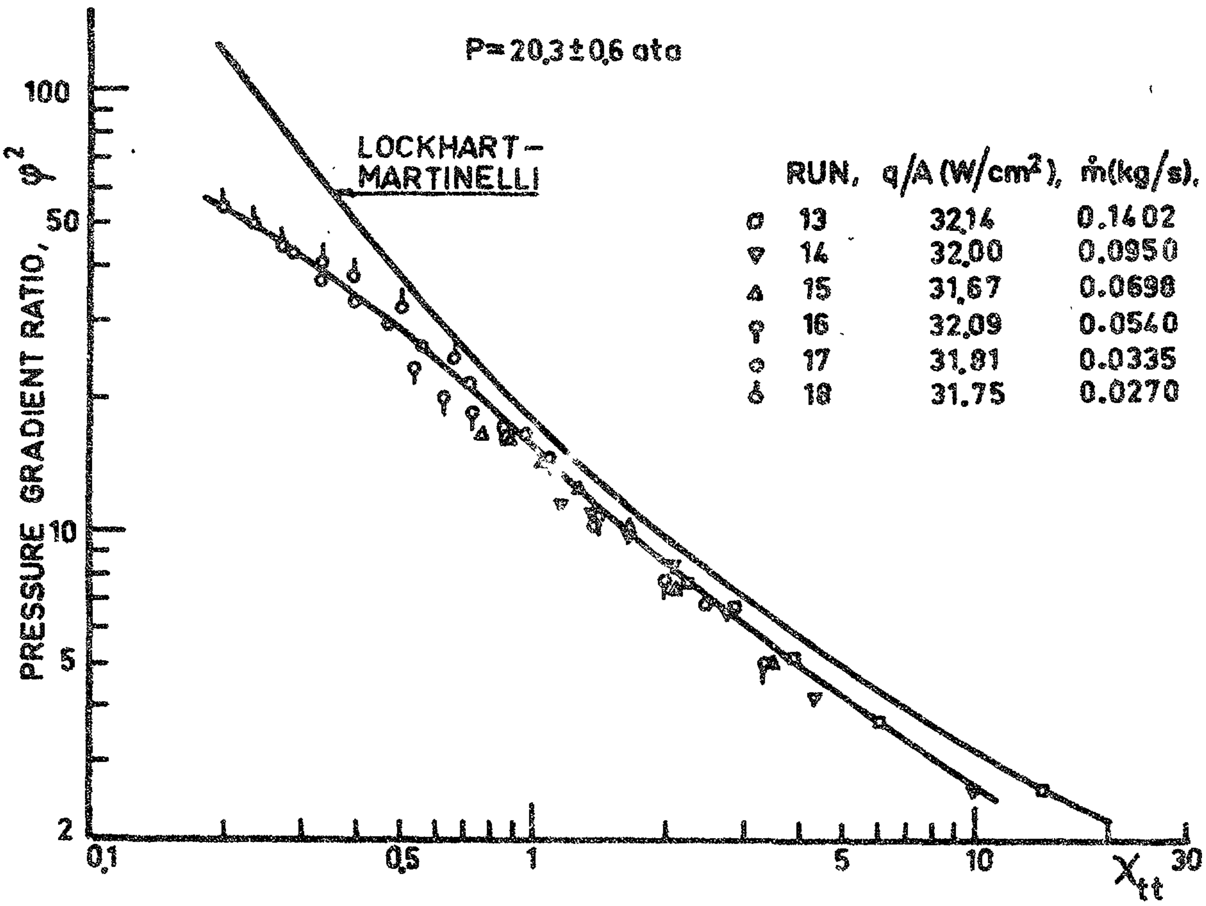

FIG. 13. MEASURED PRESSURE GRADIENT RATIOS

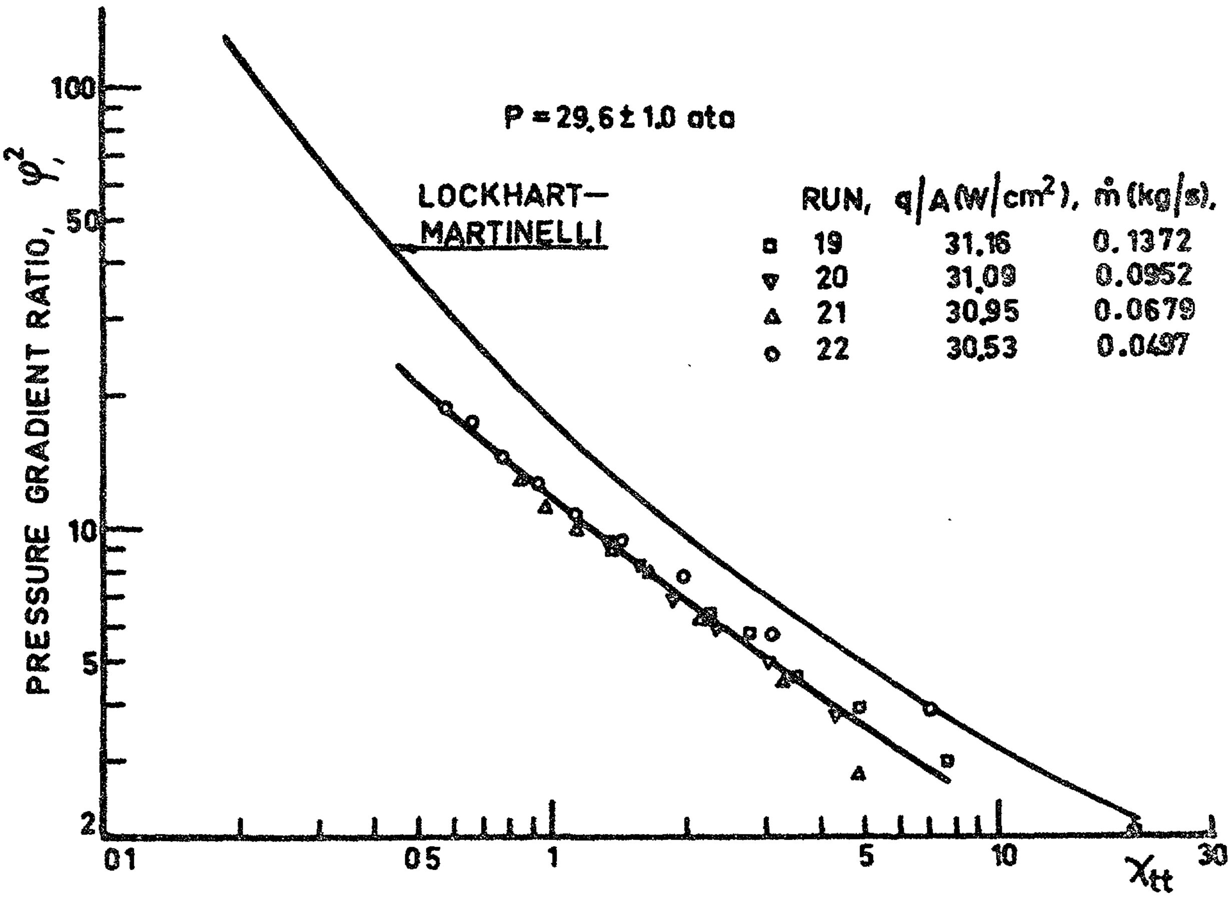

FIG. 14.

