# AN EXPERIMENTAL STUDY OF IMPROVEMENT OF A MICRO HYDRO TURBINE PERFORMANCE

by

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# IN THE NAME OF GOD

THE MOST COMPASSIONATE THE MOST MERCIFUL

## DEDICATION

**TO THE ONE WHOSE LOVE AND FEAR IS ALWAYS IN MY HEART,**
**THE ONE WHO MADE IT ALL POSSIBLE ,**
**TO THE SUPREME FRIEND,**
LORD GOD

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## ABSTRACT

The thesis includes a literature survey of small hydraulic turbines, incorporating a historical review. The possible role of "micro hydros" in generating power in various parts of the world, and particularly in Iran, is discussed.

The theory of turbo machinery, particularly with regard to axial flow turbines, is presented next. This is followed by some details on the design of guide vanes, runner blades and draft tube of axial flow turbines, these components being usually regarded as areas which have major impact on the performance of hydraulic turbines.

The next chapter gives the details of the test circuit that was constructed. This could provide water volume flow rates of up to 0.15 $m^3/s$ at heads of up 25m. The two dynamometers that were used could absorb powers of up to 25 kw and 50 kw respectively.

An existing micro-turbine, the Agnew turbine, was selected for examination and possible improvement. The first possible improvement was the introduction of guide vanes upstream of the turbine runner (this inclusion necessitated a second support for the main shaft). It was found that this gave significant improvements – efficiency raised by over 20% in some cases. The domain of high efficiency working was considerably extended.

It was observed that instability (with fall in power output) could occur after a period of running. This seemed to be associated with an accumulation of air bubbles at the highest point in the casing of the machine. Introduction of a vent from this point was found to relieve this problem and ensure stable operation.

Suggestions for further research are given.

## ORIGINAL WORK

The work described in this thesis is towards improvement of a micro hydro turbine originally designed by Mr. P.Agnew. It has been undertaken by a team of which the candidate is a member. He has taken part in all design and analysis work, as well as, much of commissioning, preparatory and experimental work. The author has been personally responsible for :

*1.* Design and construction of the laboratory for testing the turbine.

*2.* Redesigning and manufacturing of a ( 1:2 ) research model of Agnew micro turbine.

*3.* Design and commissioning of the guide blades mechanism.

*4.* Design, construction and operation of the measuring system including, velocity, pressure and torque measuring devices.

*5.* Operational and performance improvements of the turbine by establishment of essential modifications and mechanisms.

# NOMENCLATURE

| | |
|---|---|
| $A_J$ | Contracted area of the of the jet of water at the orifice plate. |
| $A_o$ | Area of the orifice. |
| C | Circulation |
| $C_c$ | coefficient of contraction |
| $C_D$ | Coefficient of drag. |
| $C_L$ | Coefficient of lift. |
| $C_V$ | Velocity coefficient. |
| D | Pipe internal diameter,Blade diameter. |
| $D_j$ | Water jet diameter at the orifice exit. |
| d | Hub diameter |
| e | Specific energy of water |
| $e_x$ | Distance between the center line of the orifice and center line of the upstream and downstream sides . |
| F | Cross sectional area of the draft tube&Force exerted by the dynamometer's arm . |
| $F_L$ | Lift force |
| $F_D$ | Drag force |
| g | Acceleration due to gravity |
| H | Effective head |
| $H_s$, $H_{vacuum}$ | Suction head |
| $H_{gauge}$ | Pressure head of the turbine |
| h | Head loss |
| hf | Head loss due to friction |
| $h_{dy}$ | Dynamic head |
| K | Velocity coefficient in Bernoulli's equation &Flow coefficient of the orifice |

