CEA-CONF--1885

ICNF
NSA
INIS E11

4. International seminar on heat and mass transfer in liquid metals

Trojir, Yugoslavia
September 6-11, 1971

MIXED CONVECTION IN SODIUM

by

M. WENDLING, R. RICQUE, R. MARTIN
Centre d'Etudes Nucléaires de Grenoble - France.
Service des Transferts Thermiques

MIXED CONVECTION IN SODIUM

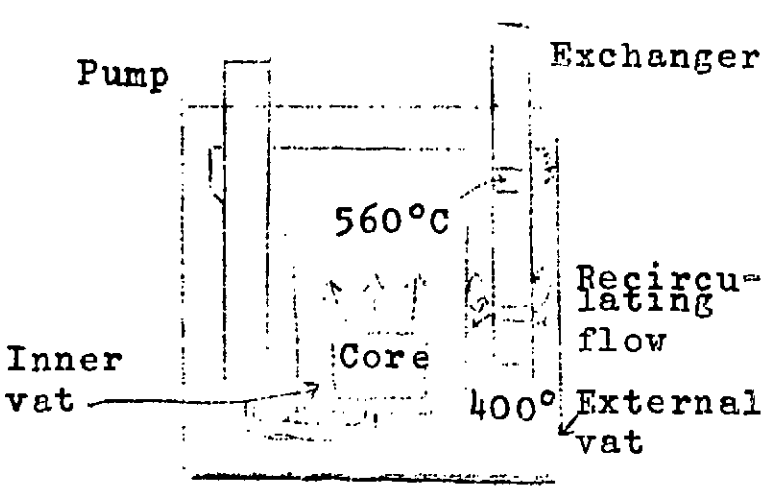
M. WENDLING, R. RICQUE, R. MARTIN.
Centre d'Etudes Nucléaires de Grenoble - France.
Service des Transferts Thermiques

ABSTRACT

Experimental heat transfer data obtained in a vertical asymmetrically heated rectangular channel at low Peclet number are presented. The effect of body forces is well correlated with G^*/Re ratio and good agreement is found with previously published Russian data. Measurement of transerve temperature profiles show that a fully developed flow may be defined on the last part of the channel.

INTRODUCTION

The construction of integrated design fast reactors obliged to develop specific heat transfer studies with liquid metal. Since very few data were available on the calculation of heat losses through the large diameter vats which, in integrated assemblies separate the hot fluid flowing from the core, from the cold fluid leaving the exchangers, it was found necessary to study heat transfer under similar conditions, i.e. in volume of large size where fluid movement resulting from natural convection is more or less perturbed by recirculating flow characterised by relatively low mean velocity.



EXPERIMENTAL UNIT

The experimental unit [1] is a loop consists pricipally of a vertical tank, an air exchanger, a pump and an electromagnetic flowmeter (figure 1). The vertical tank is parallelipipedic 3m high with a rectangular horizontal cross-section 40cm x 50cm. One of the sides of the tank heated indirectly with a constant heat flux over a height of 2m forms one of the walls of the test channel.

The test channel is defined with :

- the heated wall of the tank ; the heated surface is 2m x 0,4m
- a compartment, called "sliding wall" placed vertically in the tank and containing an argon barrier to afford adiabatic conditions. This sliding wall may be moving parallely to the heated wall so that the width of the channel can be varied from 3,4cm to 30cm approximately. It overreaches the sodium level in the tank.

The fluid flows upwards, enters the test channel by a diverging tube with one wall forming part of the tank and leaves through a rectangular (40cm x 10cm) opening in the heated wall of the tank.

Besides those concerning flowrate and heat input, measuring instruments consist mainly of chromel-alumel thermocouples in a stainless steel sheath, outer diameter 1,5mm; for recording of :

- fluid temperature at the inlet (in front of the diverging tube) and at the exit of the test channel (in the rectangular cross section tube between the tank and the exchanger).
- longitudinal temperature profiles on the heated wall and on the sliding wall (figure 2) in the vertical symmetrical plane at intervals of 8cm. On the heated wall, they

are set in holes chilled so that their hot solder arrives at about 1mm from the wetted surface.

- transverse temperature profiles with a "rotating probe". Measurements are taken at 7 levels at intervals of about 30cm. At each level the profil is noted by a thermocouple sticked at the end of a horizontal arm which is an extension of a vertical tube forming the rotation axis of the device.

All the measurements are recorded by a Solartron electronic voltmeter with automatic conversion.

EXPERIMENTAL FIELD

The experimental field was as follow :

- Heat flux at the channel wall has varied : from $2{,}6 W/cm^2$ to $8{,}1 W/cm^2$.
- Flowrate from $100 cm^3/s$ to $2750 cm^3/s$.
- Tests have been realised with the mean temperature of the fluid in the channel as reference and with hydraulic diameter as length of reference have varied in the following ranges :

  Reynolds number, Re, from 780 to 25000.

  Prandtl number , Pr, from $5{,}5.10^{-3}$ to $9{,}3.10^{-3}$.

  modified Grashof number (classic Grashof number x Nusselt number), G^*, from 3.10^7 to 3.10^{12}.

  Peclet number, Pe, from 5 to 185.

LONGITUDINAL TEMPERATURE PROFILES

On figures 3 and 4 are presented 3 characteristics tests. The longitudinal temperature profiles are given as a function of vertical abscissa z. z = 0 and z = 200 correspond respectively to the beginning and to the end of the heated surface.

It may be seen for the TOP 7, figure 3 that :

- the top curve shows the gross profile of the heated wall measured with the thermocouples. There are many marks corresponding to several recordings necessary to know whether the test is steady or not.
- the curve below shows the temperature profile of the wet surface of the heated wall.
- when they have been measured the transverse temperature profiles let to determine with extrapolation the temperature of the heated wall (on the figure, the mark is a +).
- the straight line represents the variation of the mean temperature of the fluid.
- the lower curve is the longitudinal temperature profile of the sliding wall.

<u>For the sliding wall</u> the evolution of the longitudinal temperature profile is characteristic. For the tests carried out at very low flowrates, the evolution is quasilinear, the influence of axial conduction is high and the profile is appreciably perturbed near the channel outlet. For all the other tests, three distinct zones are often observed :

- a first zone near the inlet of the channel where the temperature is constant and equal to the inlet fluid temperature. On the figure 5, we show the variations of the length of this zone as a function of Peclet number : for a given width, it increases as Peclet number increases and as heat flux decreases. For a given heat flux it is more important when the channel is wider.

- a second zone where the temperature increases rather quickly ; this second zone may not exist (TOP 120, figure 3).
- a third zone where the temperature increases linearly. The length of the two first zones or the abscissa of the beginning of the third is difficult to find on the figures, it seems to have analogous variation with the length of the first zone. This evolution of the sliding wall temperature may be compared with experimental results obtained by HANRATTY and SCHEELE |2| in aiding and opposing flow for low Reynolds number : in the axis of the channel a "paraboloïd" zone is formed where the fluid is pratically at rest.

<u>For the heated wall</u> over the first part of the channel in the "flow development" zone, the temperature profile of the heated wall presents a characteristic evolution. Study of the variation in the deviation between the temperature of the heated wall and the mean temperature of the fluid, represented by a straight line, shows that the deviation attains a maximum. The local heat transfer coefficient starts by decreasing when the location distance increases and attains either a limit value corresponding to the fully developped flow at the end of the channel , or a minimal value before approaching the value corresponding to the fully developed flow. This phenomenon, sometimes referred to as "overshoot",can be compared qualitatively to that observed in analogous conditions. PIROVANO, VIANNAY and JANNOT [3] think that this minimum heat transfer value is significant of the end of the laminary flow. This cannot explain our results because Reynolds number is always higher than the critical value (at the inlet of the channel). ALFEROV, RYBIN and BALUNOV [4] consider that this "overshoot" is the result of a relaminarisation of the flow by the natural

convection : the laminar sublayer becomes thicker till a critical value where the local heat transfer coefficient is the lowest. It may be the result of a radical reorganisation of the flow, with the local heat transfer coefficient perhaps attaining its minimal value when the mean velocity of the flow in the free stream is near the maximum local velocity produced by natural convection as suggested by HALL and PRICE [5] .

