VIBRATION PERFORMANCE OF LOW HEAD CENTRIFUGAL PUMPS USED FOR IRRIGATION

By

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1 - ABSTRACT

This research is the first one of a set aims to have some information about the vibration performance of low head centrifugal pumps under the different operating conditions.

The objective of this research is to investigate the pump's vibration performance and the effect of the suction system type on its level.

2 - INTRODUCTION

Vibration of moving or rotating parts of machines has a harmfull effects. One of these effects is the failer of machine parts, but another, if failer is stillfar enough, is the noise leading to discomfort.

Studies of failer of machine parts due to vibration belongs to machine design, meanwhile the conditions leading to discomfort are studied under the title of noise or accustics.

The wide spread of centrifugal pumps in practical life urges optimization of design and performance of centrifugal pumps.

This work is devoted to study of vibration of centrifugal pumps unsing three different types of suction system, namely steel piping, flexible hose, and plastic pipe.

The fluid used is drinking water at normal temperature.

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3 - **NOTATIONS**

A : Mean amplitude of vibration (in .m. Pk-Pk.).

$A_{x.f}$: Mean amplitude in X-direction when using the flexible hose.

$A_{x.p}$: Mean amplitude in X-direction when using the plastic pipe.

$A_{x.st}$: Mean amplitude in X-direction when using the sted pipe.

$A_{y.f}$: Mean amplitude in Y-direction when using the flexible hose.

$A_{y.p}$: Mean amplitude in Y-direction when using the plastic pipe.

$A_{y.st}$: Mean amplitude in Y-direction when using the steel pipe.

a : Acceleration of vibration (in m/sec^2. g-Pk).

$a_{x.f}$: Acceleration in X-direction when using the flexible hose.

$a_{x.p}$: Acceleration in X-direction when using the plastic pipe..

$a_{x.st}$: Acceleration in X-direction when using the steel pipe..

$a_{y.f}$: Acceleration in Y-direction when using the flexible hose.

$a_{y.p}$: Acceleration in Y-direction when using the plastic pipe..

$a_{y.st}$: Acceleration in Y-direction when using the steel pipe..

H_d : Delivery head ($m.H_2O$).

H_s : Suction head (Cm.Hg).

H_t : Total head ($m.H_2O$).

N_m : Motor horsepower (HP).

P_1 : Pressure up-stream ($m.H_2O$).

P_2 : Pressure down-stream (Cm.Hg).

Q : Rate of flow (lit./sec.).

: Specific weight. (Kg/m^3).

Subscribts:

$(\quad)_s$: Suction side.

$(\quad)_d$: Delivery side.

4 - SOURCES OF VIBRATION IN CENTRIFUGAL PUMP INSTALLATIONS:

In addition to the mechanical sources of vibration and noise in centrifugal pumps as dynamically operated machine, there are other sources of noise and vibration.

a) Cavitation:

The term cavitation refers to the conditions within the pump where, owing to a local pressure drop, cavities filled with water vapour are formed. These cavities collapse as soon as the vapour bubbles reaches regions of higher pressure on their way through the pump.

Phenomenon of cavitation is well known long time ago*1. In the last time a deep study of mechanism and mechanics of cavitation established and experimentally approved favourate to predicate cavitation*2.

The induced by cavitation vibration may lead to mechanical failer of machine parts*3-4 or even corrosion due to the betting effect of cavitation*5.

No valuable effort was directed for studying of amplitude of vibration due to cavitation.

Cavitation may occur in the suction piping system or in the impeller of the pump. The generated vibration spreads through the mechanical system to sourrounding media and foundation producing noise and may lead to failer.

b) Separation:

Separation of flow is that hydrodynamic phenomenon occuring in case of positive pressure gradient flow, leading to separation of stream lines of flow and forming local zones of unsteady character. The unsteadines of flow in such zones forms another source of vibration. Separation of flow, especially behind rear edge of blades of impellers, forms vortex way in which the flow is turbulent.

c) Intensity of Turbulence:

It is well known that flow in the most of hydraulic machines is turbulent. Turbulent flow forms a source of sound and vibration. A high intensity of turbulence in most cases is due to mechanical vibration of machine elements*6.

d) Pumping System:

In addition to mechanical vibration of the pump it-self the pumping system including piping and valves may represent a source of vibration. The interaction of flow, pipes and valves from one side and or the interaction between the pump and pumping system in addition of foundation as rigidly connected parts, from the other side, form a compound with mutual influence vibrating system.

The existence of all these vibration generating sources in the pumping system and their mutual influence, shows that the theoretical study of vibration performance of centrifugal pump in pumping system is extensively difficult. The need for vibration performance of pumps urges the experimental direction to give some informations about the problem, which may clearify the relative effect of several parameters. The results obtained from such studies may be of a great value for pump selection and pumping system design.

5 - EXPERIMENTAL WORK

The arrangement of the system used for these investigations is shown in Fig. (1). It consists of the tested centrifugal pump, with the following specifications:
Horse power 10 HP., suction diameter 6", delivery diameter 6", R.P.M. 1450., main supply tank, delivery tank, suction line which was of three kinds, steel pipe, flexible hose, and plastic pipe, and the measuring instruments.

The rate of flow was measured by a calibrated orificemeter. Each of suction head, pressure down stream was measured by two calibrated vacuum meters. Another two calibrated pressure gauges were used for measuring both delivery head and pressure up-steam.

Using a wattmeter: the motor power was obtained in (Kw.).

The mean value of the amplitude of the vibration wave was measured in two perpendicular directions at each suction and delivery sides by aid of the portable vibration analyser model 2100 with AV 100 P accelerometer. The amplitude waves were recorded by a chart-recorder connected with the analyser. All the measured amplitude values were in the wide band frequency of 10-100 Hz.

Also, using the vibration analyser, the acceleration of the vibration was measured in each direction.

All the experimental data were collected for three cases:

a) Steel suction pipe of 6". diameter,
b) Flexible hose of 6". diameter,
c) Plastic pipe of 6" diameter.

The performance of the pump was controlled by the delivery valve. The measured quantities were: H_s (Cm.Hg.), H_d (m H_2O), $\frac{P_1}{\gamma}$ (m.H_2O), $\frac{P_2}{\gamma}$ (Cm.Hg.), N_m (Kw.), A_m (μ.m), and a_m (m/sec^2).

The calculation procedure was as shown below:

Total head $H_t = H_s$ (Cm.Hg.) x 0.13595 + H_d(m.H_2O) ....(m H_2O).

Rate of flow $Q = 16.907\sqrt{\left(\frac{P_1}{\gamma}\right) + 0.13595\left(\frac{P_2}{\gamma}\right)}$ (Lit/sec.).

The motor power $N_m = N_m$(Kw.) x 1.36 .... HP.

The mean amplitude $A_m = \frac{A_{max.} + A_{min.}}{2}$ ....(μ.m).

$$a_m = \frac{a_{max.} + a_{min.}}{2} \quad ....(\mu m/sec^2).$$

The obtained hydraulic characteristic curves are shown in Fig. (2).

The accuracy of the cellected data were calculated and found to be: H = H ± 5%,

N = N ± 5%,

A = A ± 3.3%,

a = a ± 8%.

6 - DISCUSSION

It must be here imphasised that the minimum pressure achieved was 22 Cm.Hg. vacuum. The temperature of water was ranging from 20-23°C. The vapour pressure at the working temperature is about 72 Cm.Hg. vacuum. Therefore; there is no doupt that cavitation was impossible to occur. In order to achieve the cavitational regime, it is recommended to use a special deep vacuum pump.

The results of the experimental work treated as mentioned before, show that the amplitude of vibration existing in different positions on the pumping system widely varies.

The main tendency is that amplitude $(A_x)_s$ in case of flexible hose is about 2 times of that in case of steel pipes (Fig. 3). Meanwhile the amplitude of the plastic pipes is about 3 times of that of the steel pipe.

Although it was expected that the amplitude level in case of the flexible hose is less than that in case of the steel pipe owing to the work flexability of the hose; the experimental results showed the inverse, the amplitude $(A_x.f)_s$ is about 2 times of that $(A_{x.st})_s$. This can be refered to the surface irregularity of the flexible hose, which represents an additional source of vibration.

The high amplitude in case of plastic pipe may be explained on the basis of its specific weight. The most light material of pipes used, is the plastic, hense, the amplitude is the maximum.

The $(A_{x.st})_s$ behaviour differs from the $(A_{x.f})_s$ in region of small flow rates. At small Q, the $(A_{x.st})_s$ is high and have nearly the same value as $(A_{x.f})_s$. Increasing Q the amplitude $(A_{x.st})_s$

decreases sharply meanwhile $(A_{x.f})_s$ tends to increase. The minimum recorded amplitude $(A_{x.st})_s$ lays against a peak for $(A_{x.f})_s$ at $Q = 24$ lit./sec. For higher flow rates, the tendency of both kinds of piping is the same, keeping a difference about 100% of $(A_{x.st})_s$.

Concerning the $(A_{x.p})_s$, the level is nearly constant at low flow rates, with increasing tendency with increasing discharge.