| | |
|---|---|
| $L$ | Distance of the maximum deflection of the central curve from the leading edge, Length of the airfoil |
| $l$ | Depth of the airfoil, Length of the arm of the dynamometer |
| $m$ | Mass, Maximum deflection of the central curve of the airfoil |
| $N$ | Speed |
| $N_1$ | Unit speed ( Constant head ) |
| $N_s$ | Specific speed |
| $P$ | Power, Pressure |
| $P_a$ | Absolute pressure |
| $P_{atm}$ | Atmopheric pressure |
| $P_h$ | Horizontal component of force |
| $P_i$ | Input power |
| $P_{i1}$ | Unit input power (Constant head ) |
| $P_o$ | Output power |
| $P_{o1}$ | Unit output power ( Constant head ) |
| $P_v$ | Vertical component of force |
| $Q$ | Flow rate |
| $Q_1$ | Unit flow rate ( Constant head ) |
| $Re$ | Reynolds number |
| $r$ | Radius |
| $S$ | Supporting surface of the airfoil |
| $T$ | Spacing of blades, Torque |
| $T_z$ | Resultant external torque about an arbitrary fixed axis |
| $t$ | Maximum thickness of the airfoil |
| $U$ | Linear velocity |
| $U_b$ | Blade velocity |
| $U_{b-2}$ and $U_{b-3}$ | Velocities relative to blade |

| | |
|---|---|
| $U_f$ | Flow velocity |
| $U_i$ | Specific indicated velocity, Inlet velocity of the turbine |
| $U_o$ | Outlet velocity |
| $U_r$ | Radial velocity |
| $U_w$ | Tangential (Whirl ) velocity |
| $V$ | Velocity of flow |
| $W_u$ | Relative velocity |
| $Z$ | Height from datum, Number of blades |
| $\alpha$ | Angle of attack |
| $\alpha_0$ | Angle of attack for zero lift coefficient |
| $\alpha_\infty$ | Angle of attack for infinit span of the airfoil |
| $\beta$ | Lattice angle, Ratio of the orifice diameter to the pipe diameter |
| $\rho$ | Density |
| $\omega$ | Angular velocity |
| $\eta$ | Efficiency |
| $\eta_D$ | Draft tube efficiency |
| $\eta_h$ | Hydraulic efficiency |
| $\eta_m$ | Mechnical efficiency |
| $\eta_{max}$ | Maximum efficiency |
| $\eta_o, \eta_t$ | Over all efficiency |
| Indices 2, and 3 | Refer to inlet, and outlet of the turbine respectively. |
| Indices 2, and 5 | Refer to inlet, and outlet of the draft tube respectively. |

# CHAPTER

# ( I )

# MICRO HYDROS

## *PREFACE*

Basically , the term turbomachinery refers to a wide category of machines whose spectrum covers many different types of turbines , pumps ,fans , etc . Steam and gas power plants , jet engines , turbochargers , oil and water pump stations ,dams ... are examples of many number of applications of turbomachines. Water turbines may be regarded as the oldest members of this family.Thousands of years ago , old Iranians ( Persians ) and Romans made use of primitive and simple watermills (1).

The idea of using water as source of energy existed more than 2,200 years ago. The hydraulic energy was first produced in Asia (China and India ) in the form of mechanical energy , by passing water through a water wheel .The old type of water wheel made mainly from wood , still exists in India . Such types of prime movers were taken from the Asian Continent to Egypt and then from there to European countries and America . It is estimated that water wheel (undershot and overshot ) was used in Europe 600 years after its origin in India . The actual design of water wheel was first made by Leonardo da Vinci (1452 to 1519 AD ), the great Italian artist, which he did with hand sketches . The mathematical solutions of such a wheel were drawn by scientists Galileo Galilei and Descartes . A Swiss scientist Daniel Bernoulli (1700 to 1782) first wrote a theory for the conversion of water power in to other forms of energy in his book " Hydrodynamica " . Berniulli's Theorem was given practical appilcation by Segner in Goettingen (Germany ) in 1750, who designed a water wheel . Then Leonard Euler (1707 to 1783 ) from Basle (Switzerland ) wrote the theory of hydraulic machines in 1750 , which is used even to this day showing the fundamentals of the subject . Bernoulli and Euler discovered most of the mathematical work concerning the problems .

During nineteenth centuary , the rate of development of water turbines accelerated . Especially when the problem of generation of elctricity with the help of water turbines arose . Many engineers and scientists took part in this process , as the result of which present turbines such as Francis, Pelton and Kaplan were developed .

Now-a-days a wide variety of water turbines are in service for different flow rates and heads , which are scattered all over the world generating power from less than a kilo-watt to thousands of megawatts. How-ever , this work would cover a micro turbine for heads mainly less than 10 meters and flow rates of almost less than 500 (l/s) .

## Micro hydros and their importance

It has been around a few decades that attention has been attracted by small hydro potentials. After successful application of them , the idea of using mini-hydro and microhydro potentials was developed. Although micro hydros are not by any means comparable with large hydro plants , in so far as their power production capacity is concerned but , their dimensions , weights, costs , installations etc. are not also comparable .