TRANSVERSE TEMPERATURE PROFILES

Transverse temperature profiles have been measured for several tests with the rotative probe above mentioned. By extrapolation (tangent with a slope $\partial T/\partial y = \varphi/\lambda$) the temperature of the heated wall on the fluid side can be determined. Comparing with the longitudinal profiles recorded by the thermocouples set in the heated wall, it can be said, in the limit of the accuracy of measurements, that there is a good agreement.

Figures 6 and 7 show some examples of the recorded transverse profiles. The temperature difference $T_{wall} - T(y)$ is given as a function of the reduced distance y/b, for the seven levels of measurement. Temperature was fluctuating and it is the mean value which is given.

Two types of profiles can be defined :

- the first type (figure 6) obtained during tests carried out with small channel width and relatively high flowrates, corresponds, under our experimental conditions, to value of the dimensionless group G^*/Re inferior to 10^5. Under these conditions, it is noted, that apart from the first profiles measured near the channel inlet (at 4 and 36cm from the downstream limit of the heating surface), the transverse temperature profiles show little evolution. A thermal layer can be defined near the wall, showing pretically constant thickness over nearly all the channel. It seems that the thickness of this thermal layer is smaller as G^*/Re is higher. This result is one of the confirmations of the hypothesis according to which the full thermal and hydraulic flows are developed over this downstream zone in the channel.

- the second type (figure 7) is obtained for tests where the value of the non dimensional group G^*/Re is higher than 10^5 (wider channels). The transverse temperature profiles present some important differences as compared with the preceding ones: the length of the fully developed flow zone is more limited since the profile evolue near the outlet. As in the preceding case, a thermal layer of pratically constant thickness can be defined in the fully developed flow zone but, near the sliding wall, the zone with a zero horizontal temperature gradient is wider than for the first type. These profiles differ also from the preceding ones by the fact that a minimum of temperature exists between these two zones, which can be explaineby the presence of reverse flow [2] .

HEAT TRANSFER COEFFICIENT IN FULLY DEVELOPED FLOW

The heat transfer coefficient is calculated from the longitudinal profile of the heated wall and from the straight line showing the evolution of the mean temperature of the fluid in the channel. Considering the precision of measurements, the temperature evolution of the heated wall can be represented by a straight li e, in most cases, parallel to the line indicating mean fluid temperature. That is to say, over a large downstream part of the channel full flow development seems to be attained. With this hypothesis, the precision of calculation of mean heat transfer coefficient is included between 12% and 15%. Maximum errors occur for the lower velocities tests where the graphics are inaccurate and for the higher velocities tests where, as it is well known for the liquid metal tests, it is difficult to measure the mean temperature of fluid at the exit of the channel. Figure 8 shows the different values of the transfer coefficient in fully developed flow as a function of mean velocity in the channel. In the experimental field under consideration the influence of natural convection for low velocities tests may be seen : for a given width and heat flux, the heat transfer coefficient decreases when the flow-rate increases.

CORRELATION OF THE NUSSELT NUMBER IN FULLY DEVELOPED FLOW

As shown by HALLMAN [6] , variations of Nusselt number for the zone of fully developed flow may be presented as a function of the dimensionless group G^{*}/Re. On the figure 9, where all the results are produced in the plane Nu, G^{*}/Re, it can be seen that a correlation such as

$$Nu = 2 + 0,31 \left(\frac{G^{*}}{Re}\right)^{0,28}$$

may be used to compute the heat transfer coefficient within nearly $\pm$12%. The influence of Reynolds and Prandtl numbers; which appear when a dimensional analysis of the phenomenon is carried out, is not clearly marked when examining overall results. For the Prandtl number, no valid opinion can be given since the mean fluid temperature was subject to little variation during the course of the tests. Other experimenters such as BROWN [7] have used the parameter Z to correlate Nusselt number (Z is such as $Z^4 = \frac{Dh}{16b} \frac{G^*}{Re}$ for a rectangular cross section channel).

On the figure 10, we have compared our results with calculation and with other experimental results. There is a rather good agreement with the correlation proposed by VOLCHKOV [8] for his experimental results and with those of AMPLEEV [9] for a variation range of Z beeing between 10 and 20. For cases where full flow exists, using the equations of the boundary layer in laminar flow, the transverse velocity and temperature profiles can be determined, as well as Nusselt number, as a function of G^*/Re (or Z). Calculations have been carried out by OSTROUMOV [10] , HALLMAN [6] and BROWN [7] for mixed convection laminar flow in a tube. VERNIER [11] has established analytical solutions which have been adopted for the interpretation of our results. Figure 10 shows that our experimental results as those of VOLCHKOV and AMPLEEV are situated well above the theoretical curves obtained on the basis of laminar flow, even for low values of Z.

CONCLUSIONS

The results obtained in this study are of two orders :

- on the one hand, the interest attached to the adoption of the parameter G^{*}/Re has been tested. For the experimental conditions under consideration, the following empirical correlation is suggested :

$$Nu = 2 + 0{,}31 \left(\frac{G^{*}}{Re} \right)^{0{,}28}$$

Compared with results published in the literature, these results agree relatively well with those of VOLKCHOV et al.

- on the other hand, results of a more qualitative order have been confirmed :
  a) For low flow rates (Pe <200) heat transfer coefficient decreases when fluid velocity increases.
  b) The measurement of transverse distribution of temperature shows that a thermal boundary layer can be defined and as its thickness becomes constant a mean transfer coefficient may be calculated (existence of a full flow zone).
  c) Finally, in accordance with other experimenters, it has been found that the theoretical results for laminar flow give much higher Nusselt number values than the experimental ones.

NOTATIONS

b   Width of the test section

Dh   Hydraulic diameter of the channel

h   Heat transfer coefficient ($h = \frac{\varphi}{\Delta T}$)

G	GRASHOF number
G^*	Modified GRASHOF number ($G^* = G \times Nu$)
Nu	NUSSELT number
Pe	PECLET number
Pr	PRANDTL number
Re	REYNOLDS number
W	Mean fluid velocity
y	Transverse distance from the heated wall
z	Longitudinal distance from the beginning of the heating
Z	Non dimensional group ($Z^4 = \frac{Dh}{16b} \; \frac{G^*}{Re}$).
φ	Heat flux at channel wall.

Calculations of non dimensional number are made at the mean fluid temperature (for physical properties) and with hydraulic diameter (as characteristic length).

LITERATURE CITED

[1] M. WENDLING - Contribution à l'étude de la convection dans le sodium. Thèse de Docteur-Ingénieur, Faculté des Sciences de l'Université de Grenoble, 1970.

[2] G.F. SCHEELE and T.J. HANRATTY - Effect of natural convection instabilities on rates of heat transfer at low Reynolds numbers. A.I.Ch.E. Journal, 9, 2, 183-185 (1963)

[3] A. PIROVANO, S. VIANNAY, M. JANNOT - Convection naturelle en régime turbulent le long d'une plaque plane. 4ème Congrès International de Transfert de Chaleur PARIS-VERSAILLES, Papier N°N.C.1.8., (1970).

[4] N.S. ALFEROV, R.A. RYBIN and B.F. BALUNOV - Heat transfer with turbulent water flow in vertical tube under conditions of appreciable influence of free convection. Teploenergetika, 16, 2, 66-70, (1969).