The $(A_{y.st})_s$ behaviour has the same features, except for small flow rate, the first recorded peak in $(A_{y.st})_s$ refers to the first minimum value for $(A_{y.f})_s$ which occurs at lower rate of flow (15 lit/sec.). The lvel of $(A_{y.f})_s$ is higher than $(A_{y.st})_s$ by about 30% as can be seen from Fig. (4).

The amplitude $(A_{y.p})_s$ is somewhat lower than that $(A_{y.f})_s$.

It is noticed that the $(A_x)_s$ is more bigger than $(A_y)_s$. This is connected with the direction of flow in the foot valve, which was noticed to open about an axis parallel to (y) axis.

Measurments on the delivery side shown on Figs(5,6) indicates that the level of the amplitude is lower than that on the suction side. This was expected since the strong source of vibration is the foot valve, which is mounted on the far and of the suction line. The induced by the foot valve vibration reaching the measuring point on the delivery side, suffers from damping effect of the pump, casing, and foundation more than the point on the suction side do.

It is noticed, for vibration in both directions (x,y) on the delivery side, that the level of $(A_{st})_d$ is higher than $(A_f)_d$. This may means that the damping effect of the system in case of flexible has is more than that in case of steel pipe.

The amplitude of vibration on the delivery side in y-direction in case of the plastic pipe $(A_{y.p})_d$ is nearly constant at a level lower than that for the other two cases.

Comparison between the vibration in the X-direction on suction and delivery sides shows that the $(A_{x.f})_s$ is about 7 times its value on the delivery side, which means that the damping effect of the system in this case is very big. For the steel pipe, the $(A_{x.st})_s$ is about 2 times its value on the delivery, which means that the damping effect of the system in this case is less than the previous one.

The ratio of the amplitude in X-direction on the suction side to that on the delivery in case of the plastic pipe is about 13 times.

In Y-direction, the $(A_{y.f})_s$ is about 6 times of $(A_{y.f})_d$, meanwhile $(A_{y.st})_s$ reaches only 1.7 times of $(A_{y.st})_d$, and the $(A_{y.p})_s$ about 3 times of $(A_{y.p})_d$.

The variation of the amplitude as a function of the suction head, (Figs. 7,8,9,10) is the same as the relation $(A - Q)$, drawn with a different scale. This scale is the relation between H_s and Q.

Concerning the acceleration in the measuring points, it is noticed that its variation is nearly straight line with the rate of flow variation. The shut off acceleration was noticed to be the same as in steady condition, but decaying with time.

The acceleration $(a_{x.f})_s$ is about 1.2 times of the $(a_{x.st})_s$. Meanwhile the $(a_{x.p})_s$ is about 1.6 times of the same $(a_{x.st})_s$, (Fig. 11).

The $(a_{y.f})_s$ is nearly the same of the $(a_{y.st})_s$, but the $(a_{y.p})_s$ is about 1.3 times of the $(a_{y.st})_s$. (Fig. 12).

It is very clear that $(a_x)_s$ for the three kinds is more bigger than those $(a_y)_s$.

Measurments on the delivery side (Fig. 13, 14) show that $(a_{x.st})_d$ and $(a_{y.st})_d$ are somewhat higher than $(a_{x.f})_d$, $(a_{y.f})_d$,

$(a_{x.p})_d$ and $(a_{y.p})_d$.

Comparison between the acceleration in the X-direction on suction and delivery sides shows that the $(a_{x.st})_s$ is about 1.9 times of the $(a_{x.st})_d$. For the flexible hose, the $(a_{x.f})_s$ is about 3 times of the $(a_{x.f})_d$, meanwhile the $(a_{x.p})_s$ equal nearly 3.9 times of the $(a_{x.p})_d$.

In Y-direction, the $(a_{y.st})_s$ has nearly the same level of the $(a_{y.st})_d$. But the $(a_{y.f})_s$ is about 1.5 times of the $(a_{y.f})_d$ meanwhile the $(a_{y.p})_s$ reaches 2 times of the $(a_{y.p})_d$.

CONCLUSIONS

1 - The level of the vibration at entrance of centrifugal pumps varies in accordance with the piping system material. The plastic pipes showed the highest level and the steel pipes showed the lowest.

2 - The origin of the high level of vibration on suction side is the usually used foot valve. It is recommended, for purpose of reducing vibration at entrance of the centrifugal pumps, to find a new construction for the foot valves.

3 - Inspite of the wide use of the flexible hose with centrifugal pumps used for irrigation, it is not the suitable form of piping from vibration point of view because of the high level of amplitude.

4 - The recorded in this paper vibration is mainly due to, separation, turbulence, and mechanical vibration of the mechanical parts, but not due to cavitation, since the minimum achieved suction pressure is still faraway from vapour pressure.

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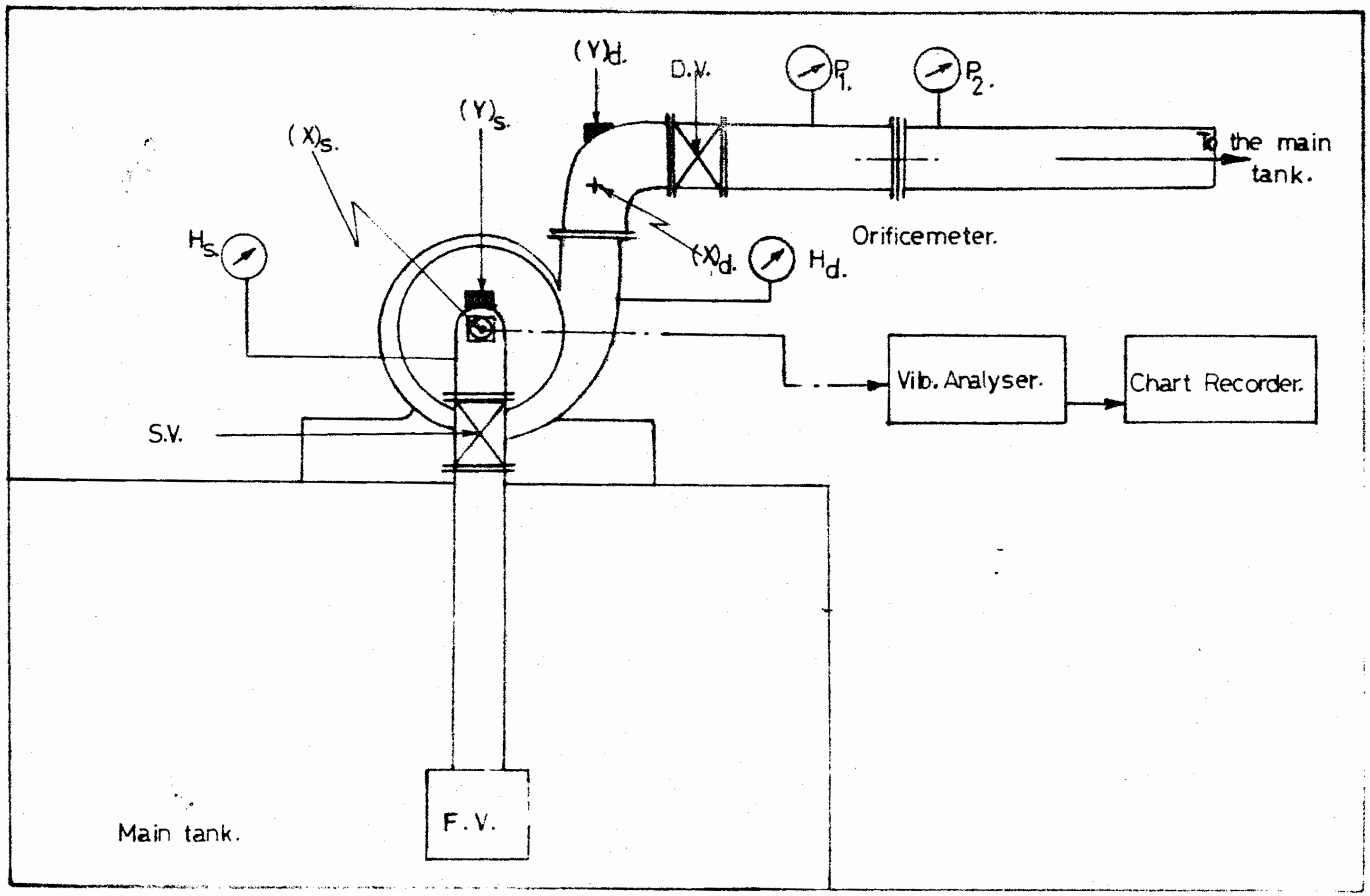


(Y)d.
D.V.
P1.
P2.
(Y)s.
(X)s.
To the main tank.
Hs.
(X)d.
Hd.
Orificemeter.
Vib. Analyser.
Chart Recorder.
S.V.
F.V.
Main tank.