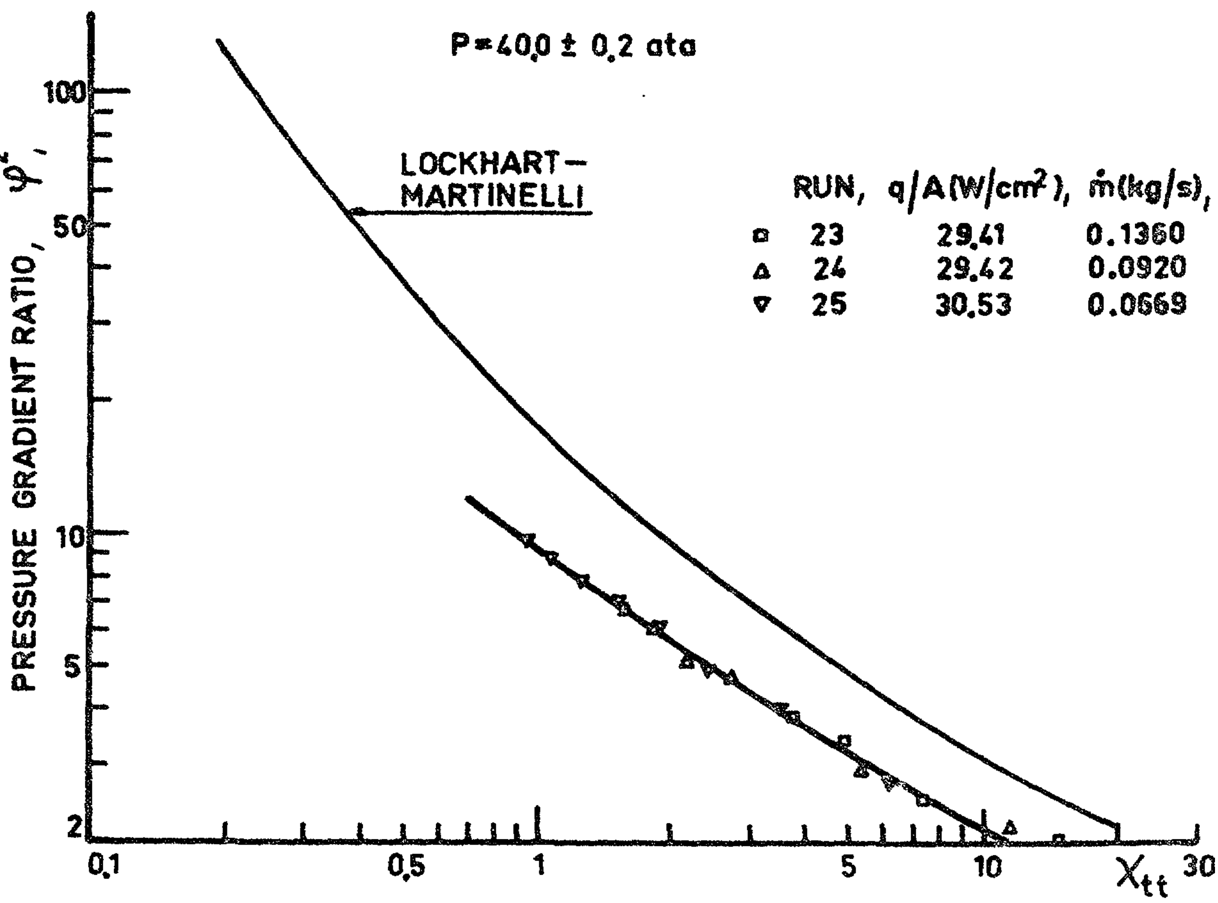

FIG. 15. MEASURED PRESSURE GRADIENT RATIOS

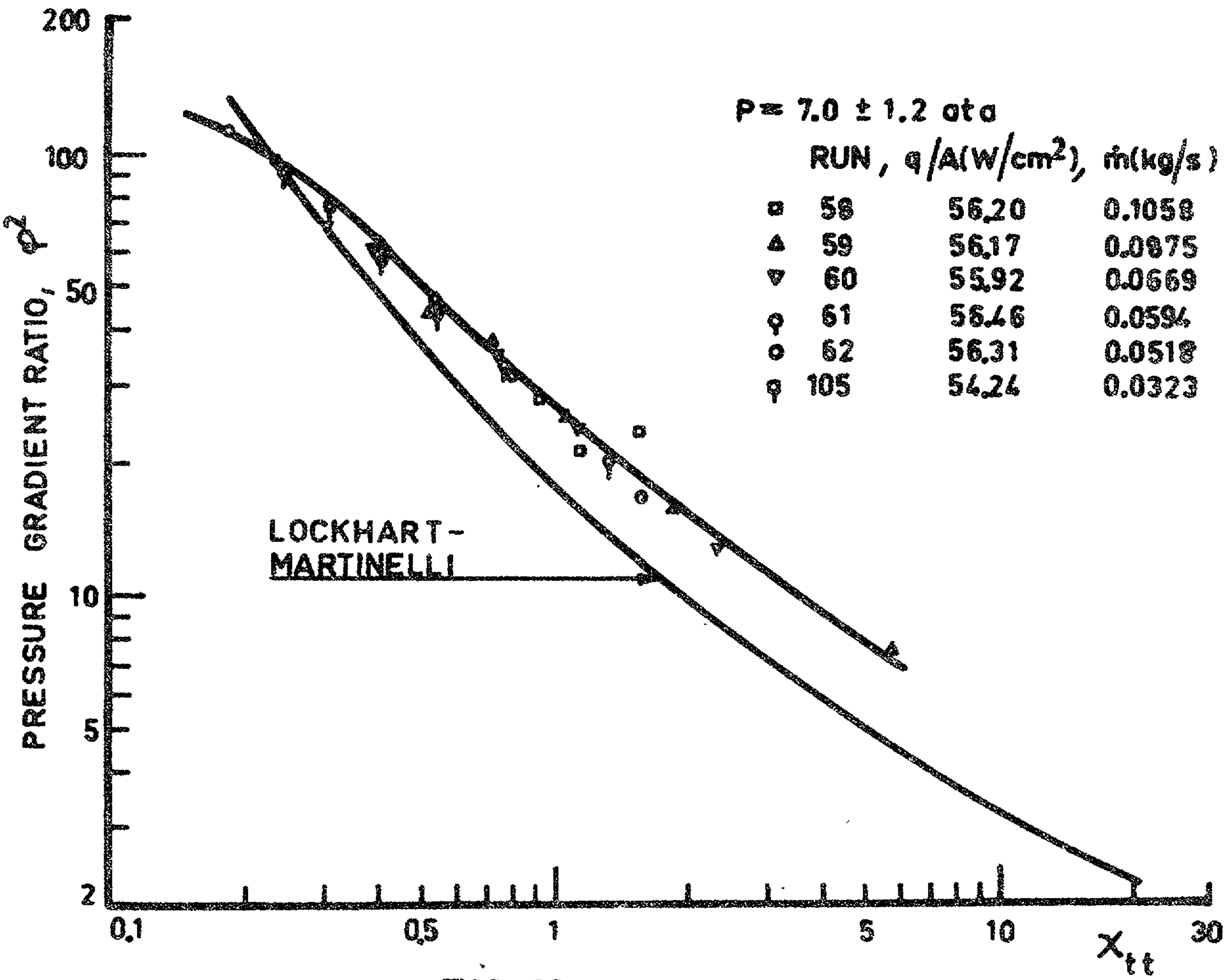

FIG. 16.