The purpose of this argument is not to produce reasons for selection and comparison of either of the hydro plants , but is to justify application of micro hydros where applicable .

The following would highlight a number of reasons which can advocate the idea of micro hydros :

1. Relatively simple design ; light weight , and small size .
2. Simple mass production process .
3. Low production and over all costs due to items 1 and 2 .
4. Easy and simple installation , without requiring any heavy construction such as dams , massive generators , penstocks , weir etc. .
5. Cheap and easy maintenance.

6. Application of these turbines to agricultural activities, especially their installation on irrigation canals .It is important to notice that , the utilization of hydro power has strongly been advocated , in recent years in Iran .

7. Application of micro hydros at almost any site even with low heads and limited flow rates . Especially at rural areas and villages which some of them are situated many miles away from electricity network.

## Micro Hydros and the world

It has not been more than two decades that attention has been drawn towards small hydro potentials (1 to 100 MGW) . It was only after successful application of small hydro turbines, that thoughts were diverted towards minihydros (200 KW to 1 MGW ) and micro hydros (up to 200 KW ) potentials .

Republic of China is the leading country in applying microhydro turbines at her rural areas (2, 4 ) . Other countries such as U.K . , Russia , Japan , and France have also worked in this context . Never-the-less , the main users of micro hydro turbines who have also performed some work on them , are mostly the third world countries. Peru , New Guinea , Philippines , and Nepal are amongst those countries which are benefiting from micro hydros .

For the purpose of introduction to the work performed by different countries on micro hydros , Republic of China , Japan ,New Guinea ,and U.K. were selected . The following provides a summary of the work on micro hydros in these countries .

## Micro hydros in China

Due to existence of numerous sites in China suitable for micro hydro turbine installations , and due to the great need for power at many points of the country ,work on micro hydros began in early 1950 's.

At present there are more than 60000 micro hydro stations scattered all over the country . How.ever,it should be pointed that, the above figure only accounts for those sites which generate less than 50 KW. Information concerning models,flow rates and generator capacity of turbines employed in China are represented in table (1.1) . (..4) .From tables (1.1.A & 1.1.B ) , it may be observed that the main types of turbines used in China are as follows :

1. Fixed blade axial .
2. Mixed flow - Francis .
3. Diagonal impulse - Turgo
4. Impulse - Pelton

For future reference it may be noticed that ,45 ° inclined , adjustable blade, axial turbines are not amongst those employed in China .

## Japan and micro hydro

Facing the first and second energy crisis , development of domestic energy was advocated in Japan . Amongst all ideas and projects, development of small hydro turbines seemed to be more prosperous.

Fuji Electric Co . is one of the companies which has undertaken some work on this issue . As a matter of fact their work mainly concerns small hydros with generating power of less than 20 ( MW).They developed different turbine designs , some of which operate under heads as low as