[5] W.B. HALL and P.H. PRICE - Mixed forced and free convection from a vertical heated plate to air. 4ème Congrès International de Transfert de Cheleur, PARIS-VERSAILLES, papier N° N.C.3.3. (1970).

[6] T.M. HALLMAN - Experimental study of combined forced and free laminar convection in a vertical tube. NASA-TN-D-1104, (1961).

[7] W.G. BROWN - The superposition of natural and forced convection at low flowrates in a vertical tubes. Forsch. auf dem Geb. des Ing. 26 VDI-Forschungsheft 480, (1960).

[8] L.G. VOLCHKOV, M.K. GORTCHAKOV, P.L. KIRILLOV and F.A. KOZLOV - Echanges thermiques du sodium et de l'alliage sodium-potassium dans des tubes verticaux courts en convection mixte. "Métaux Liquides" Editions Moscou, pp. 15-32, (1967).

[10] G.A. OSTROUMOV - Free convection under the conditions of the internal problem. NACA-TM-1407, (1958).

[11] Ph. VERNIER - Le problèmes des propriétés physiques variables pour un écoulement monophasique dans un canal chauffant prismatique vertical. Int. J. of Heat and Mass Transfer, 13, pp. 1199-1214, (1970).

CN Na LOOP

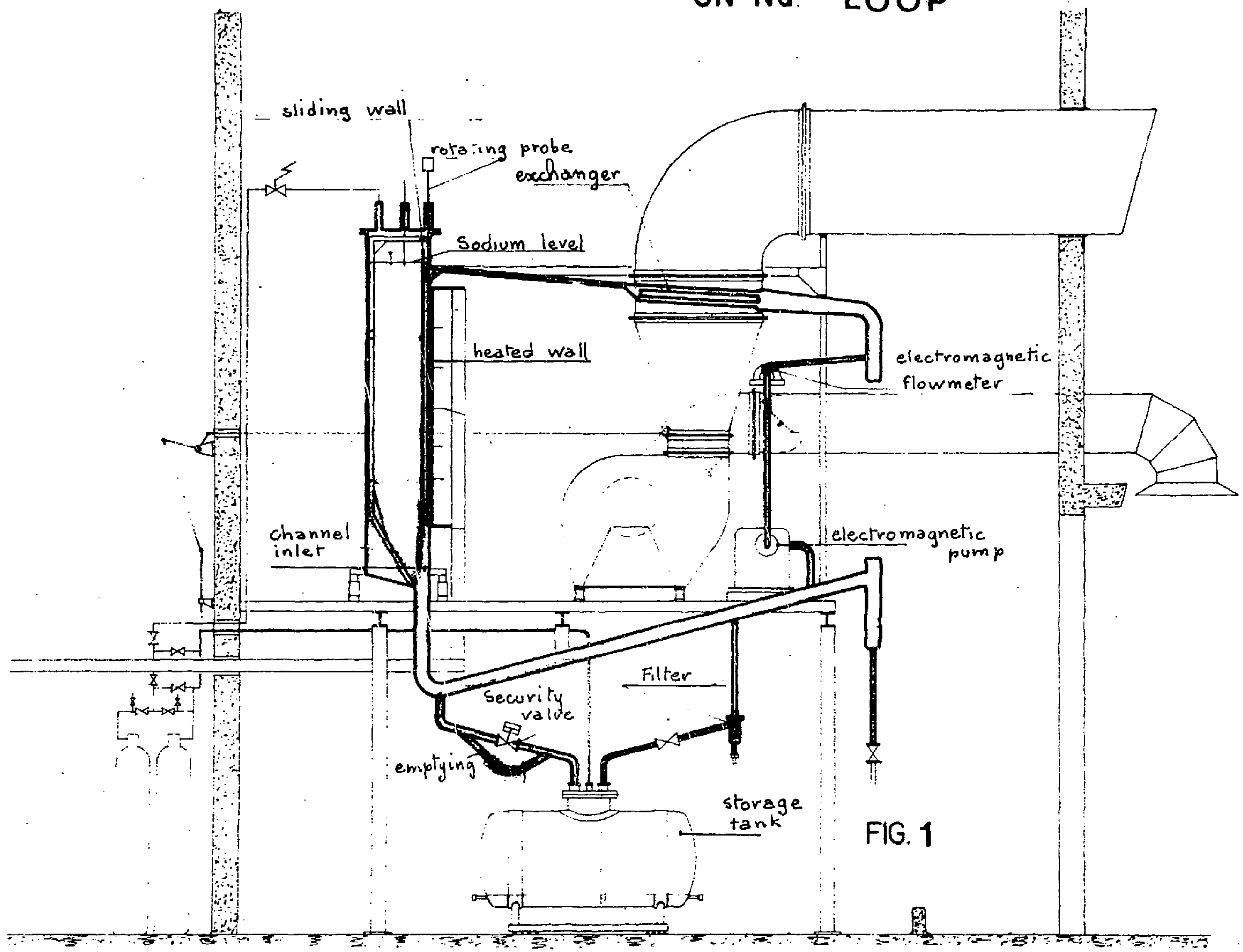


FIG. 1

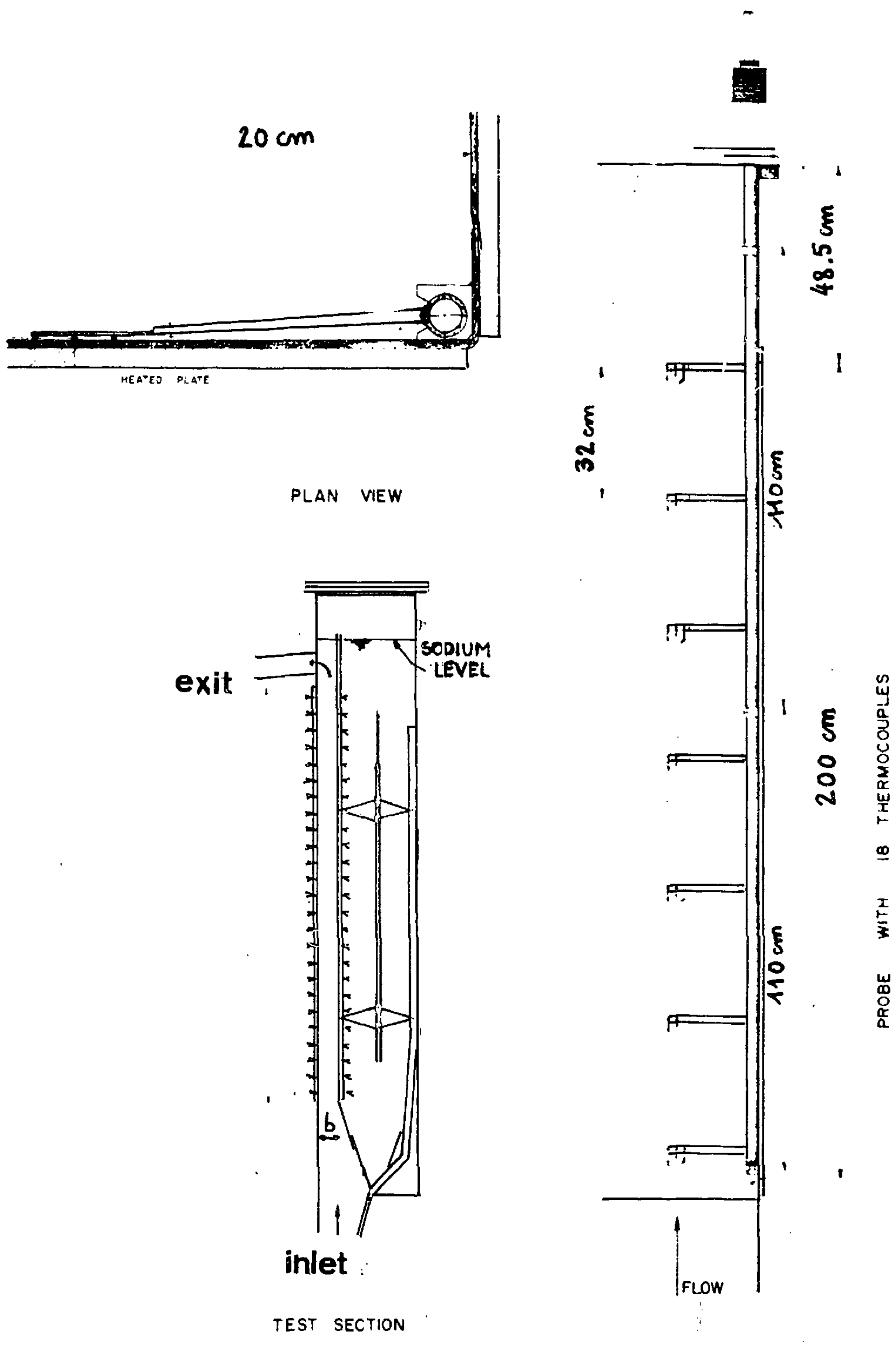
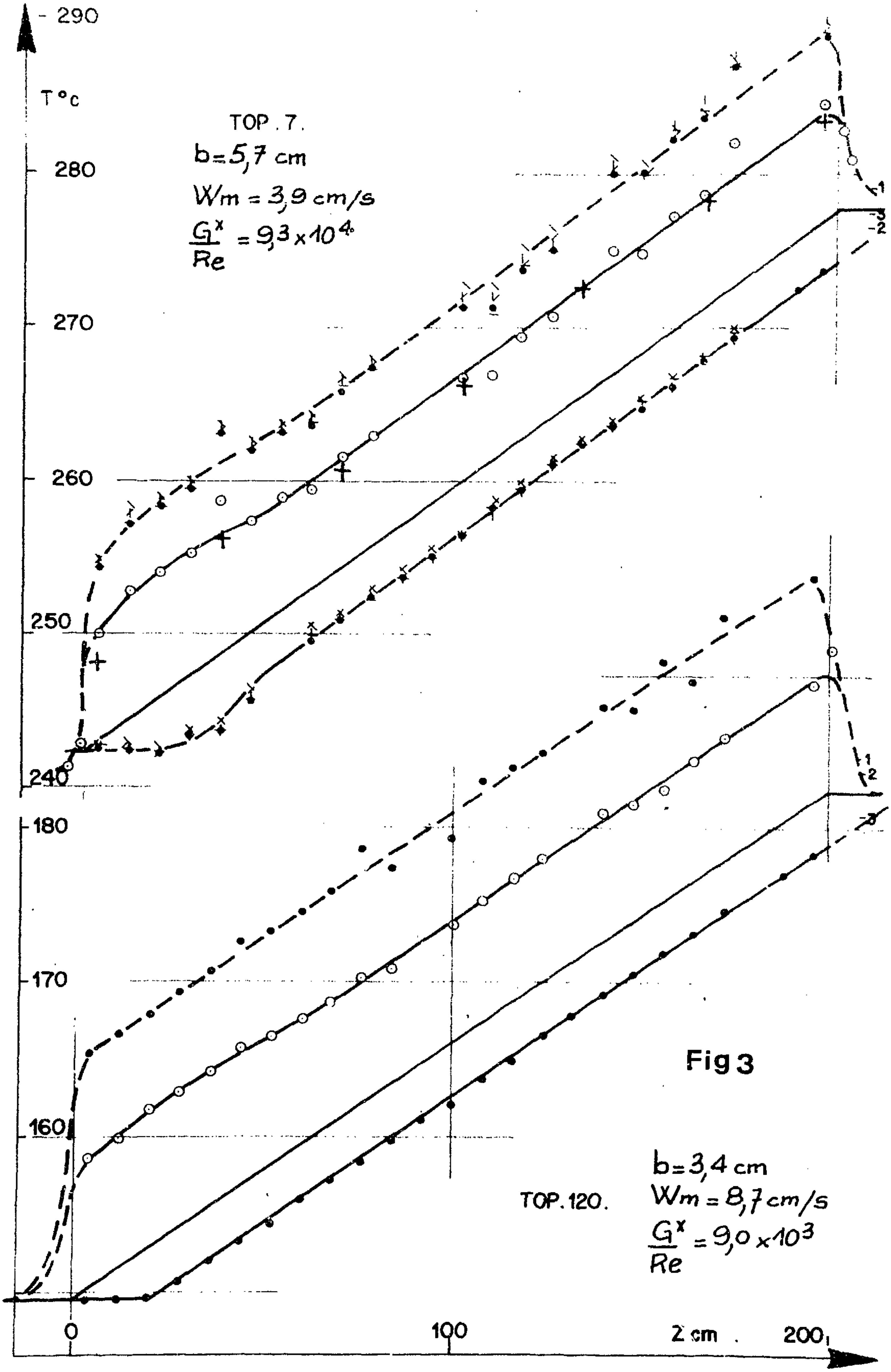


FIG. 2 INSTRUMENTATION OF THE TEST SECTION.



290
T °c
TOP . 7.
b = 5,7 cm
Wm = 3,9 cm/s
G^x / Re = 9,3 x 10^4
280
270
260
250
240
180
170
160
TOP. 120.
b = 3,4 cm
Wm = 8,7 cm/s
G^x / Re = 9,0 x 10^3
0
100
200
Z cm
1
2
3