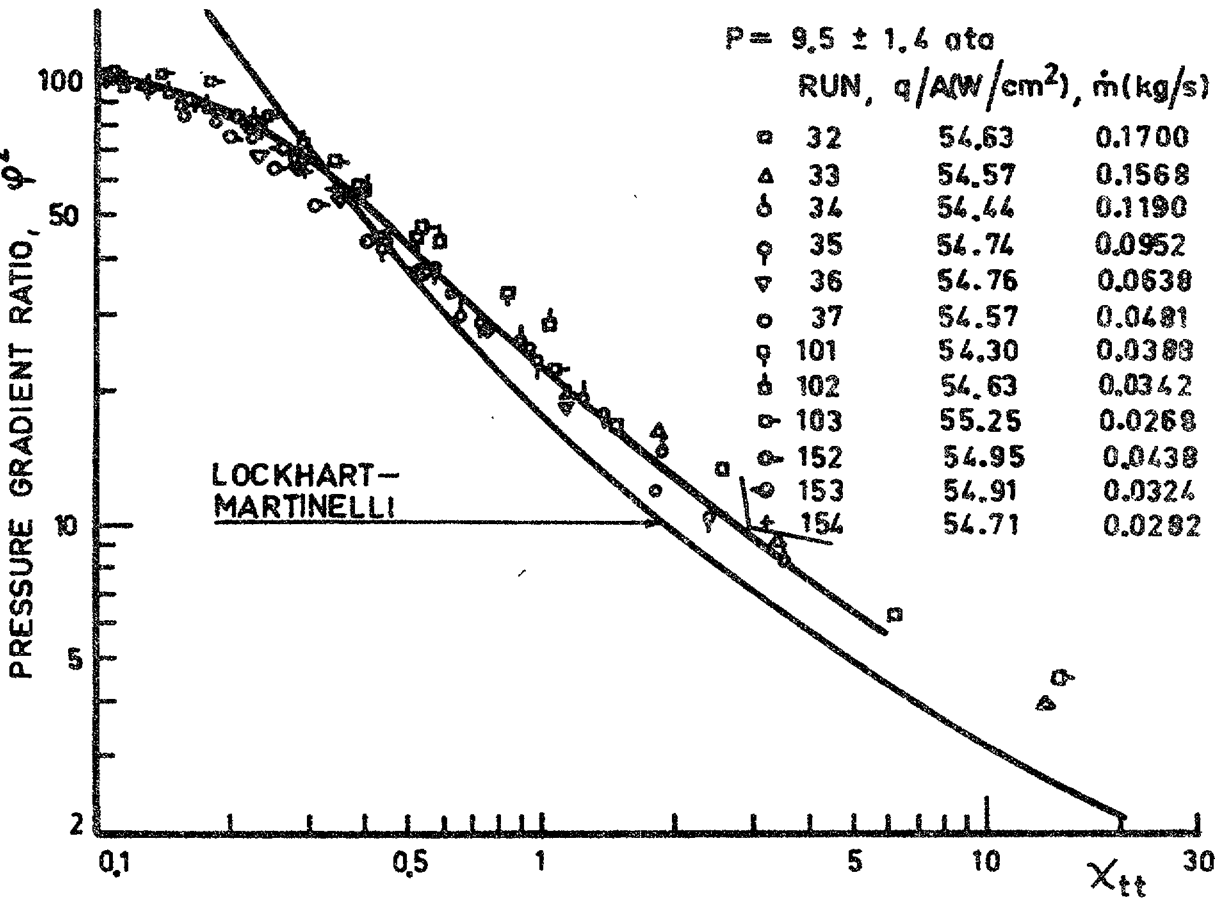

FIG. 17. MEASURED PRESSURE GRADIENT RATIOS

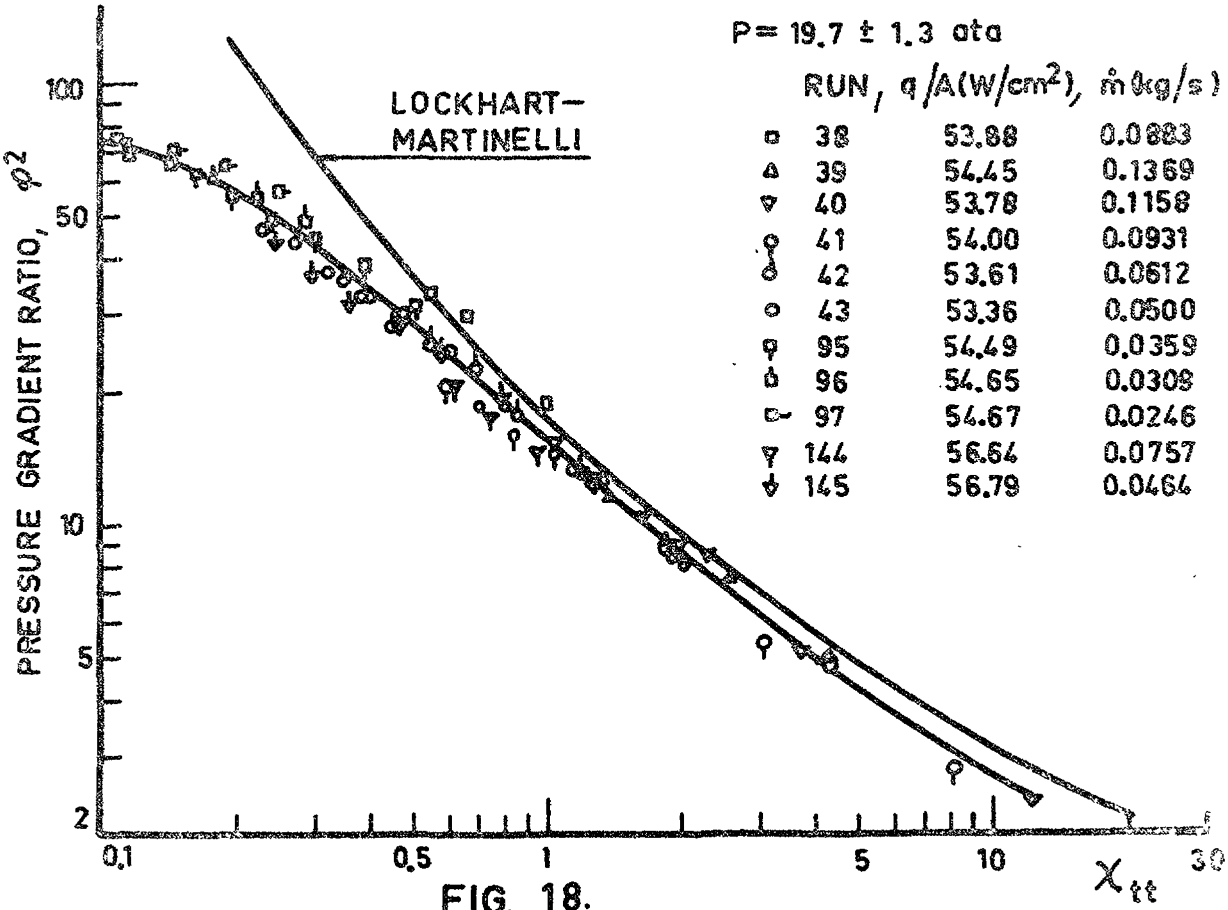

FIG. 18.