Fig.(1)

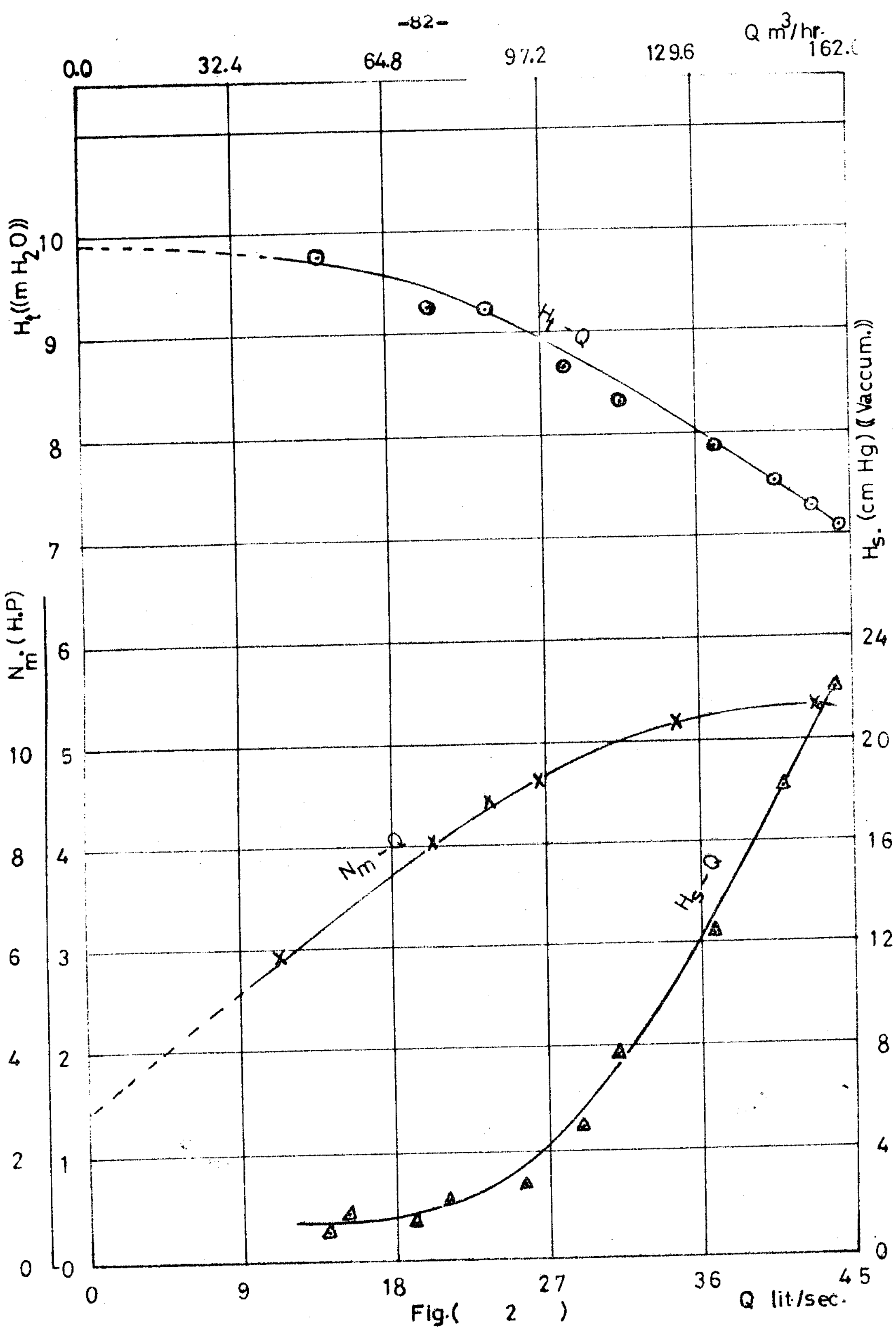

Q m3/hr.
0.0
32.4
64.8
97.2
129.6
162.0
Ht ((m H2O))
10
9
8
7
6
5
4
3
2
1
0
Nm. (H.P)
10
8
6
4
2
0
Hs. (cm Hg) (Vaccum.)
24
20
16
12
8
4
0
Ht - Q
Nm - Q
Hs - Q
0
9
18
27
36
45
Q lit/sec.

Fig.(2)

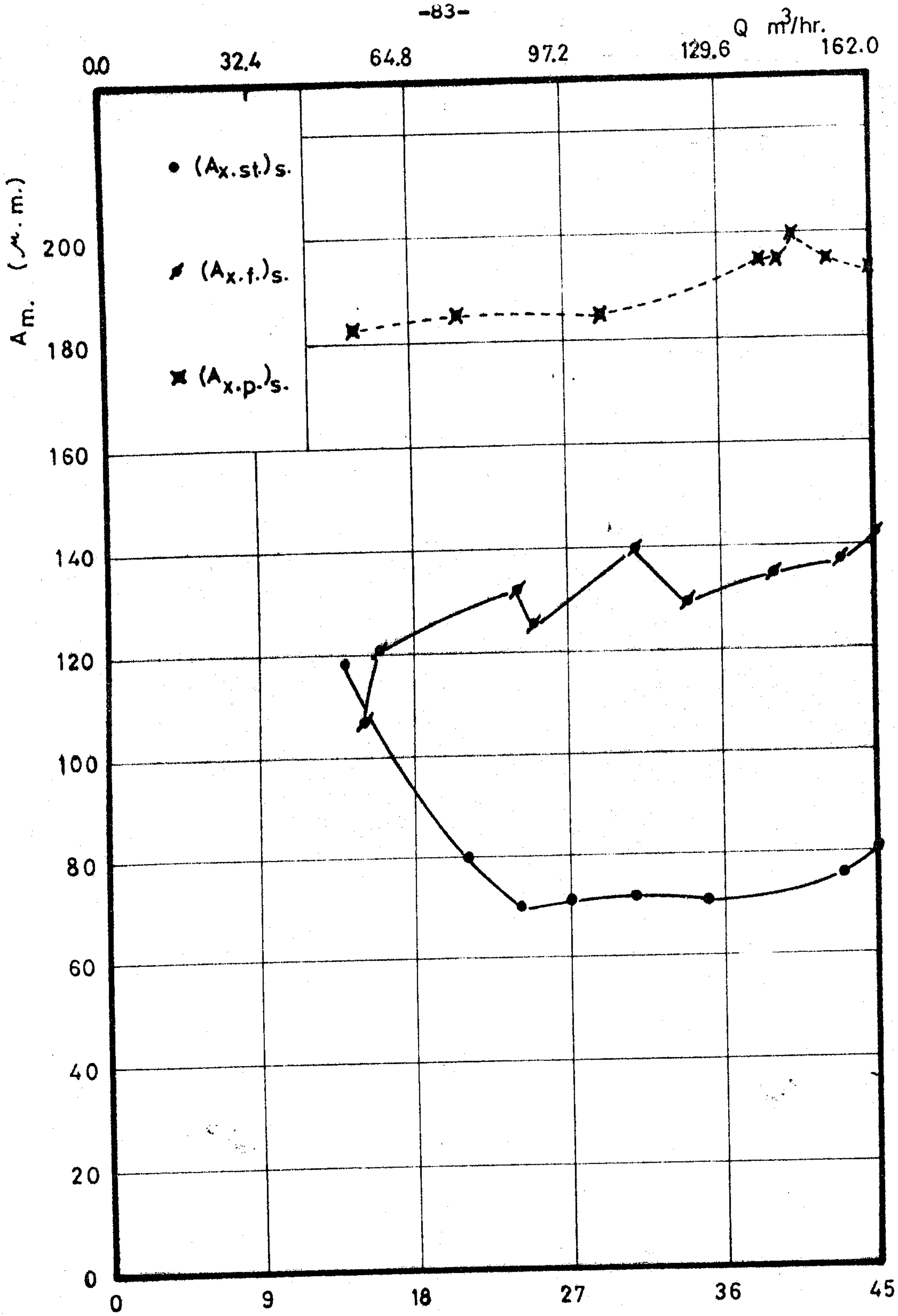

Q m³/hr.
0.0
32.4
64.8
97.2
129.6
162.0
● (Ax.st.)s.
(Ax.f.)s.
(Ax.p.)s.
Am. (μ. m.)
200
180
160
140
120
100
80
60
40
20
0
0
9
18
27
36
45
Q lit/sec.

Fig.(3).