Fig 3

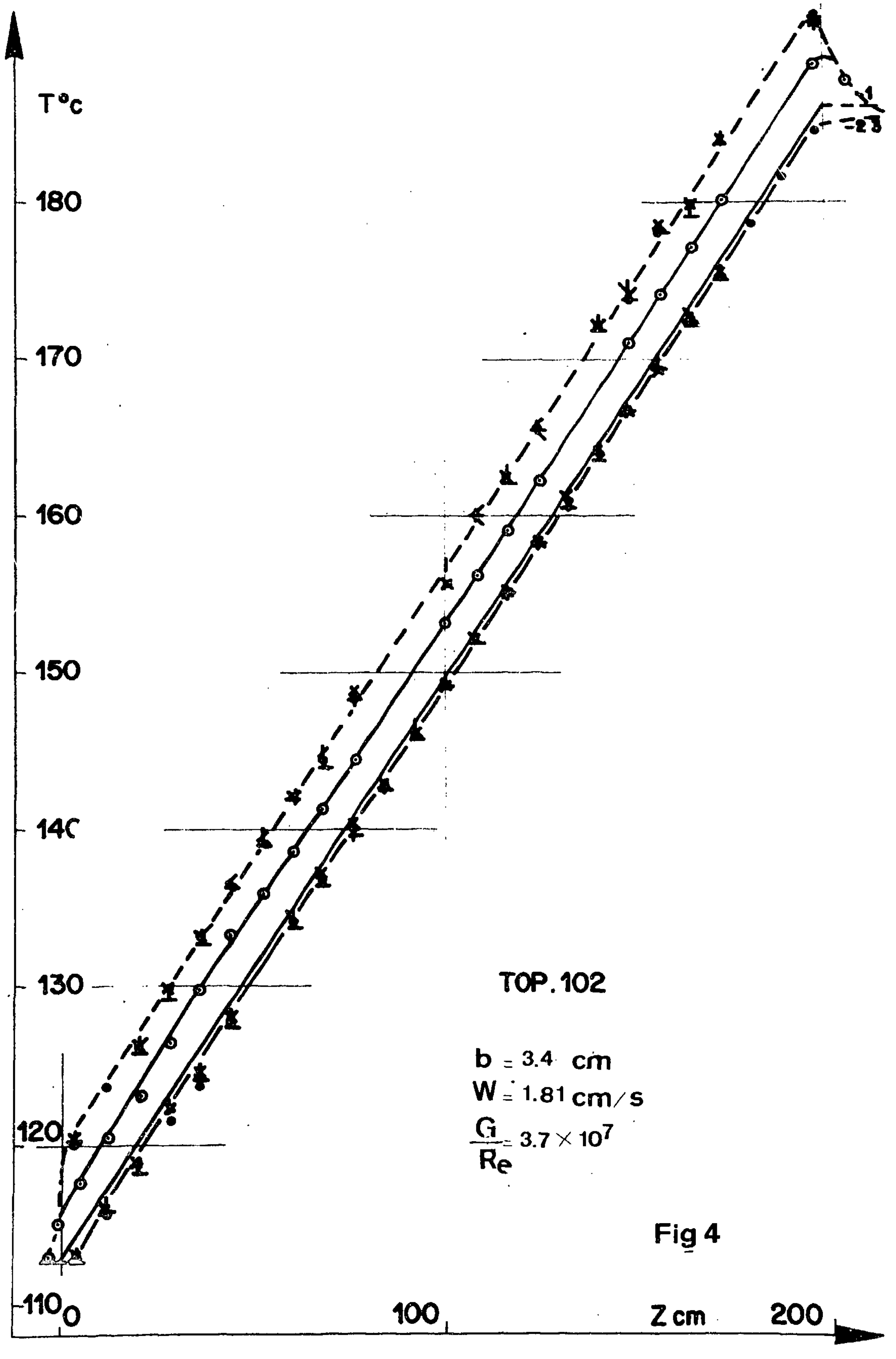

T °c
180
170
160
150
140
130
120
110
0
100
200
Z cm
TOP. 102
b = 3.4 cm
W = 1.81 cm/s
G/Re = 3.7 × 10^7