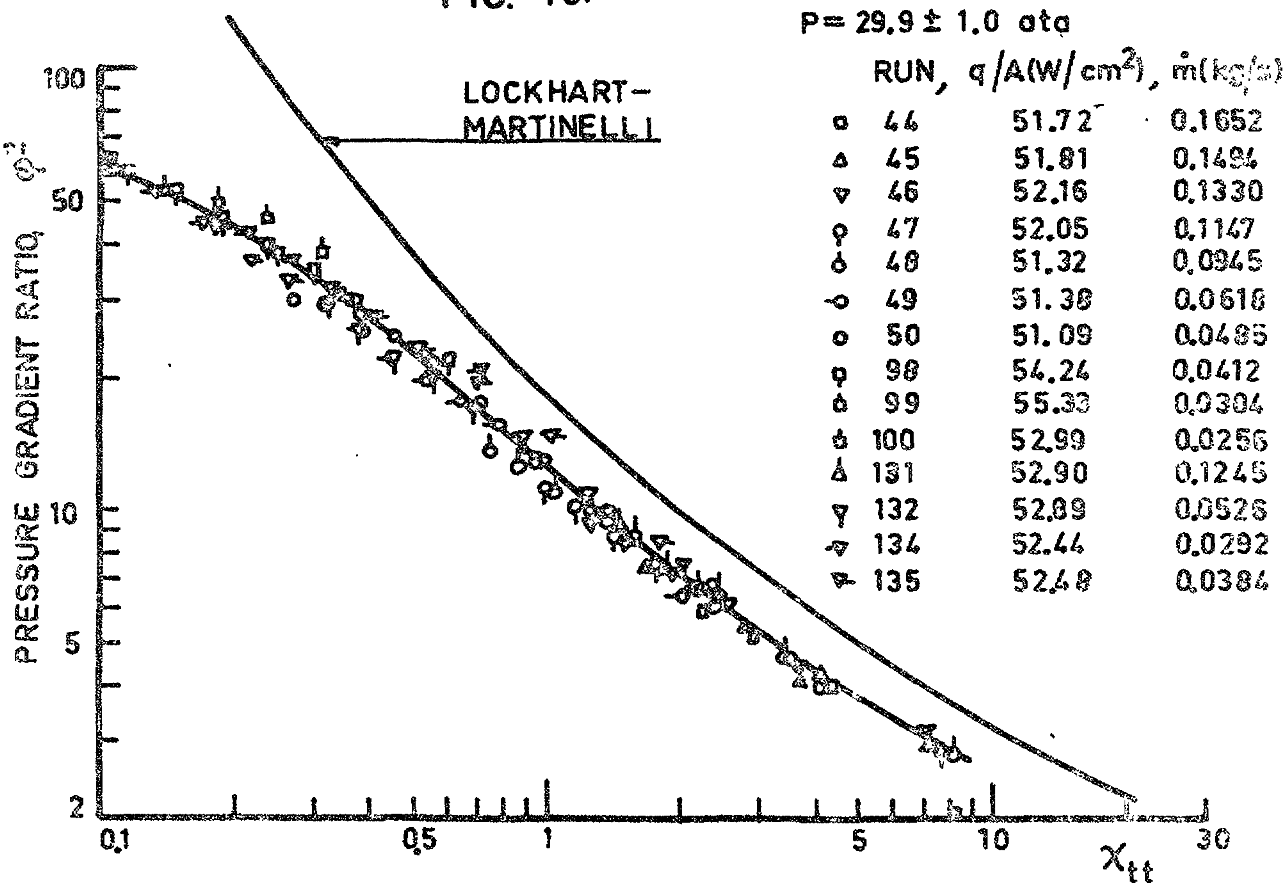

FIG. 19. MEASURED PRESSURE GRADIENT RATIOS

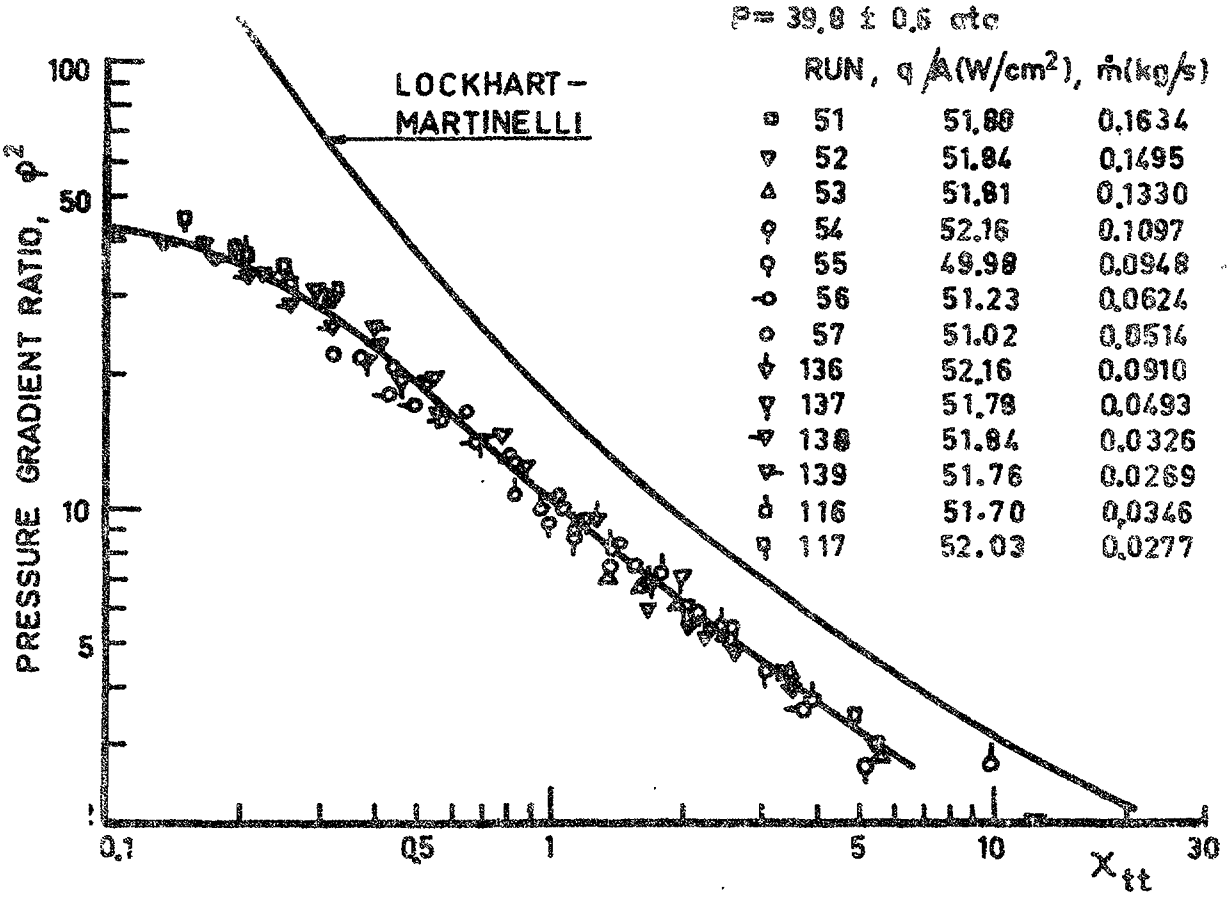

FIG. 20. MEASURED PRESSURE GRADIENT RATIOS

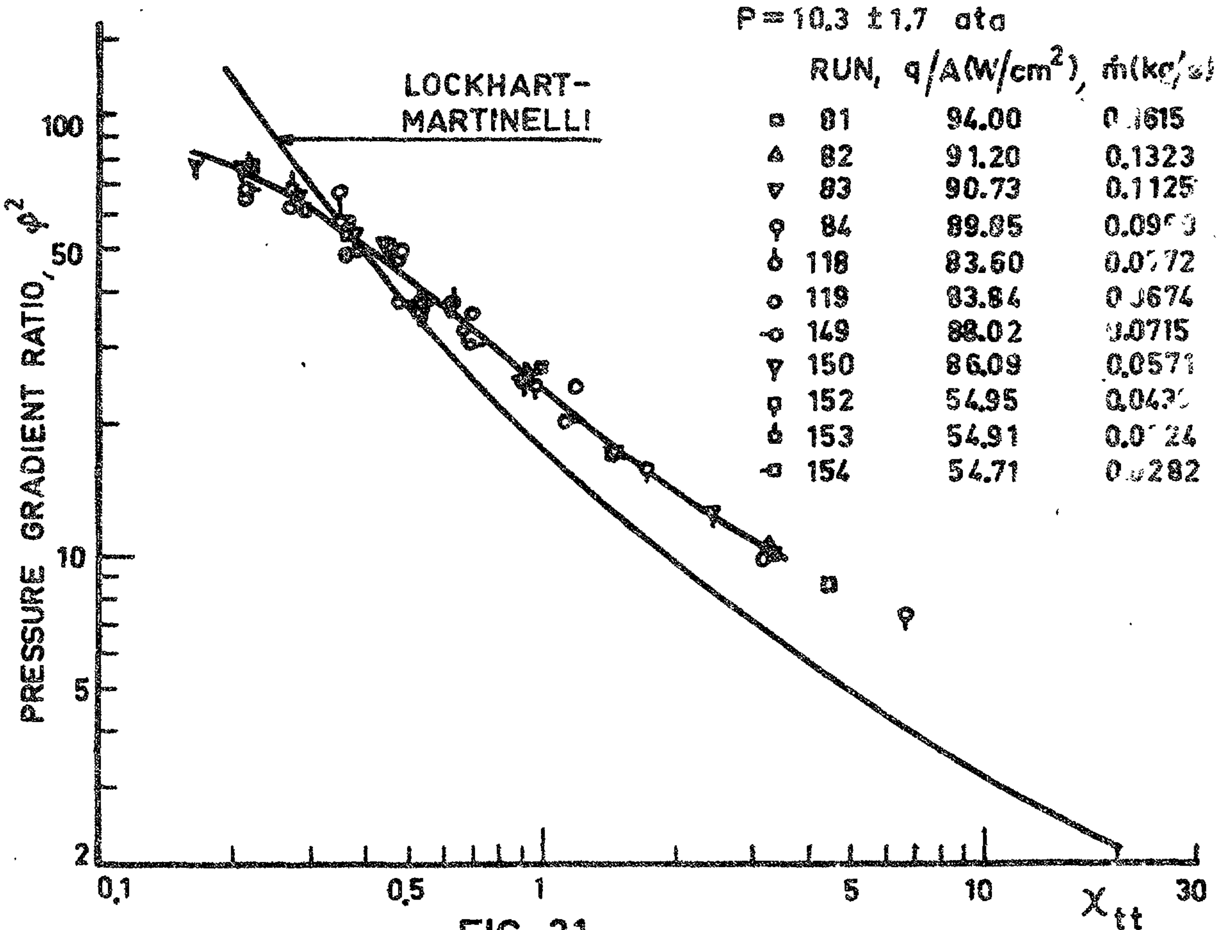

FIG. 21.

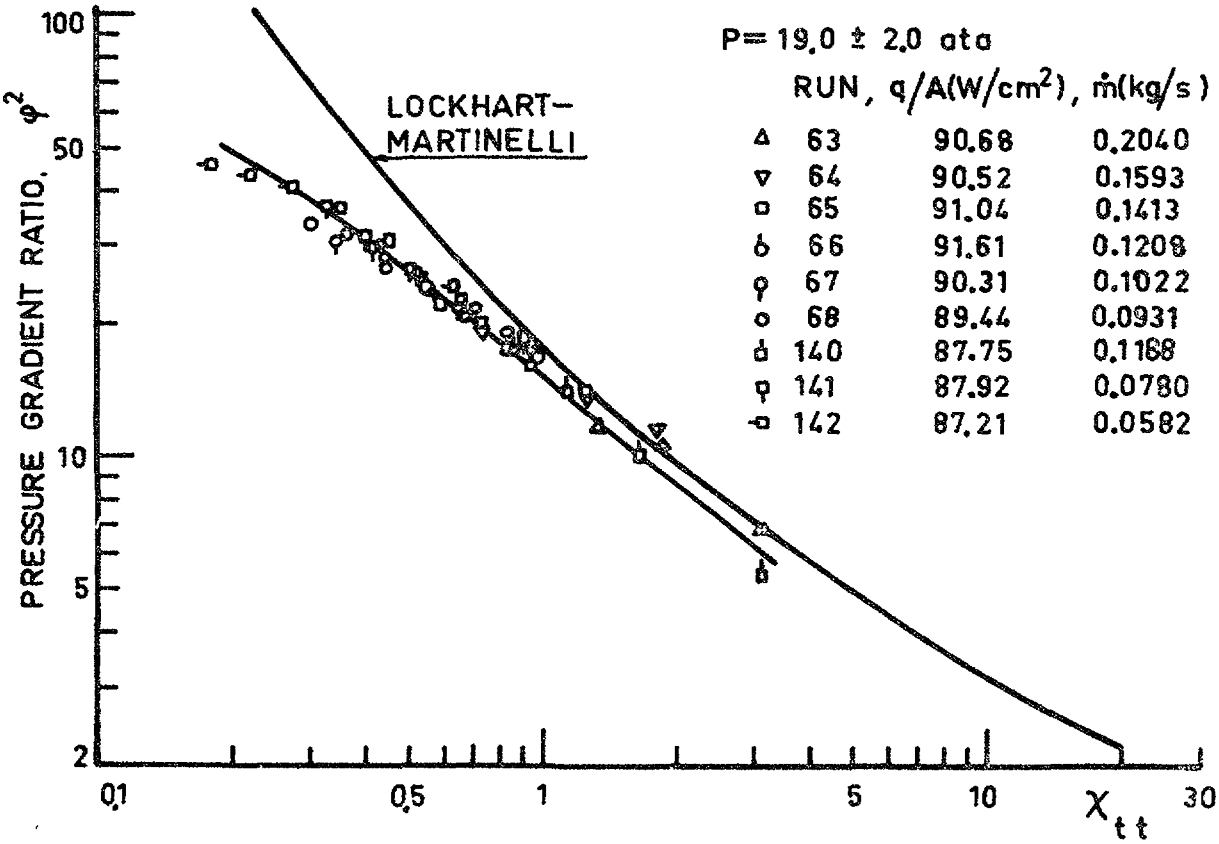

FIG. 22. MEASURED PRESSURE GRADIENT RATIOS

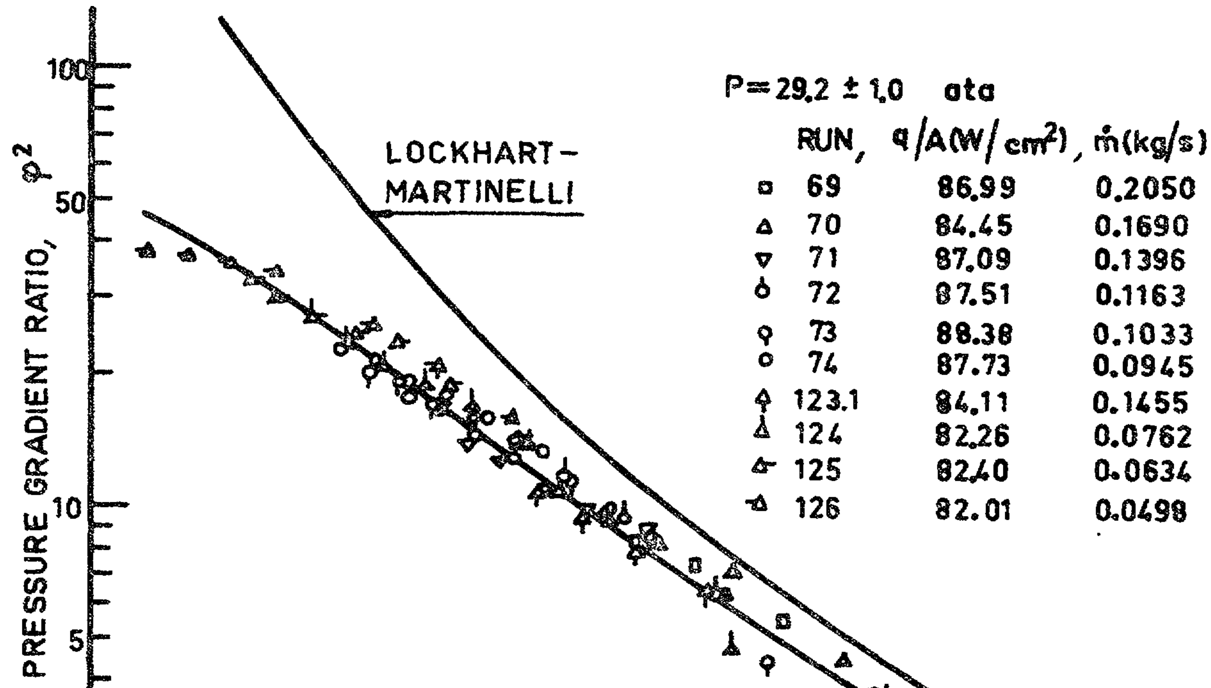

FIG. 23.