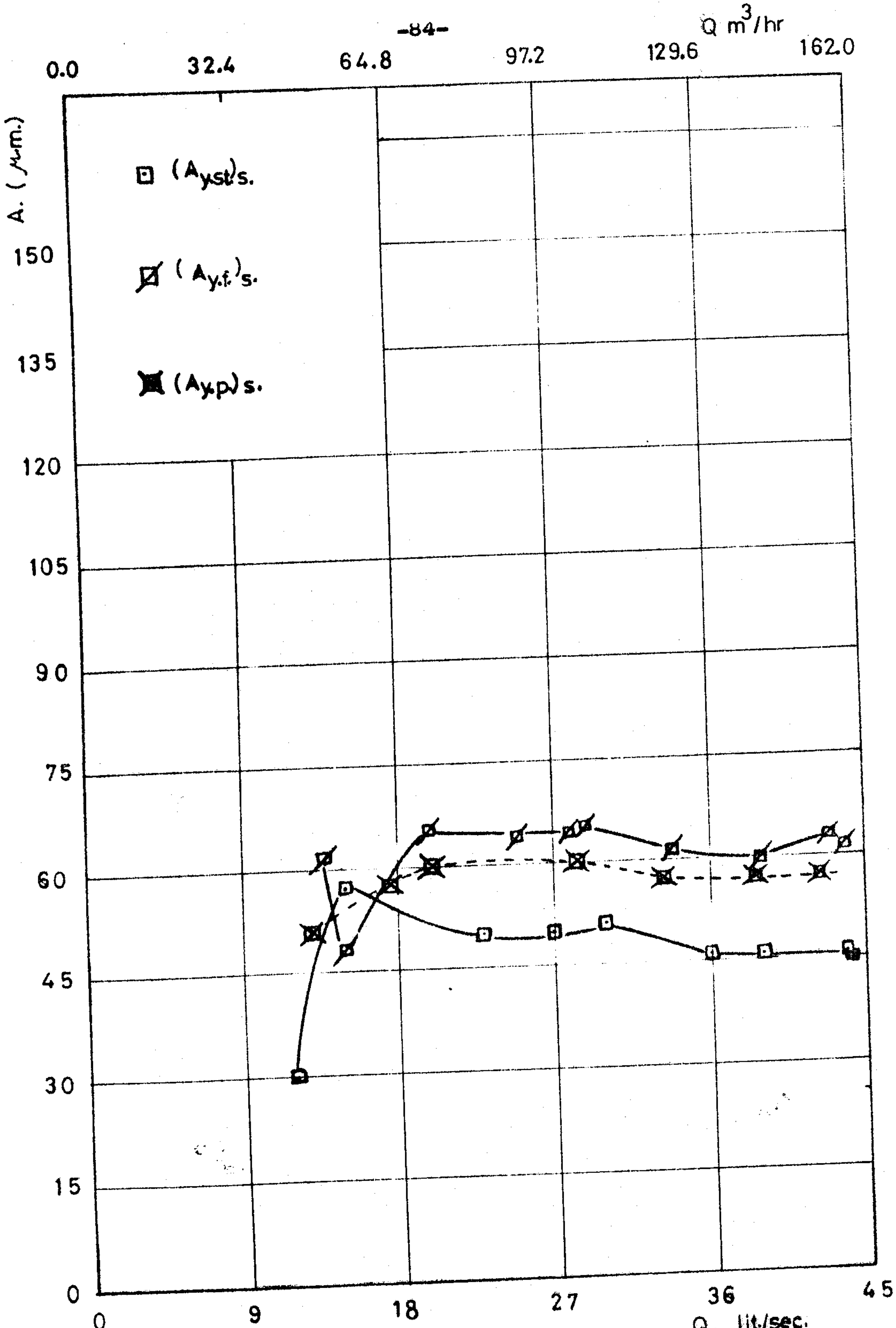
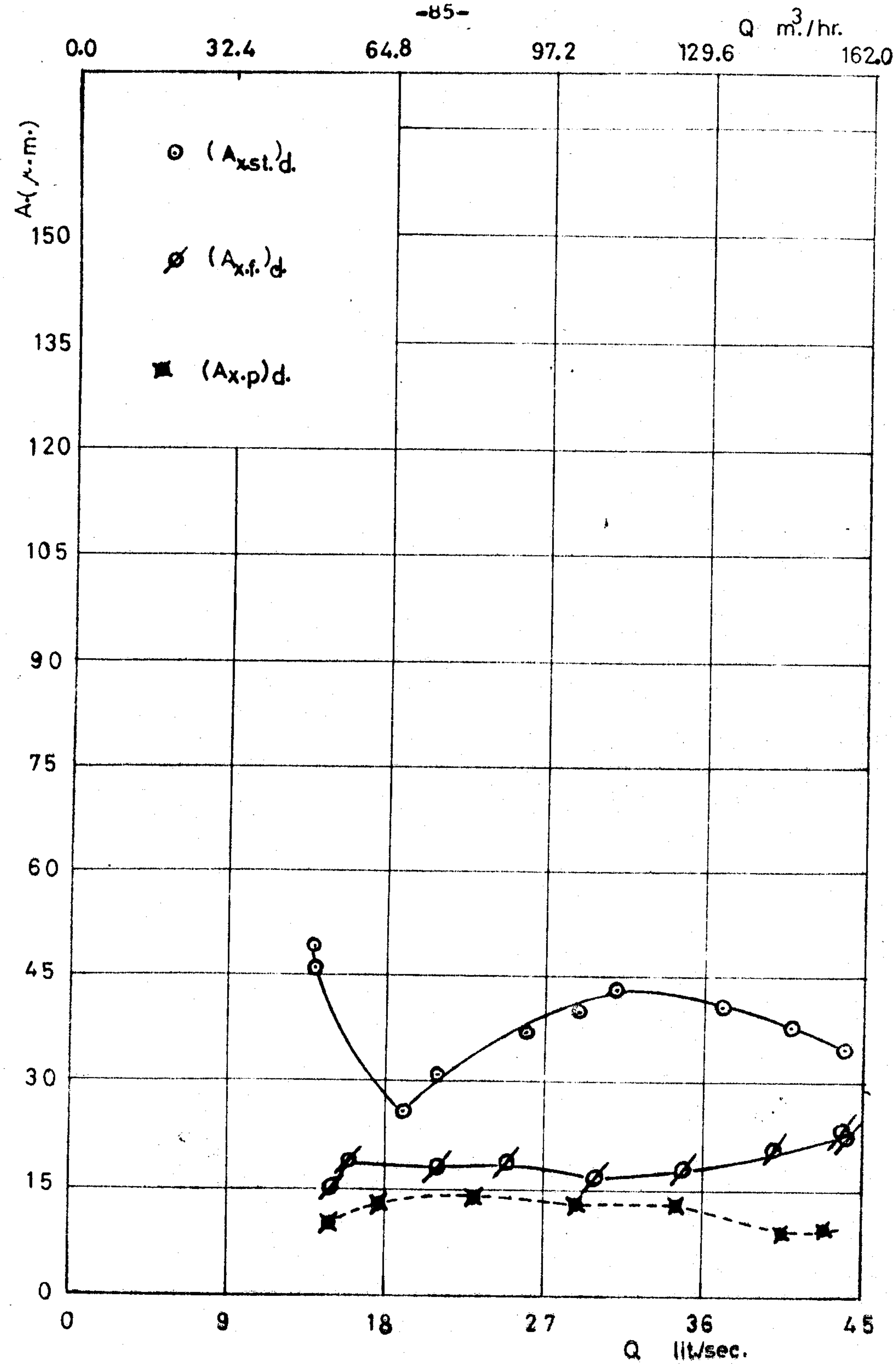

Q m^3/hr
0.0
32.4
64.8
97.2
129.6
162.0
A. (μm.)
(Ay.st)s.
(Ay.f.)s.
(Ay.p.)s.
150
135
120
105
90
75
60
45
30
15
0
0
9
18
27
36
45
Q. lit/sec.

Fig (4)



Q m³/hr.
0.0
32.4
64.8
97.2
129.6
162.0
A (μ.m.)
150
135
120
105
90
75
60
45
30
15
0
(Axst.)d.
(Ax.f.)d.
(Ax.p)d.
0
9
18
27
36
45
Q lit/sec.

Fig.(5)