1
2
3

Fig 4

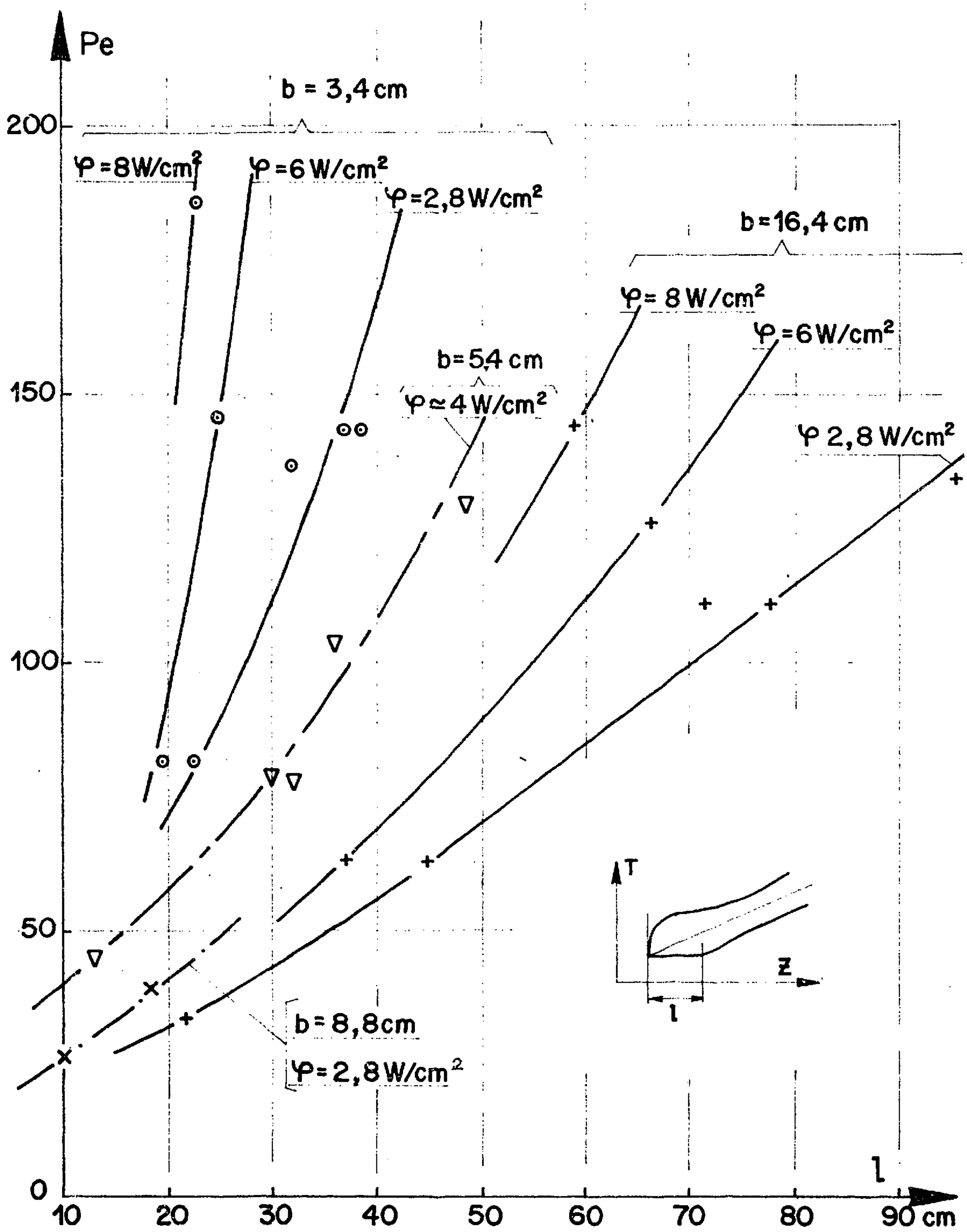
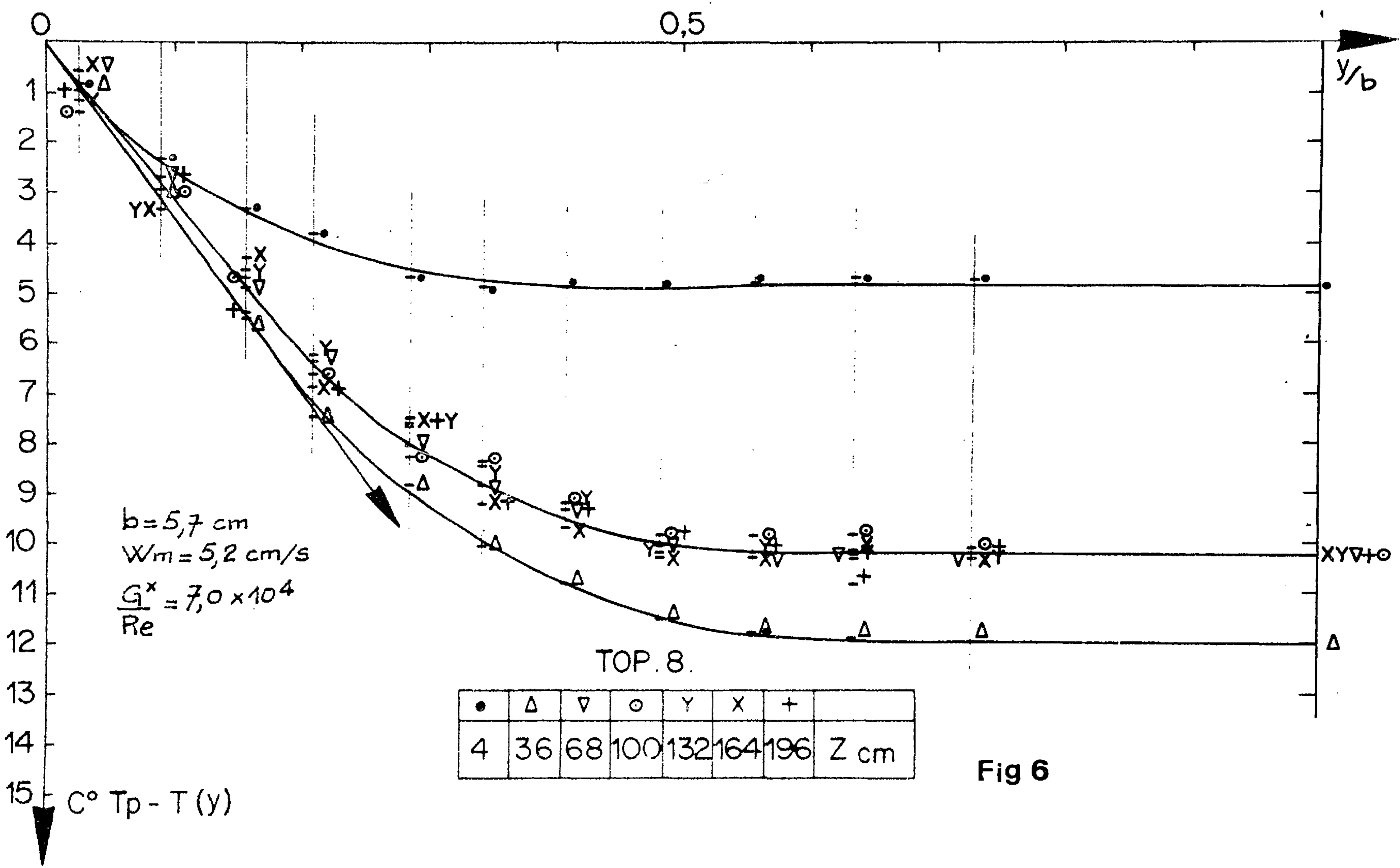
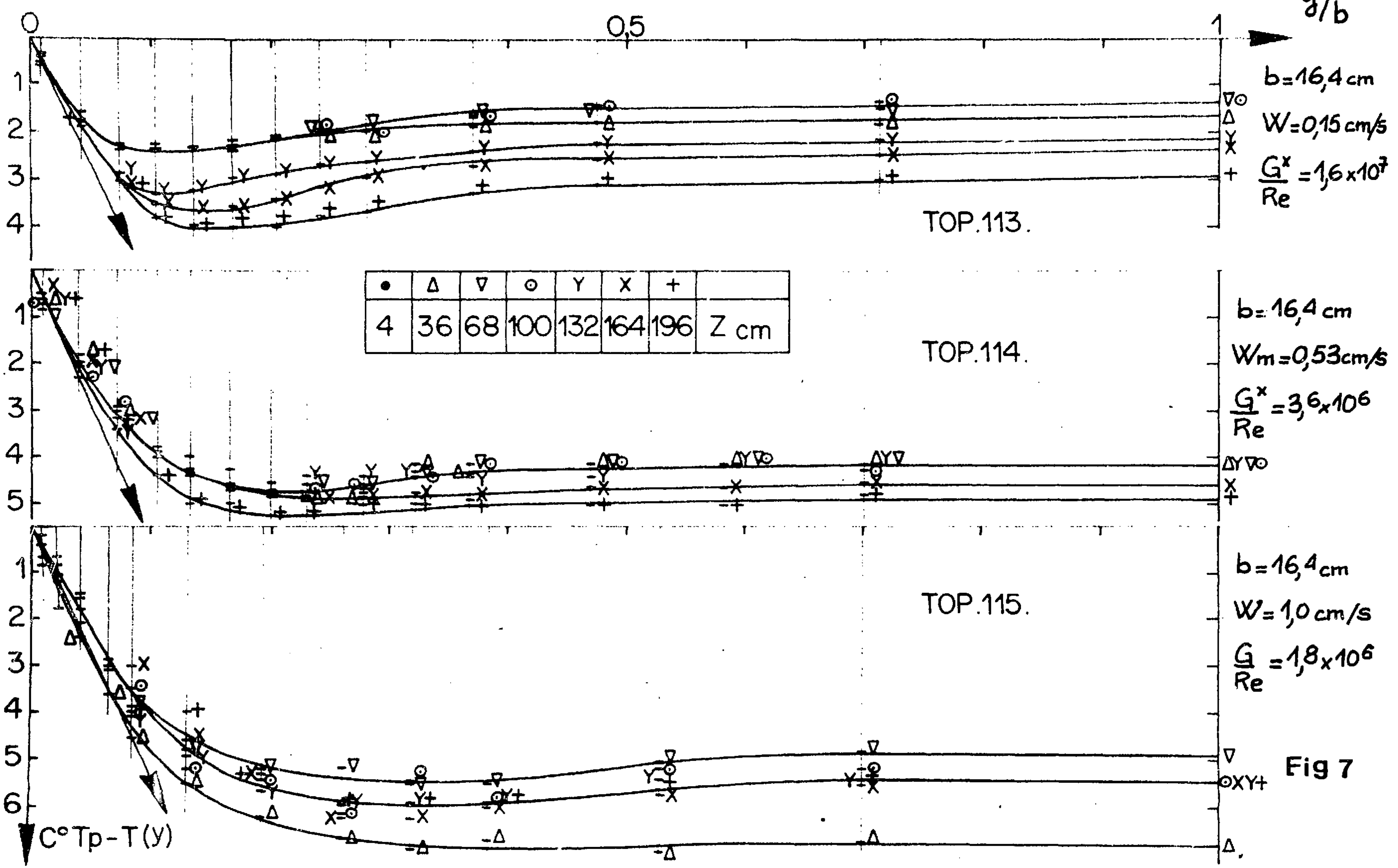