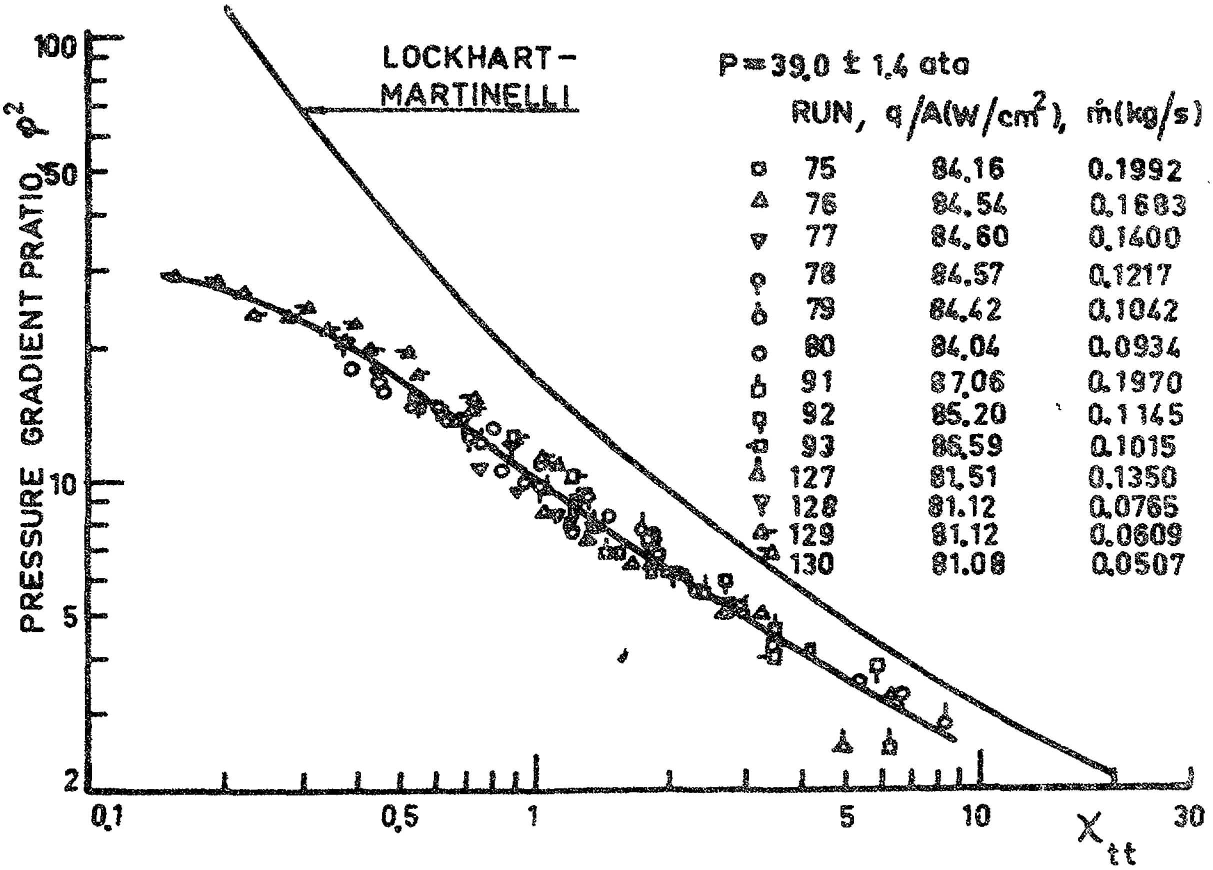

FIG. 24. MEASURED PRESSURE GRADIENT RATIOS

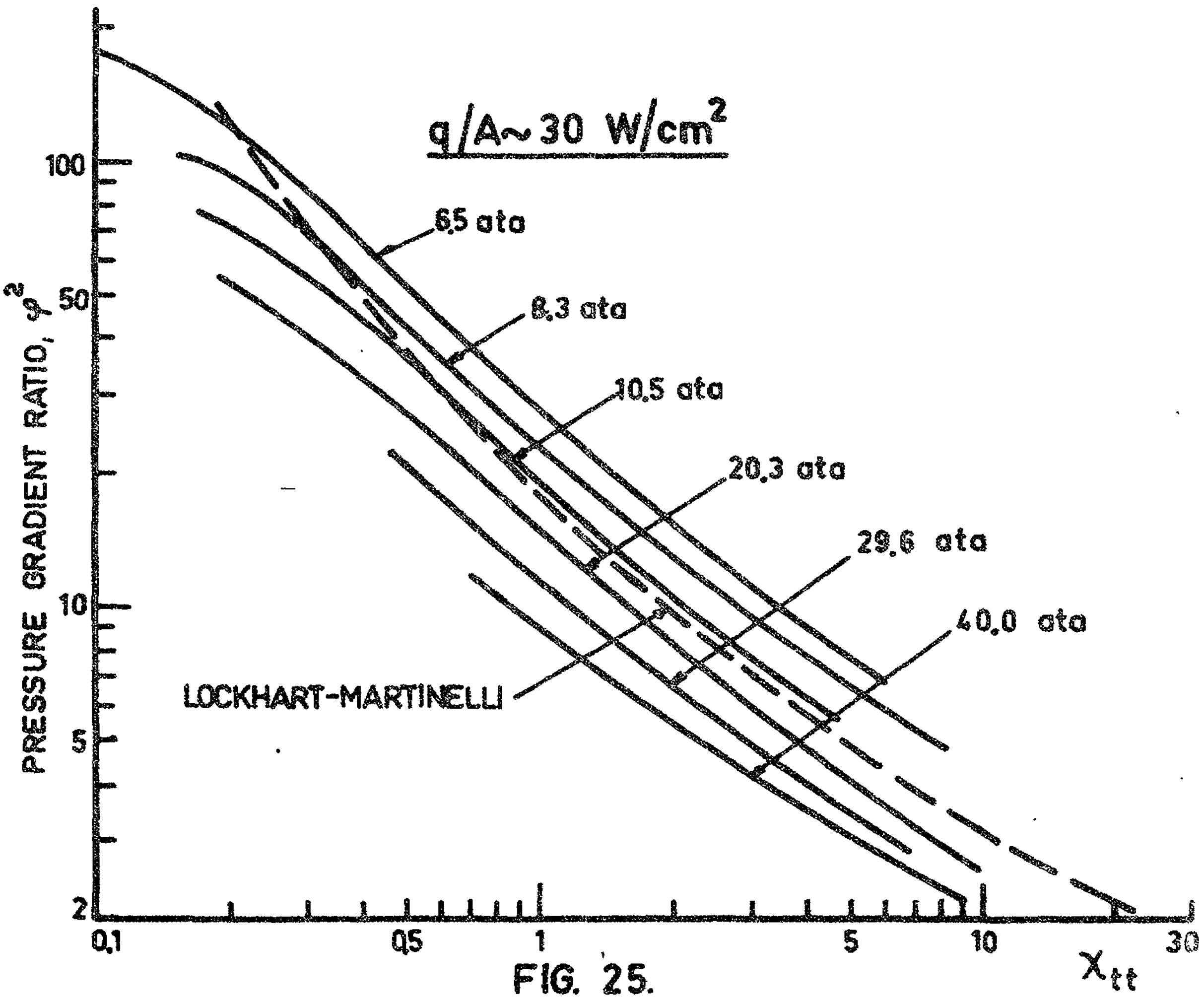

FIG. 25.