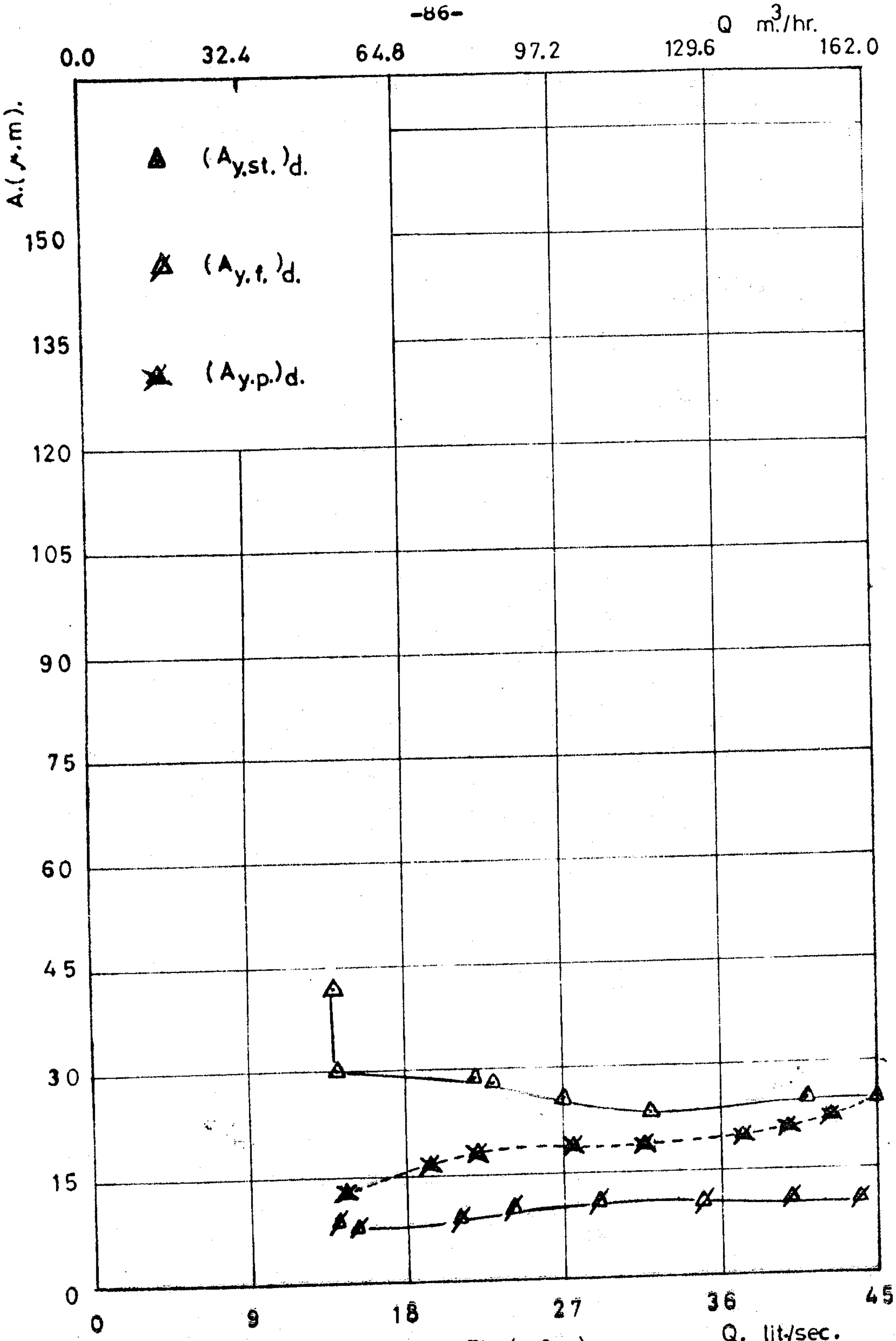
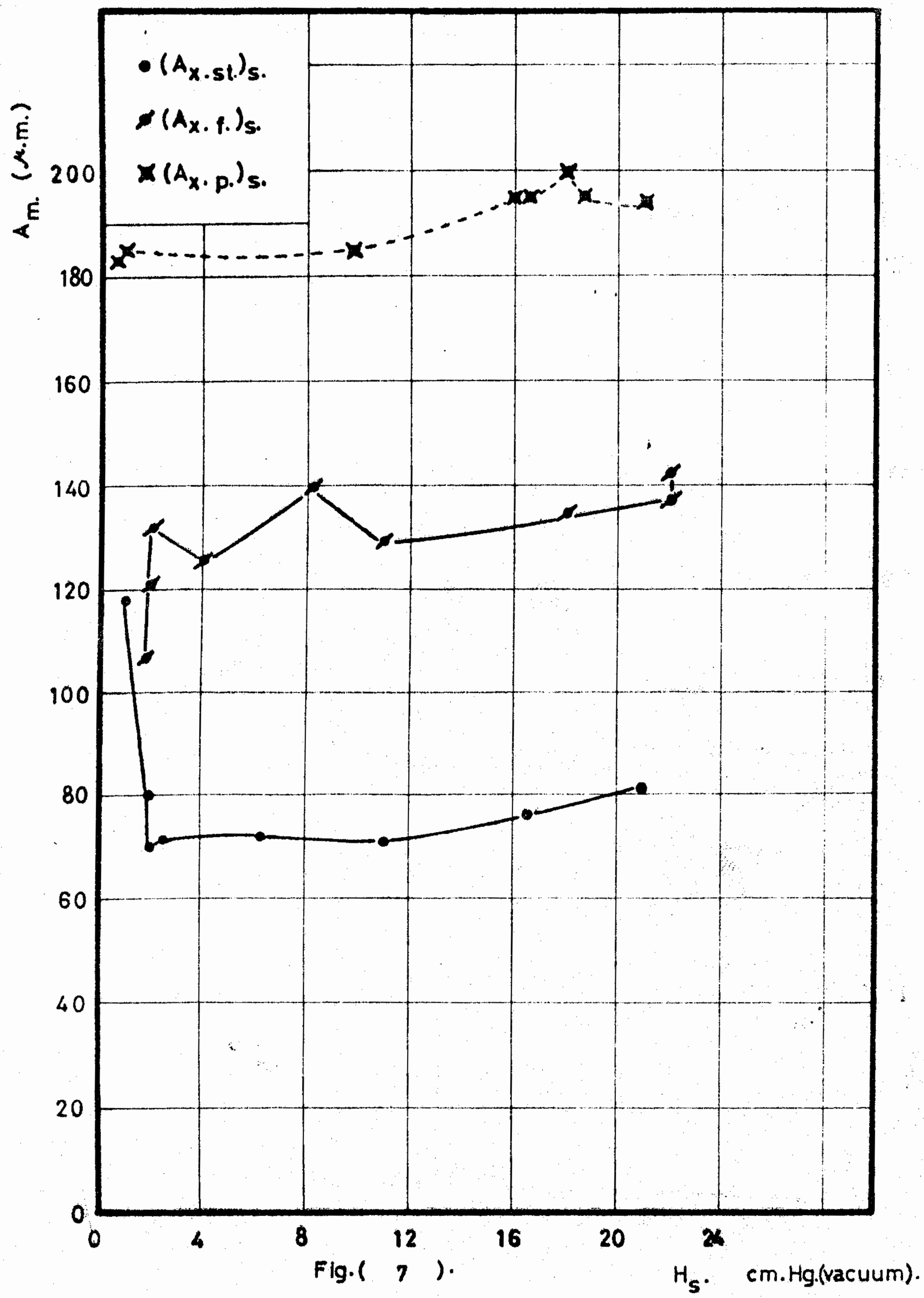

Q m.3/hr.
0.0
32.4
64.8
97.2
129.6
162.0
A.(μ.m).
(Ay.st.)d.
(Ay.f.)d.
(Ay.p.)d.
150
135
120
105
90
75
60
45
30
15
0
0
9
18
27
36
45
Q. lit/sec.

Fig.(6)



$(A_{x.st.})_s$.
$(A_{x.f.})_s$.
$(A_{x.p.})_s$.
A_m. (μ.m.)
200
180
160
140
120
100
80
60
40
20
0
0
4
8
12
16
20
24
H_s. cm.Hg.(vacuum).

Fig.(7).