Pe
b = 3,4 cm
φ = 8 W/cm²
φ = 6 W/cm²
φ = 2,8 W/cm²
b = 16,4 cm
φ = 8 W/cm²
φ = 6 W/cm²
b = 5,4 cm
φ ≃ 4 W/cm²
φ 2,8 W/cm²
b = 8,8 cm
φ = 2,8 W/cm²
T
Z
l
200
150
100
50
0
10
20
30
40
50
60
70
80
90 cm
l

Fig 5



•	Δ	∇	⊙	Y	X	+	
4	36	68	100	132	164	196	Z cm

Fig 6



y/b
0
0,5
1
b=16,4 cm
W=0,15 cm/s
G^x/Re = 1,6x10^7
TOP.113.
• Δ ∇ ⊙ Y X +
4 36 68 100 132 164 196 Z cm
TOP.114.
b=16,4 cm
Wm=0,53cm/s
G^x/Re = 3,6x10^6
TOP.115.
b=16,4 cm
W=1,0 cm/s
G/Re = 1,8x10^6
C° Tp-T(y)

Fig 7

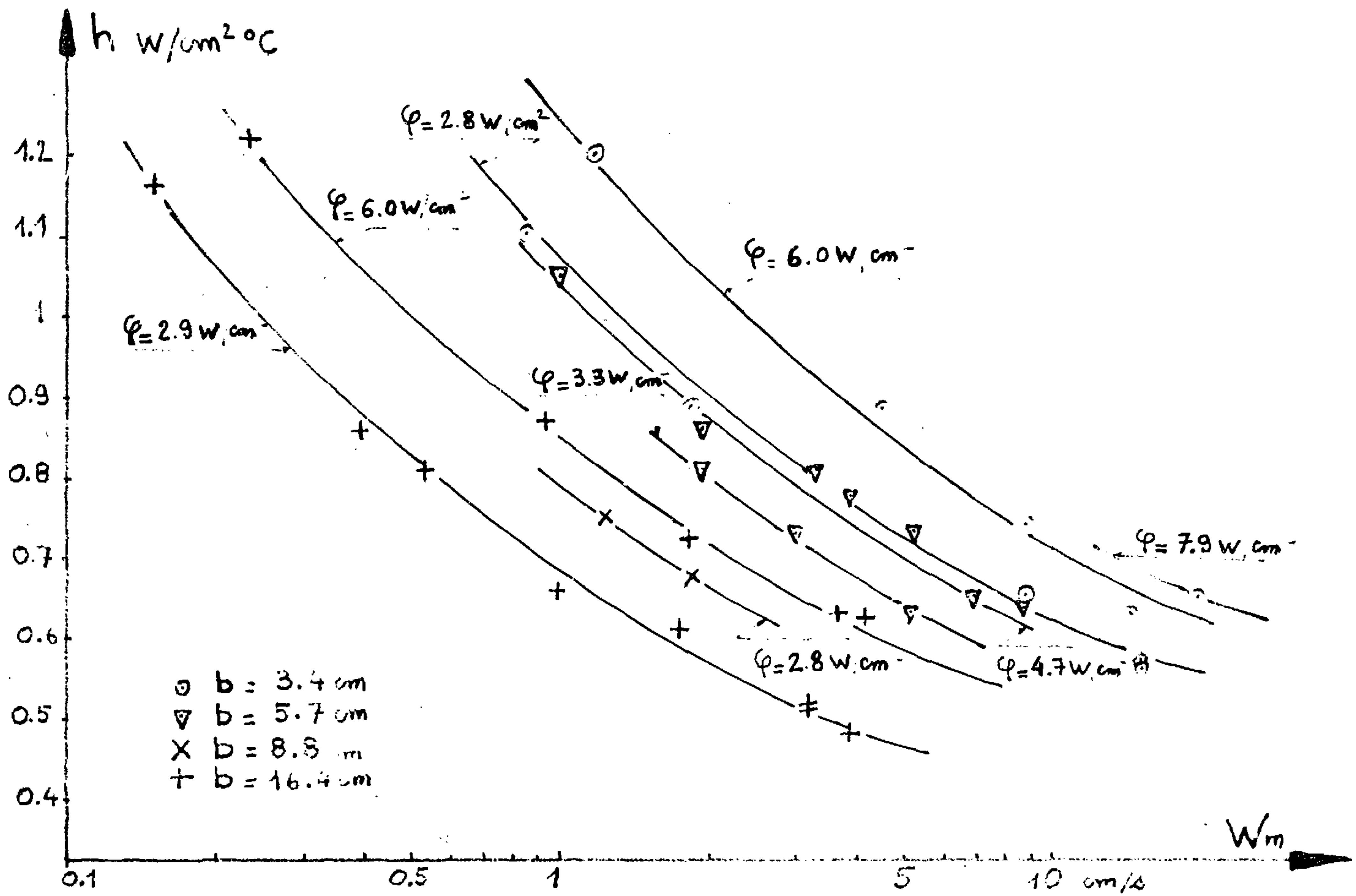
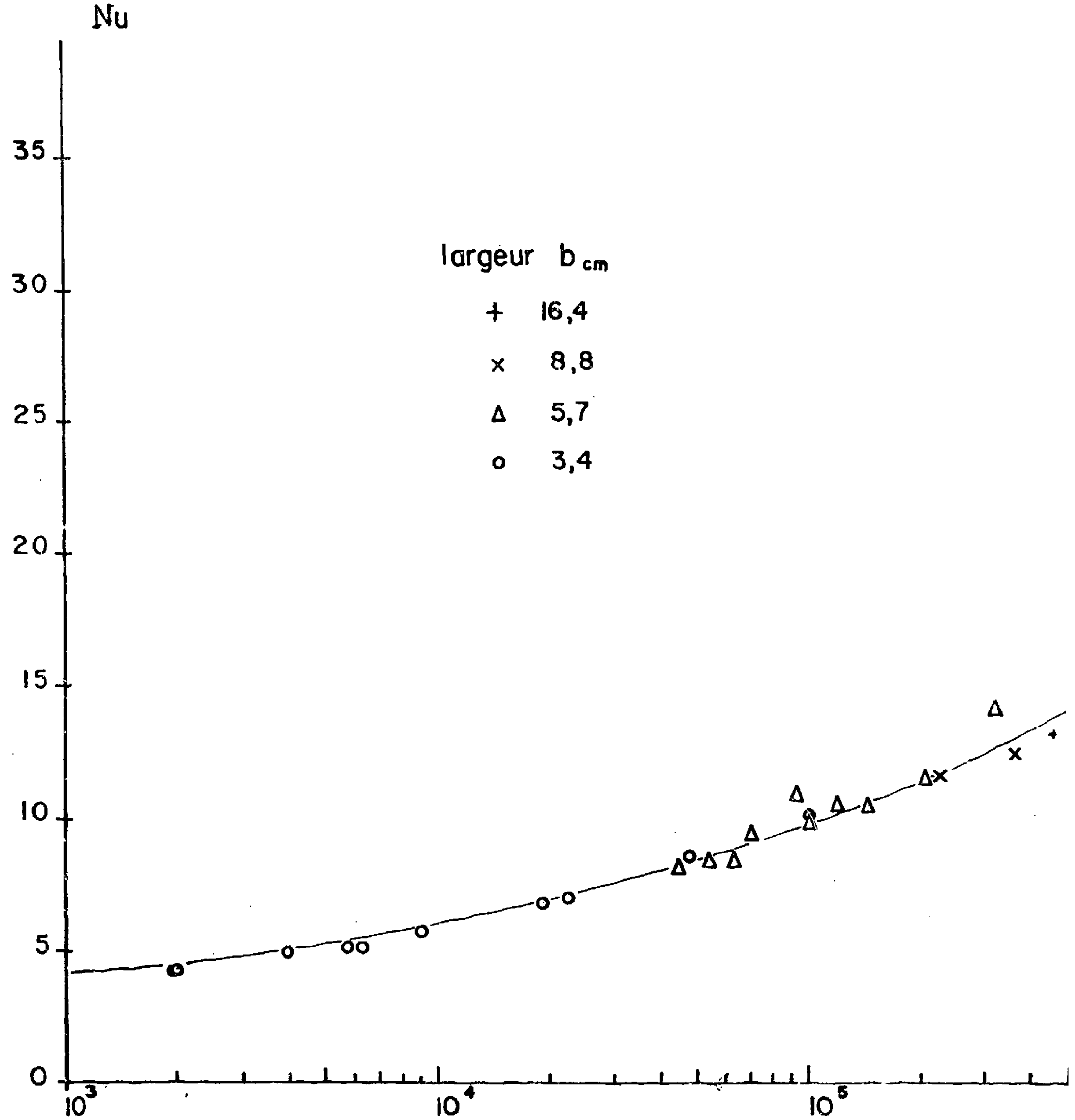

h W/cm² °C
1.2
1.1
1
0.9
0.8
0.7
0.6
0.5
0.4
φ= 2.8 W/cm²
φ= 6.0 W/cm²
φ= 6.0 W/cm²
φ= 2.9 W/cm²
φ= 3.3 W/cm²
φ= 7.9 W/cm²
φ= 2.8 W/cm²
φ= 4.7 W/cm²
b = 3.4 cm
b = 5.7 cm
b = 8.8 cm
b = 16.4 cm
Wm
0.1
0.5
1
5
10 cm/s