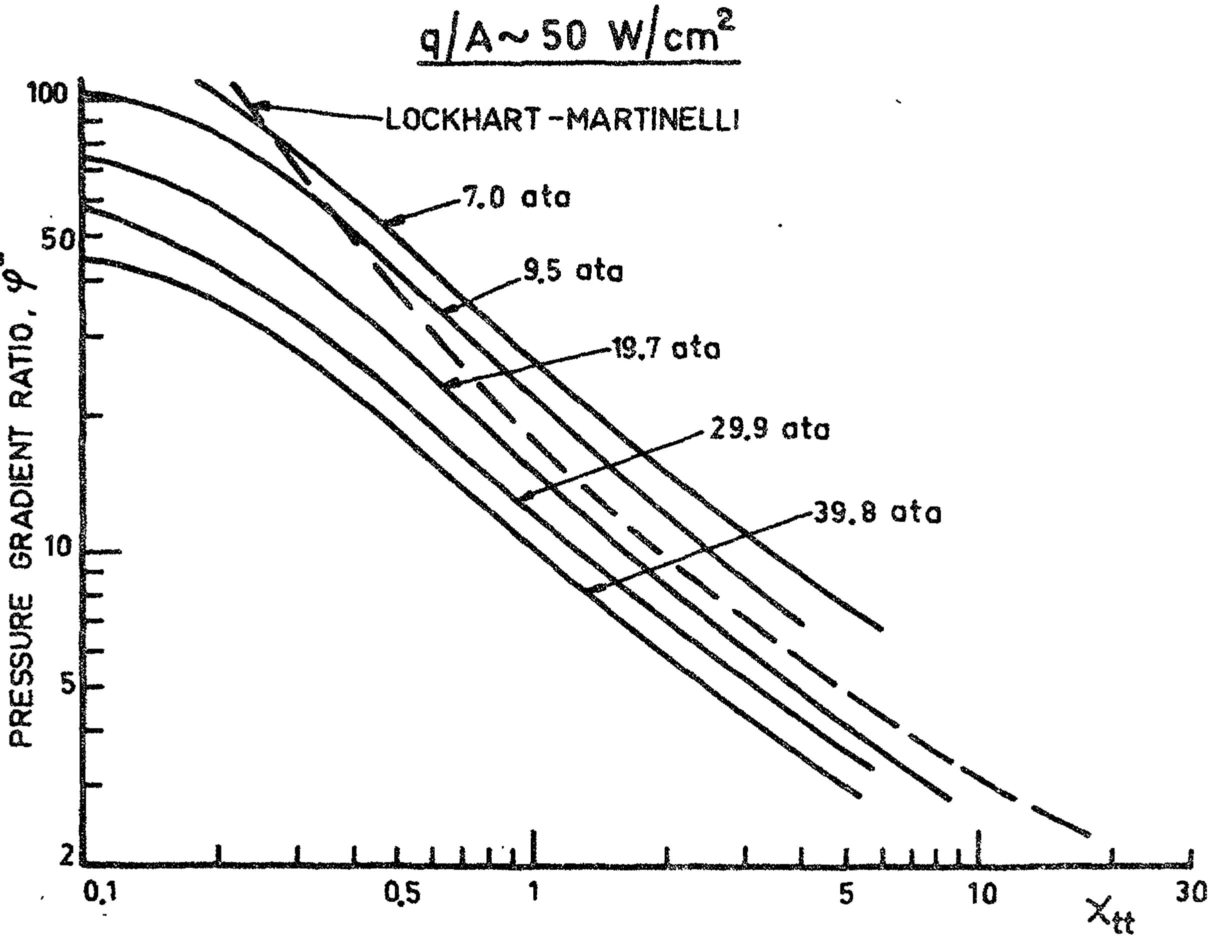

FIG. 26. PRESSURE GRADIENT RATIOS AT CONSTANT HEAT FLUX

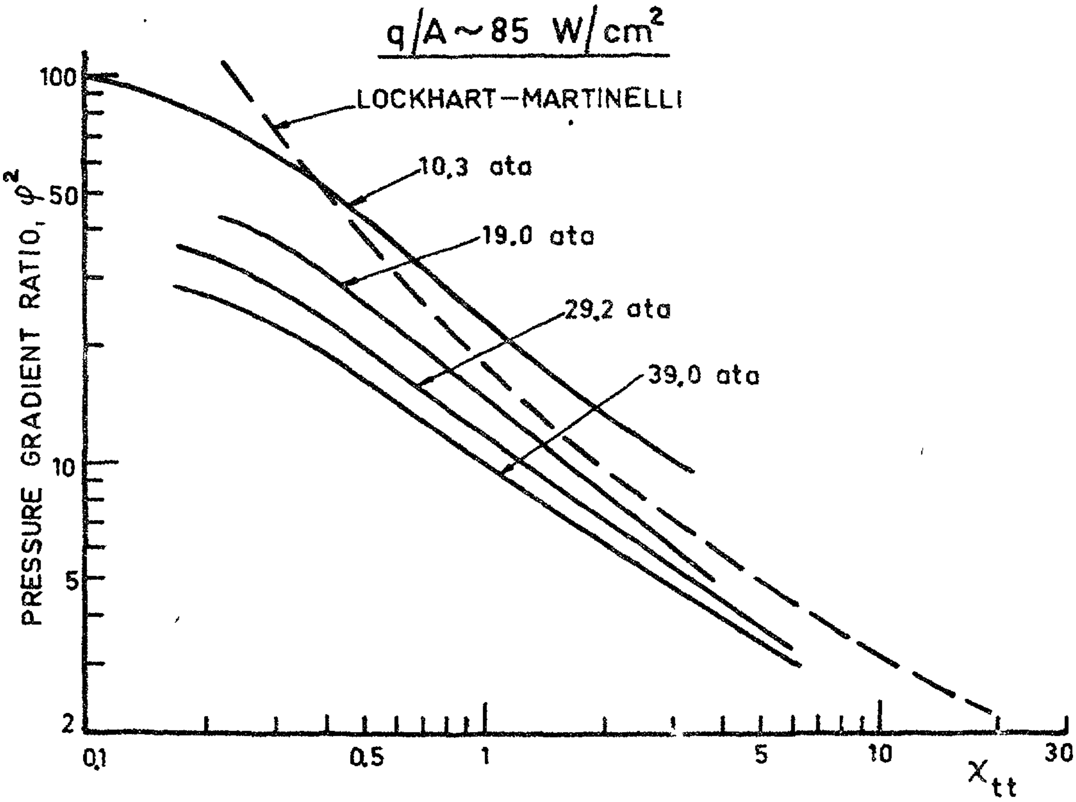

FIG. 27. PRESSURE GRADIENT RATIOS AT CONSTANT HEAT FLUX

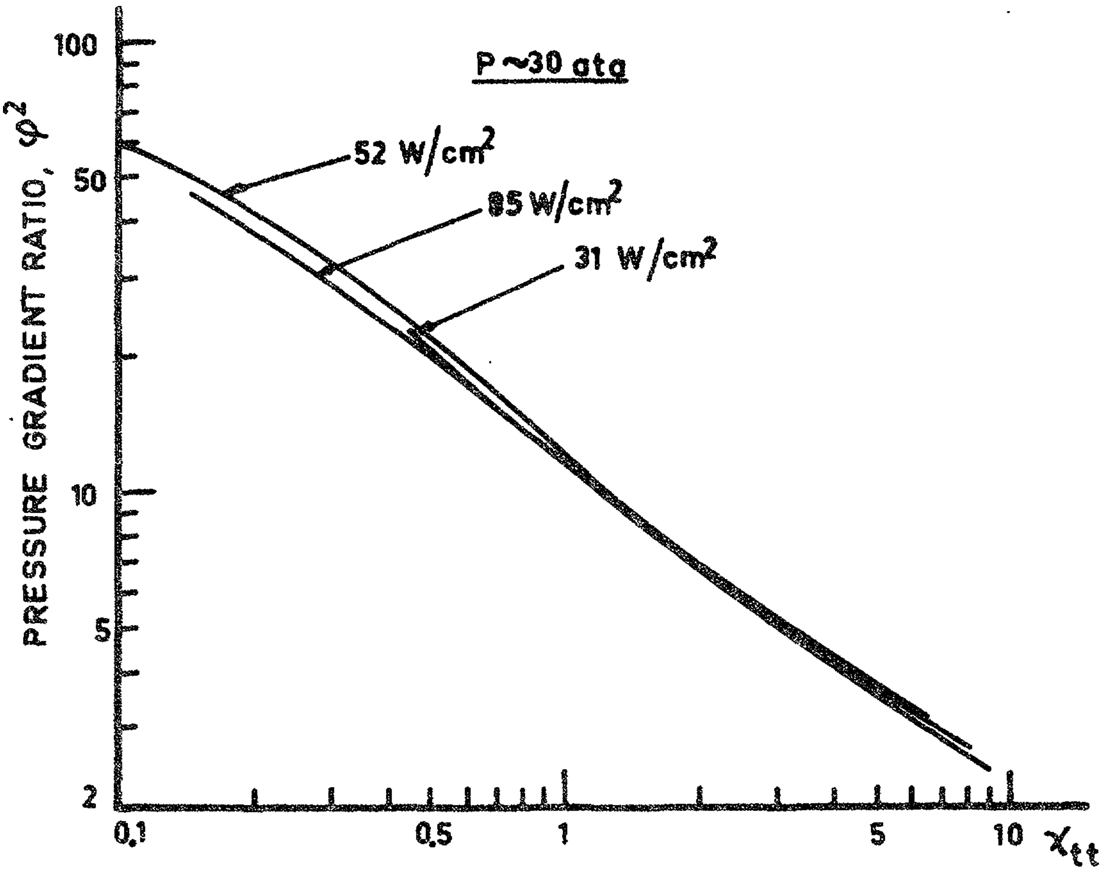

FIG. 28