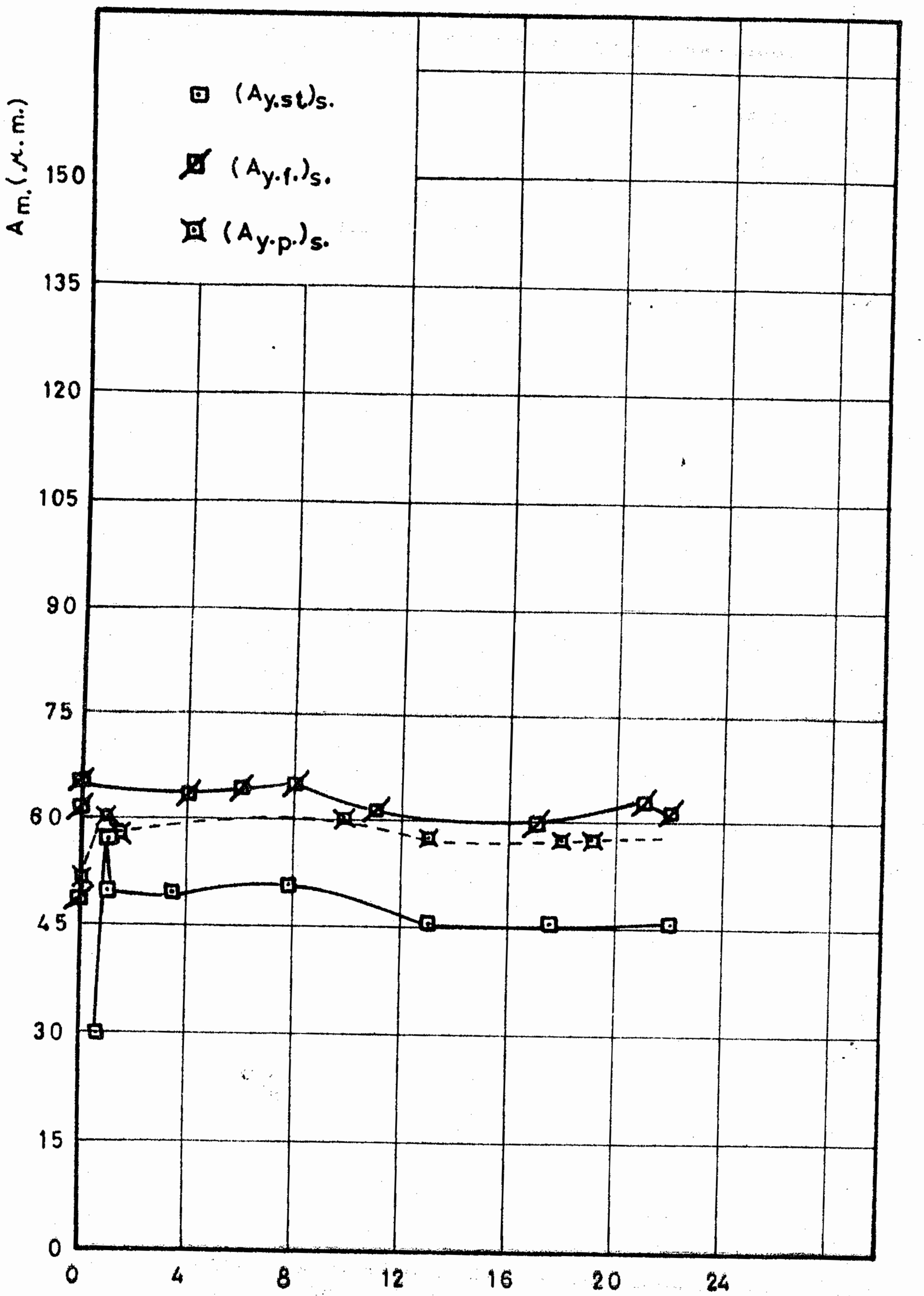
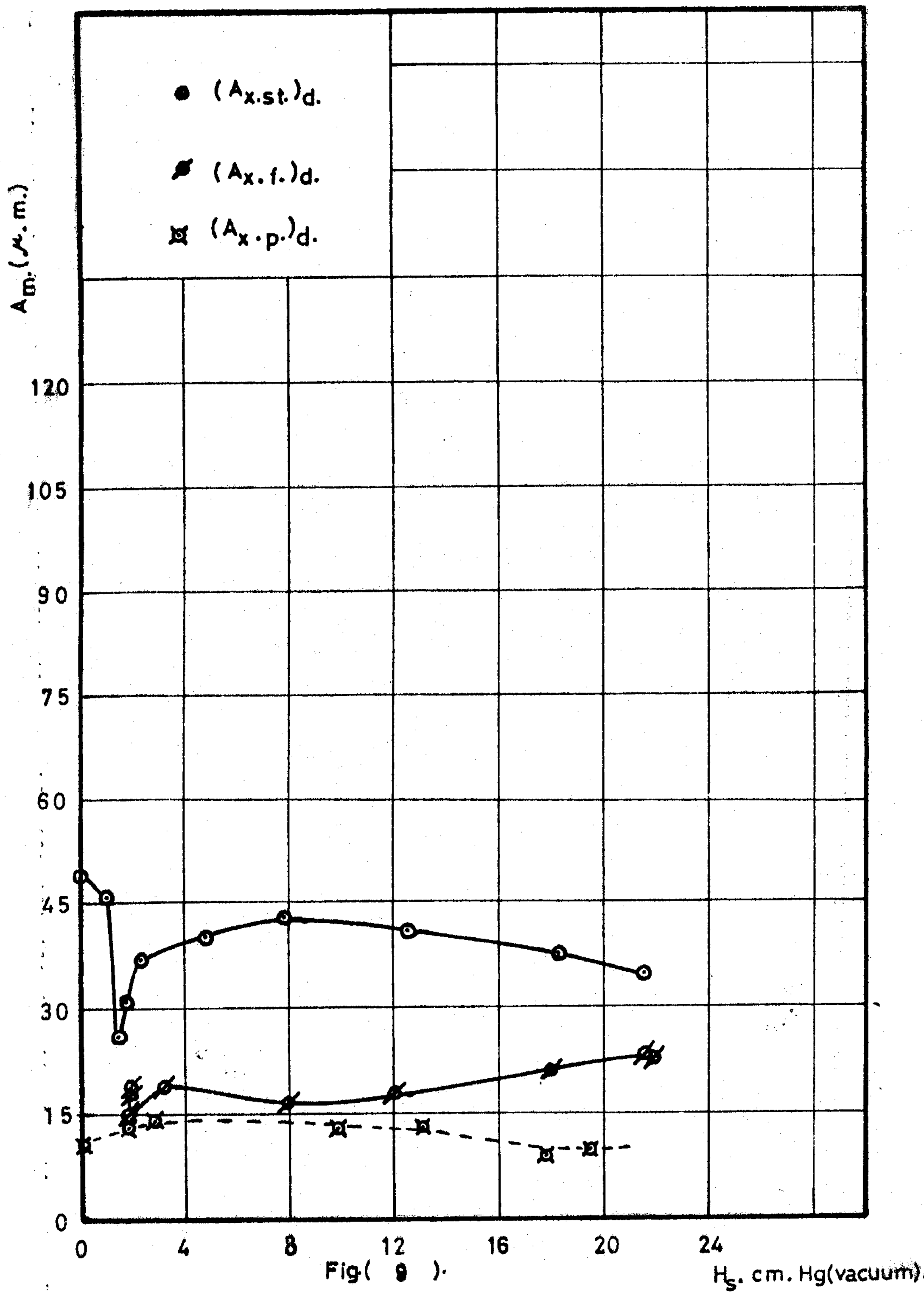

(Ay.st)s.
(Ay.f.)s.
(Ay.p.)s.
Am. (μ.m.)
150
135
120
105
90
75
60
45
30
15
0
0
4
8
12
16
20
24
Hs. cm.Hg(vacuum).

Fig.(8).


(Ax.st.)d.
(Ax.f.)d.
(Ax.p.)d.
Am.(μ.m.)
120
105
90
75
60
45
30
15
0
0
4
8
12
16
20
24
Hs. cm. Hg(vacuum).

Fig(9).

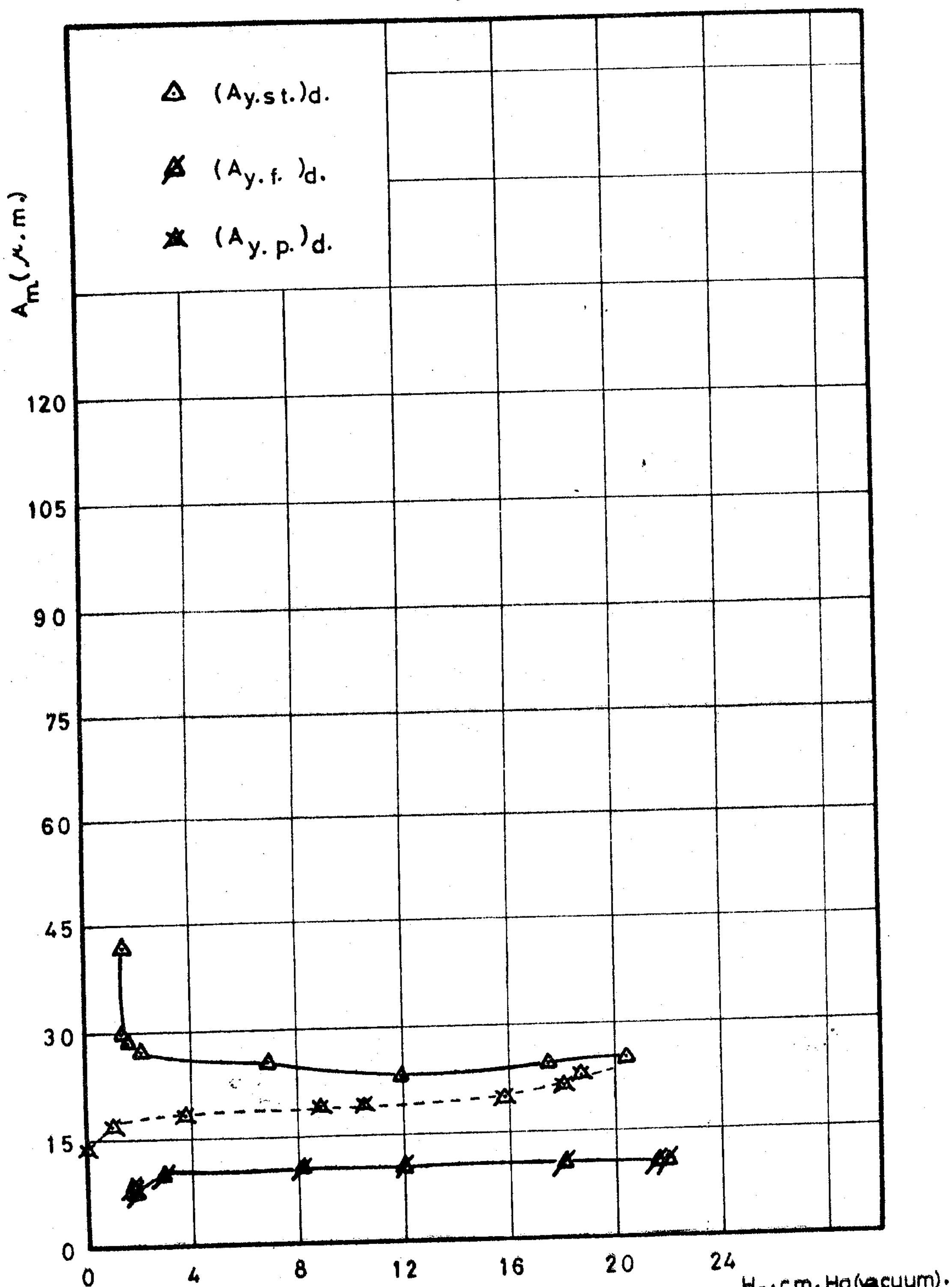

(Ay.st.)d.
(Ay.f.)d.
(Ay.p.)d.
Am(μ.m.)
120
105
90
75
60
45
30
15
0
0
4
8
12
16
20
24
Hs.cm.Hg(vacuum).