Fig 8



Nu
35
30
25
20
15
10
5
0
largeur b cm
+ 16,4
× 8,8
Δ 5,7
o 3,4
10^3
10^4
10^5

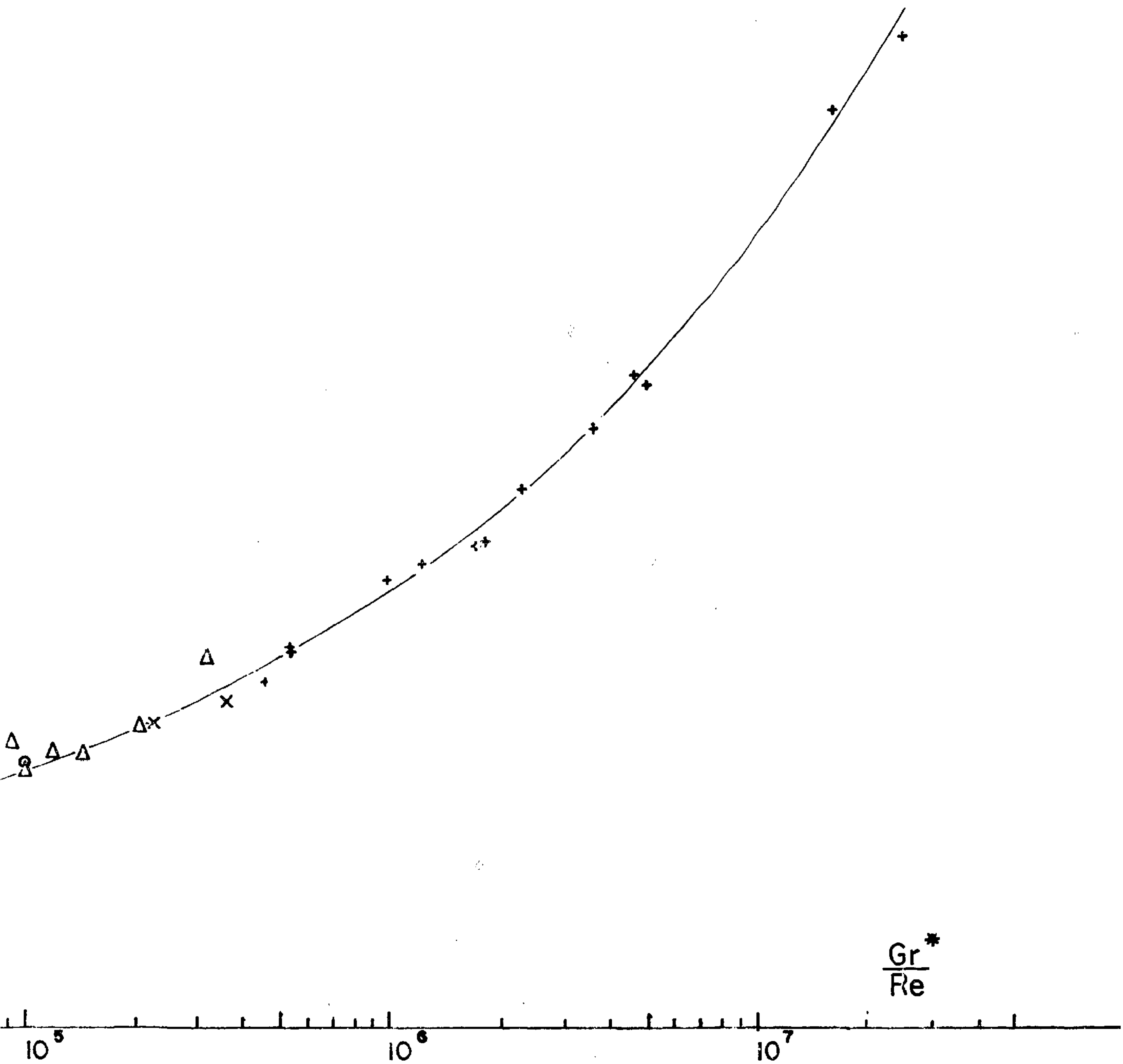


FIG. 9

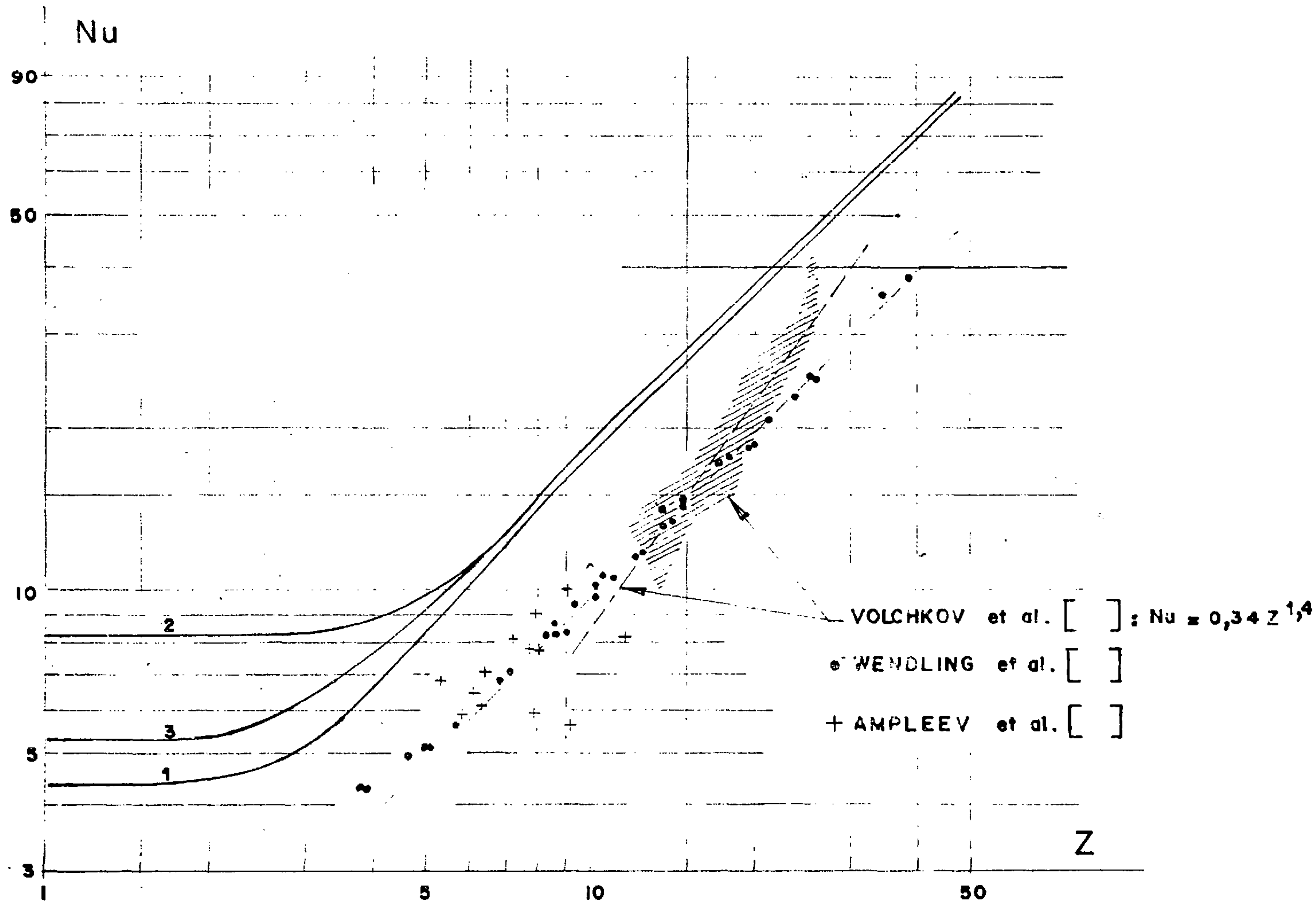

Nu
90
50
10
5
3
1
5
10
50
Z
2
3
1
VOLCHKOV et al. [ ] : Nu = 0,34 Z 1,4
WENDLING et al. [ ]
+ AMPLEEV et al. [ ]

FIG.10.