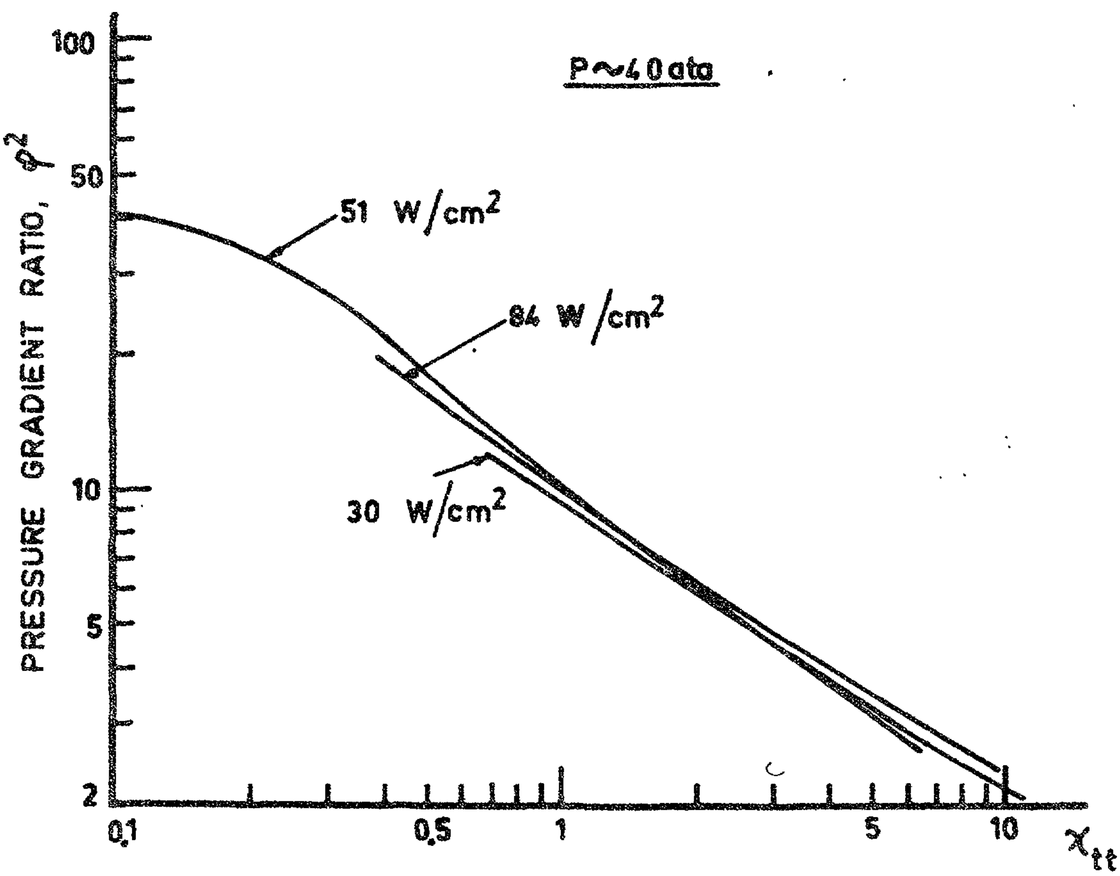

FIG. 29. EFFECT OF HEAT FLUX ON PRESSURE GRADIENT RATIOS

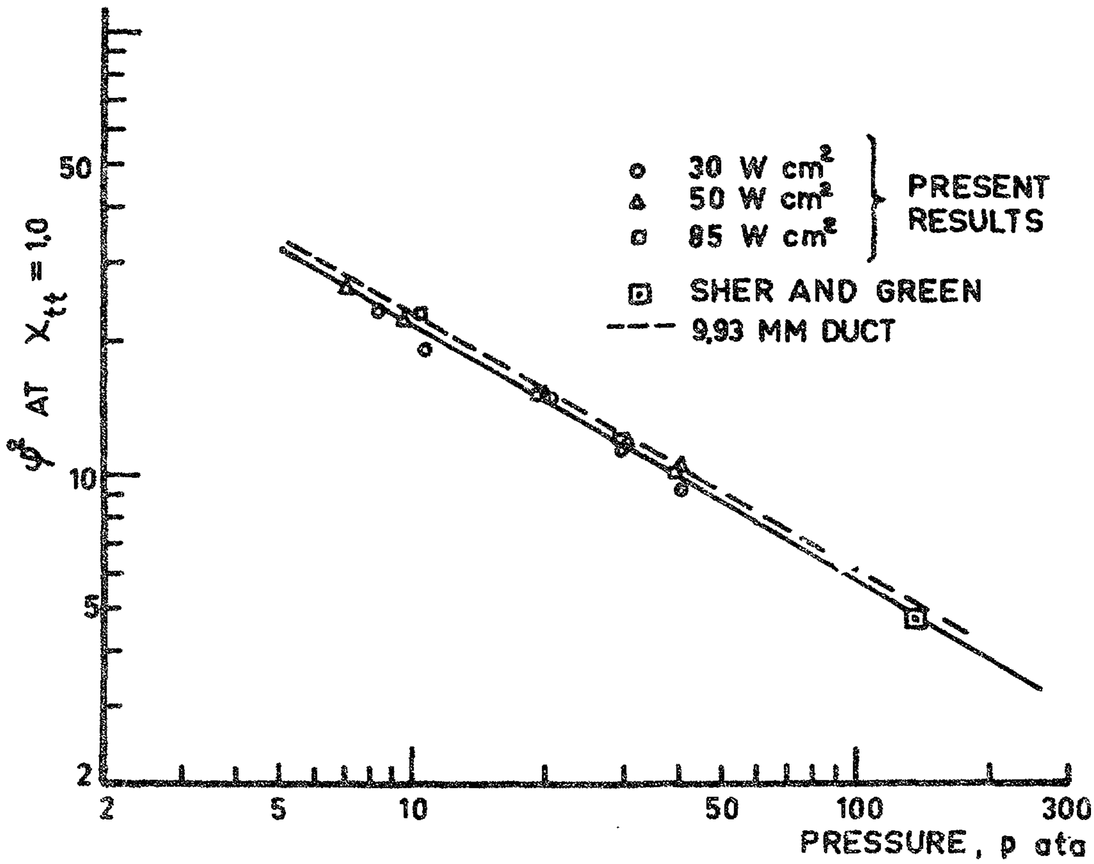

FIG. 30. EFFECT OF PRESSURE ON PRESSURE GRADIENTS

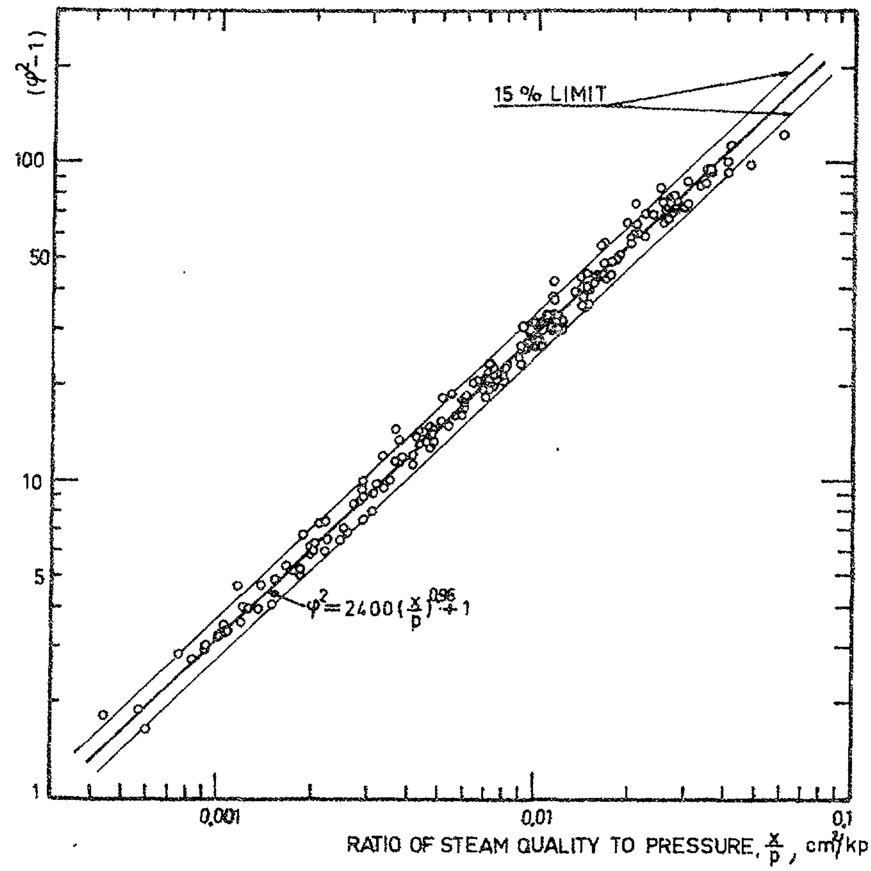

FIG. 31. PRESSURE DROP CORRELATION.