Fig.(10).

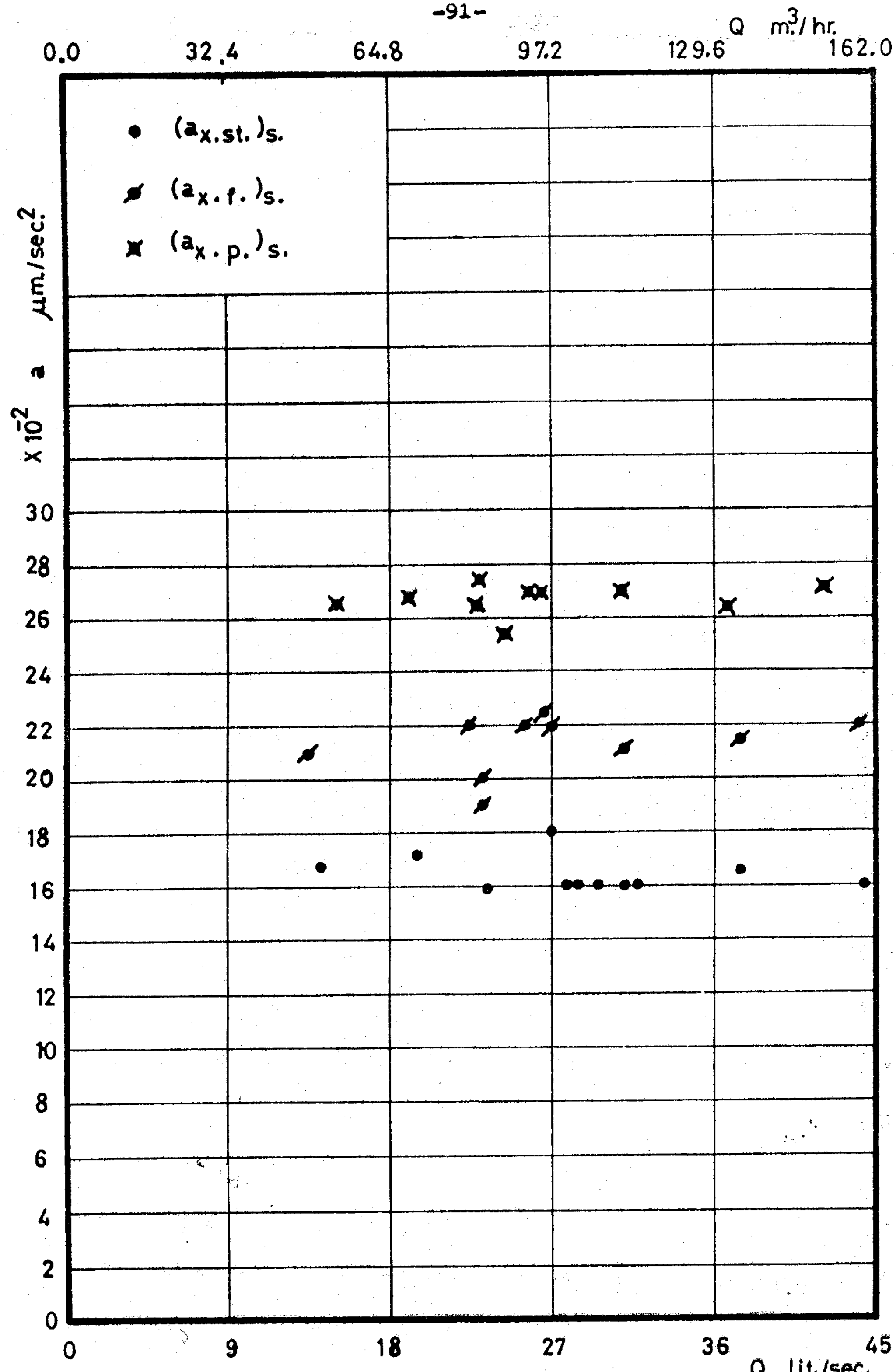

Q m³/hr.
0.0
32.4
64.8
97.2
129.6
162.0
(a_x.st.)s.
(a_x.f.)s.
(a_x.p.)s.
a X 10⁻² μm./sec.²
30
28
26
24
22
20
18
16
14
12
10
8
6
4
2
0
0
9
18
27
36
45
Q lit./sec.

Fig. (11)

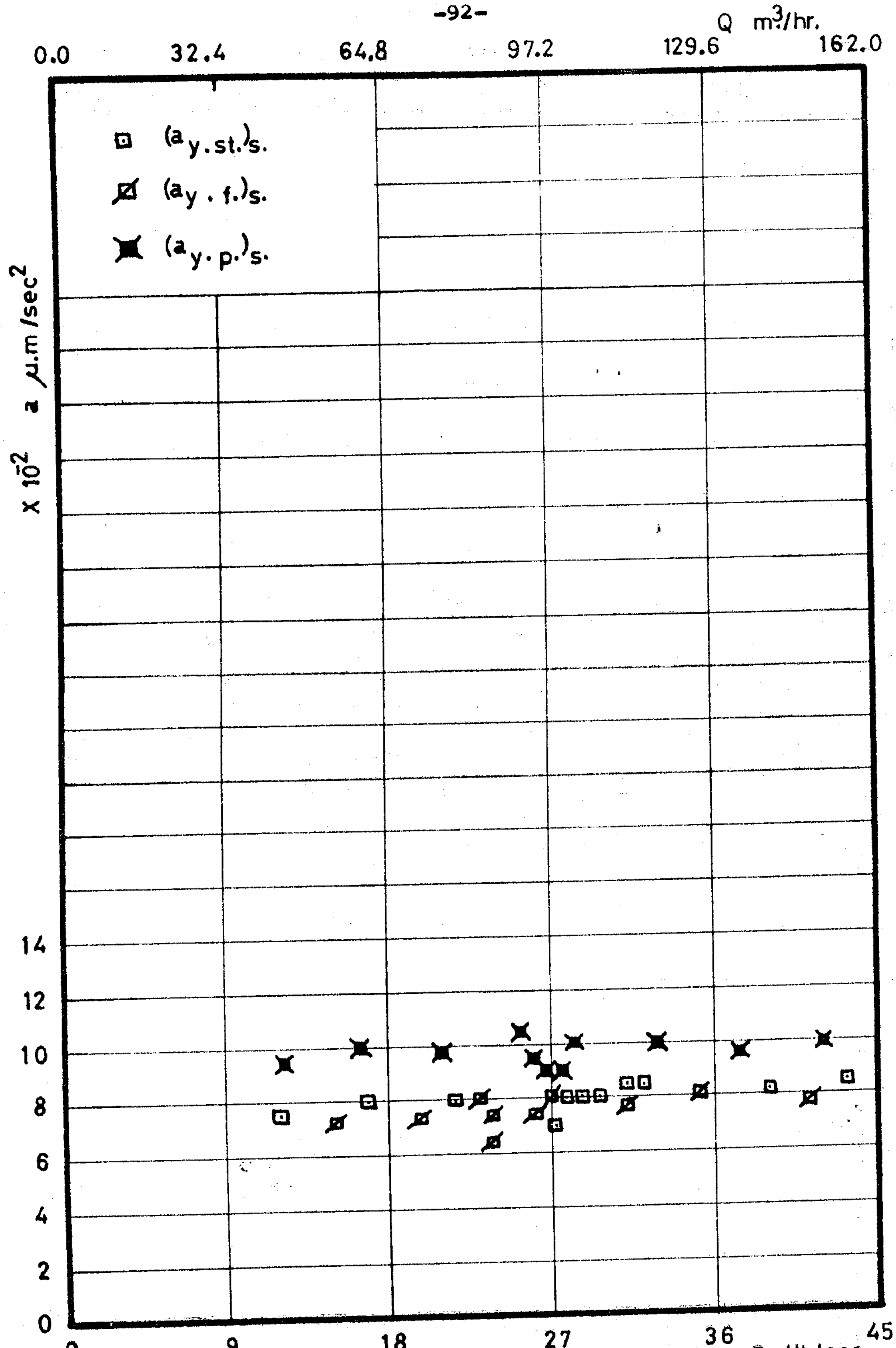

Q m³/hr.
0.0
32.4
64.8
97.2
129.6
162.0
(a y.st.)s.
(a y.f.)s.
(a y.p.)s.
X 10⁻² a μ.m/sec²
14
12
10
8
6
4
2
0
0
9
18
27
36
45
Q lit./sec.

Fig.(12)

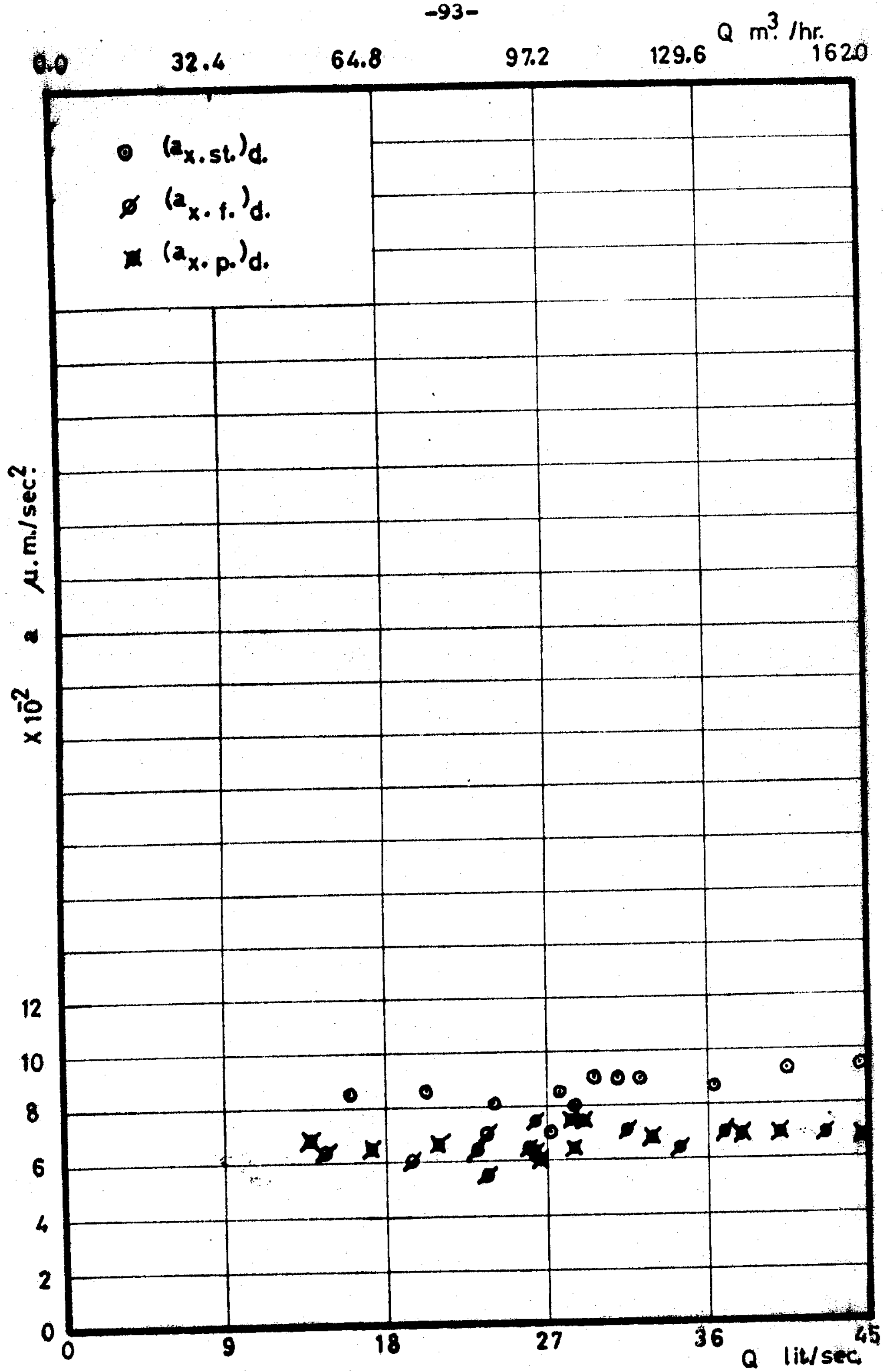

Q m³/hr.
0.0
32.4
64.8
97.2
129.6
162.0
(a x.st.)d.
(a x.f.)d.
(a x.p.)d.
X 10⁻² a μ.m./sec.²
0
2
4
6
8
10
12
0
9
18
27
36
45
Q lit/sec.

Fig. (13)

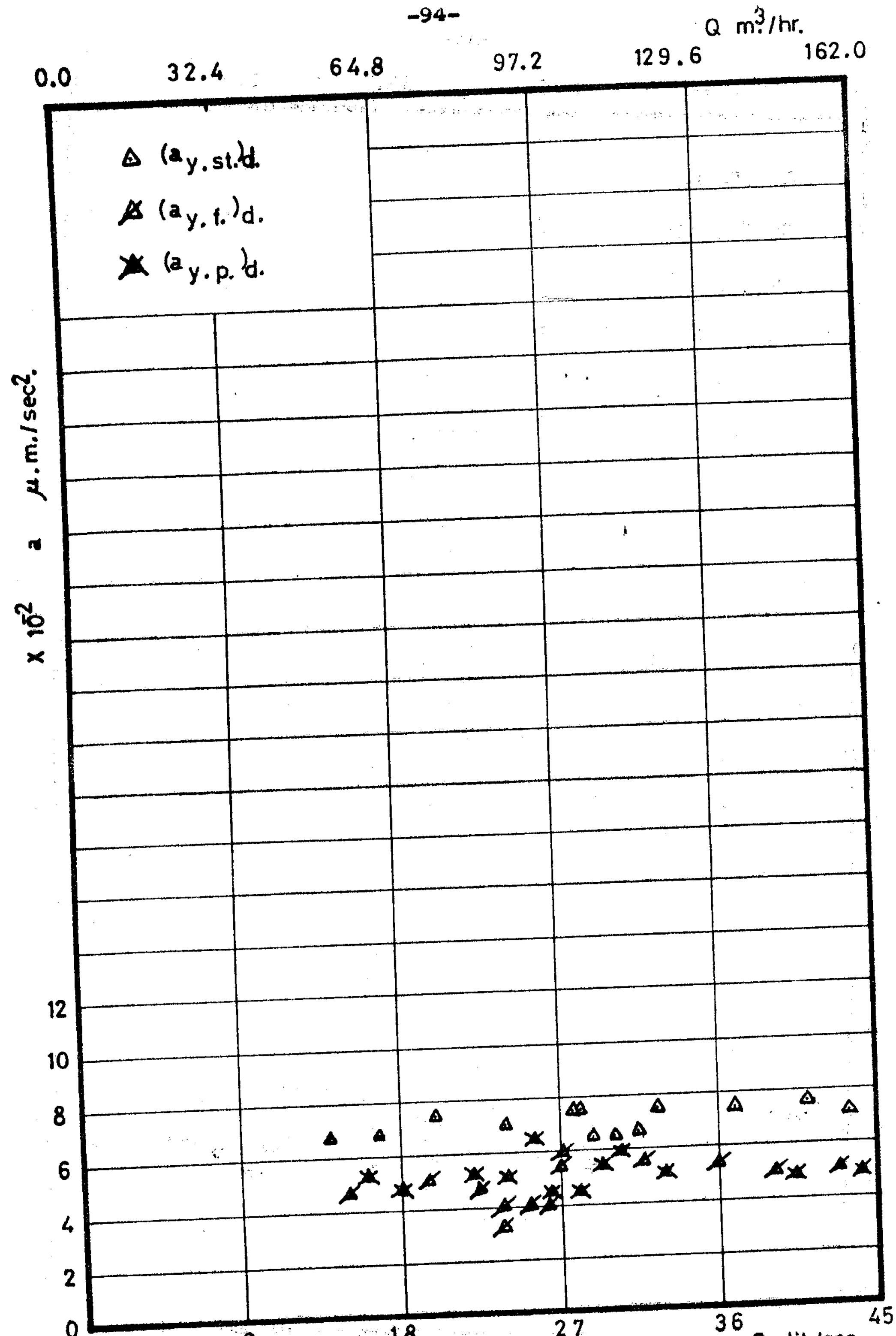

Q m³/hr.
0.0
32.4
64.8
97.2
129.6
162.0
(a y.st.)d.
(a y.f.)d.
(a y.p.)d.
x 10⁻² a μ.m./sec².
0
2
4
6
8
10
12
0
9
18
27
36
45
Q lit./sec.

Fig. (14)

VIBRATION PERFORMANCE OF LOW HEAD CENTRIFUGAL PUMPS USED FOR IRRIGATION

الاهتزازات فى مضخات الطرد المركزى ذات الضغط المنخفض المستخدمة فى الرى .

من المعروف ان الاهتزازات الميكانيكية للأجزاء المتحركة او الدوارة فى الماكينات لها اثار ضارة قد تصل الى حد انهيار اجزاء الماكينات .

ويهتم هذا البحث بدراسة اهتزازات مضخات الطرد المركزى واثر نوعيه تركيبه السحب على مستوى هذه الاهتزازات .

وقد اوضحت النتائج المعملية التى تم الحصول عليها انه بالرغم من الانتشار الواسع لاستخدام الخراطيم الكاوتشوك مع مضخات الطرد المركزى المستخدمة فى الرى لما لها من مرونة تتناسب مع طبيعة الاراضى ومصادر المياه الا انها ليست الوسيلة المثلى من وجهة نظر الاهتزازات الميكانيكية ، اذ يتسبب عدم انتظام سطحها الداخلى فى رفع مستوى الذبذبات عنها فى حالة استخدام المواسير الصلب . كذلك اوضحت النتائج زيادة كبيرة فى مستوى الاهتزازات عند استخدام المواسير البلاستيك بدلا من الخراطيم .

وقد اوضحت النتائج ايضا ان اهم مصدر للاهتزازات الميكانيكية فى جانب السحب هو صمام عدم الرجوع الشائع الاستعمال مما يوجد ضرورة البحث عن تصميم افضل بهدف خفض مستوى الاهتزازات فى مدخل المضخة .