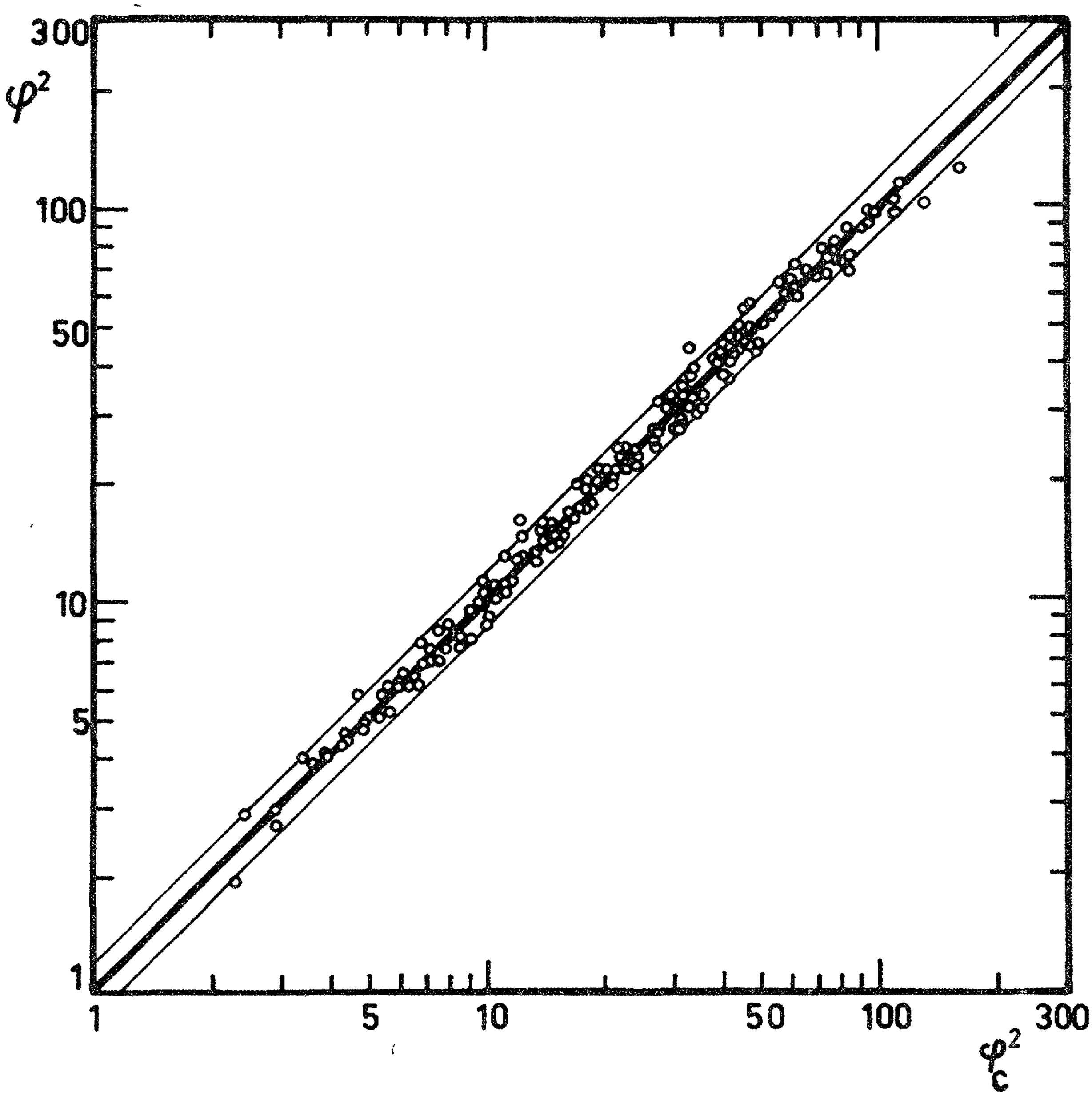

FIG. 32. MEASURED PRESSURE DROP RATIO VERSUS COMPUTED PRESSURE DROP RATIO

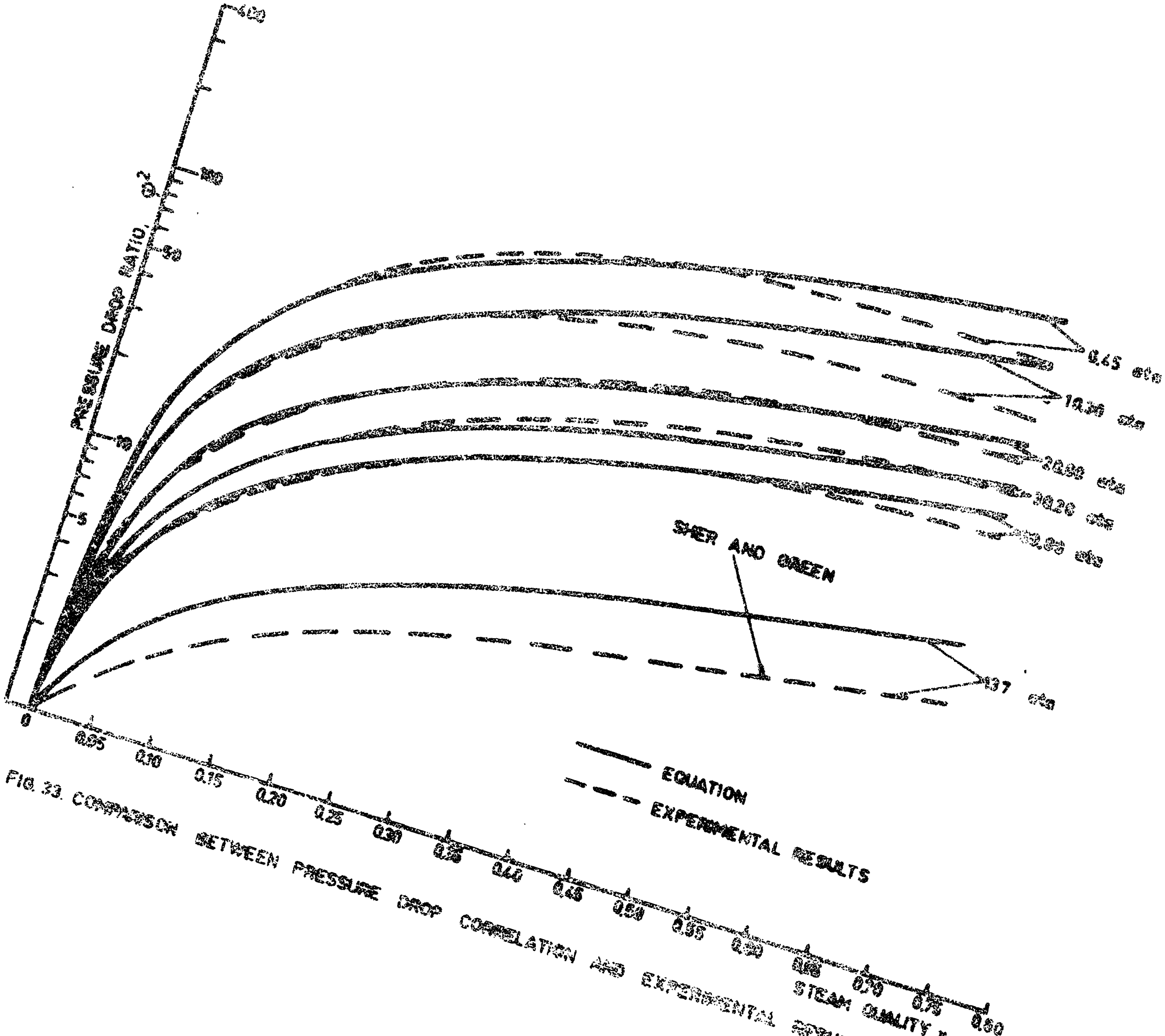

FIG. 33. COMPARISON BETWEEN PRESSURE DROP CORRELATION AND EXPERIMENTAL RESULTS.

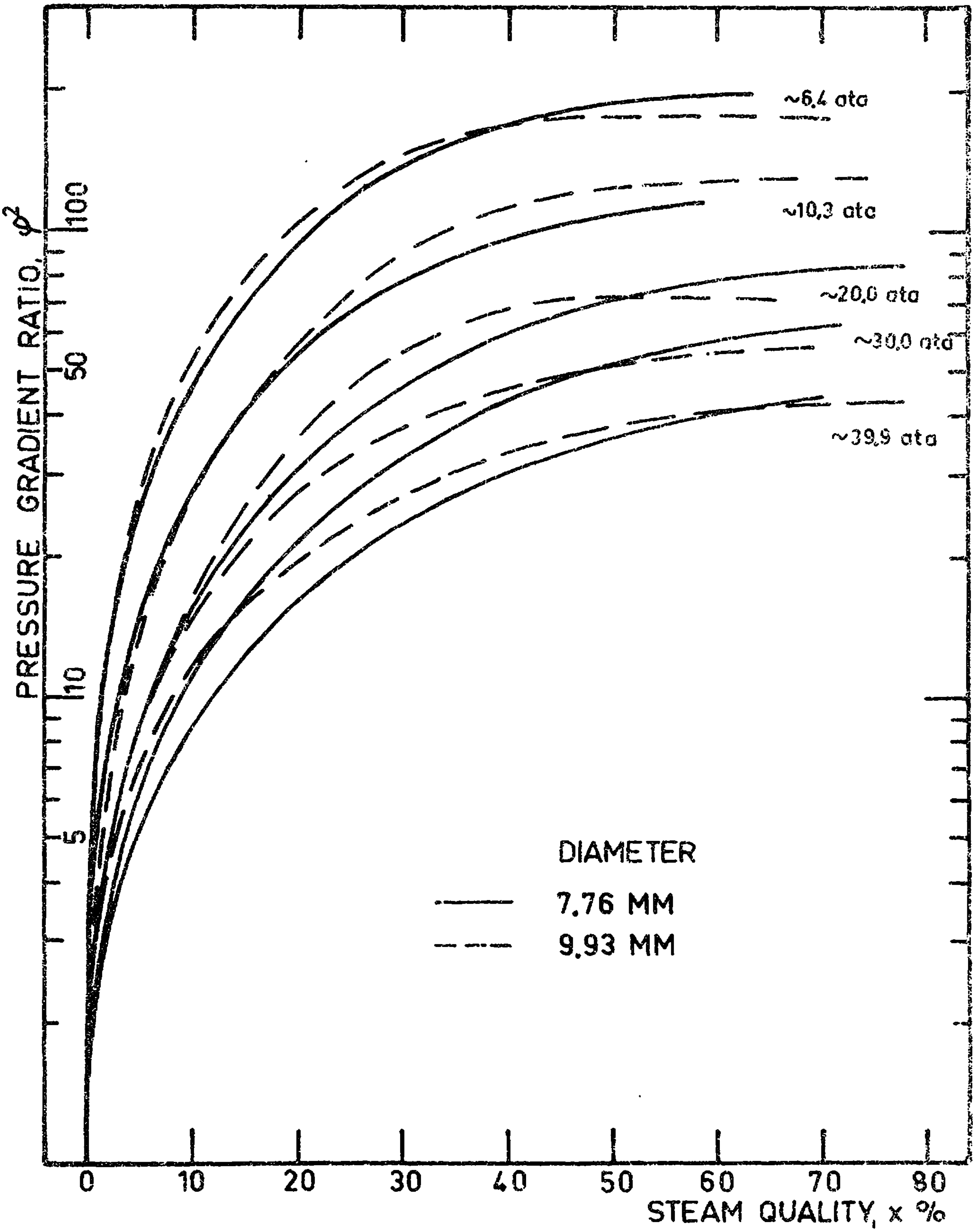

FIG.34. COMPARISON BETWEEN EXPERIMENTAL RESULTS FOR 9,93 MM AND 7,76 MM INNER DIAMETER DUCTS.

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