| Turbine | | | Generator | | | | |
|---|---|---|---|---|---|---|---|
| Model | Head (m) | Flow ($m^3/s$) | Generator Capacity (kw) | Speed (rev / min) | | Voltage (v) | |
| | | | | 50 Hz | 60 Hz | 50 H | 60 Hz |
| ZD760 - LM - 60 | 2 ~ 6 | 0.3 ~ 1.64 | 18.30.40.55.75 | 1000 | 1200 | 400 | 460 |
| ZD760 - LM - 80 | 2 ~ 6 | 1.4~3.54 | 30.40.55.75.100.125 | 1000 | 1200 | 400 | 460 |
| | | | 160 | 500 | 514 | 400 | 460 |
| ZD760 - LM 100 | 5.5 | 4.174 | 160 | 428.6 | 450 | 400 | 460 |
| ZD760 - LM -120 | 2.6~ 6.6 | 4.17~6.58 | 75 100. 125 | 750 | 900 | 400 | 460 |
| ZD760 -LMY-120 | | | 160 200 | 300 | 300 | 400 | 460 |
| | | | | | (360) | (6300) | (6300) |
| | | | 250 320 | 375 | 360 | 6300 | 6300 |
| ZD760-LH - 200 | 4.2~5.7 | 16.4~187 | 500 600 | 137.5 | 189.5 | 3150 | 3150 |
| | | | 800 | 214.3 | 211.7 | 6300 | 6300 |
| ZD560-LMY- 60 | 9.3 ~12.7 | 1.21~1.35 | 75 | 1000 | 900 | 400 | 460 |
| | | | 100. 125 | 750 | 900 | 400 | 460 |
| ZD560- LH - 60 | 7.8~12.7 | 1.41~1.85 | 18.30 40.55.75 | 750 | 720 | 400 | 460 |
| | | | | | (900) | | |
| ZD560-LMY- 80 | 6.3~13.5 | 1.77~2.97 | 75 | 1000 | 1200 | 400 | 460 |
| | | | 200.250 | 600 | 600 | 400 | 460 |
| | | | 160.200 | 500 | 600 | 400 | 460 |
| ZD560 - LH - 80 | 8 ~ 14 | 2.9~3.99 | 160 | 428.6 | 450 | 400 | 460 |
| | | | 250 | 500 | 514 | 400 | 460 |
| | | | | | (600) | | |
| | | | 400 | 600 | 600 | 6300 | 6300 |
| ZD560 -LH 100 | 12~15 | 5.53~5.5 | 500.630 | 500 | 514 | 6300 | 6300 |
| ZD560- LH- 120 | 8~14 | 5.1~8.5 | 500.630 | 375 | 360 | 6300 | 6300 |
| | | | | | (450) | | |
| ZD560 - LH -180 | 10 | 16.5 | 1250 | 250 | 257 | 6300 | 6300 |
| HL110 - WJ-33 | 18.5~50 | 027~0.33 | 75.100.125 | 1000 | 1200 | 400 | 460 |
| HL110 - WJ-42 | 49 | 044 | 160 | 1000 | 1200 | 400 | 460 |
| HL110 - WJ-50 | 33.3~92 | 0.51~091 | 125.160.200.250 | 750 | 720 | 400 | 460 |
| | | | | | (900) | | |
| | | | 320.500.630 | 1000 | 900 | 400 | 460 |
| | | | | | (1200) | (6300) | (6300) |
| HL110 -WJ - 60 | 55.5~80 | 0.97~110 | 400.500 | 750 | 720 | 6300 | 6300 |
| | | | | | (900) | | |
| | | | 630 | 1000 | 900 | 6300 | 6300 |

Table 1.1. A

| *Turbine* | | | *Generator* | | | | |
|---|---|---|---|---|---|---|---|
| *Model* | *Head (m)* | *Flow ($m^3/s$)* | *Generator Capacity (kW)* | *Speed (rev/min)* 50Hz | *Speed (rev/min)* 60Hz | *Voltage (V)* 50Hz | *Voltage (V)* 60Hz |
| XJ-W-42/1×10 | 50~160 | 0.24~0.43 | 75.100.125 | 750 | 720 (900) | 400 | 460 |
| | | | 160.200.250.320.400 | 1000 | 900 (1200) | 400 | 460 |
| XJ-W-42/1×11 | 50~100 | 0.29~0.39 | 100.125.160 | 750 | 720 (900) | 400 | 460 |
| | | | 200.250 | 1000 | 900 | 400 | 460 |
| $XJ_{02}$-W-25/1×7 | 40~90 | 0.07~0.14 | 26.40.55 75 | 1000 1500 | 1200 1200 | 400 400 | 460 460 |
| CJ-W-92/1×11 | 150 | 0.416 | 500 | 500 | 514 | 6300 | 6300 |

The turbine model type consists of three parts: turbine type and specific speed; shaft arrangement; and form of inlet — runner diameter(mm) or number of jets × jet dia.(mm) for impulse turbines. Here, for *turbine type*: ZD — fixed-blade axial; HL — mixed-flow — Francis; XJ — diagonal impulse (Turgo); CJ — impulse (Pelton). *Shaft arrangement*: L — vertical; W — horizontal. *Form of inlet*: J — metallic spiral case; H — reinforced spiral case; M — open flume; MY — pressurized open flume; CJ — Pelton turbine.

Table 1.1 B

7.5 (m), and flowrates as small as 150 (l/s), ( cross - flow turbines ). In their work they have made an attempt to standardize their turbines , so thet , for various conditions of water systems, different designs and different sizes of turbines would be available . Chart (1.1) shows Fuji 's chart for their standard hydraulic turbines.

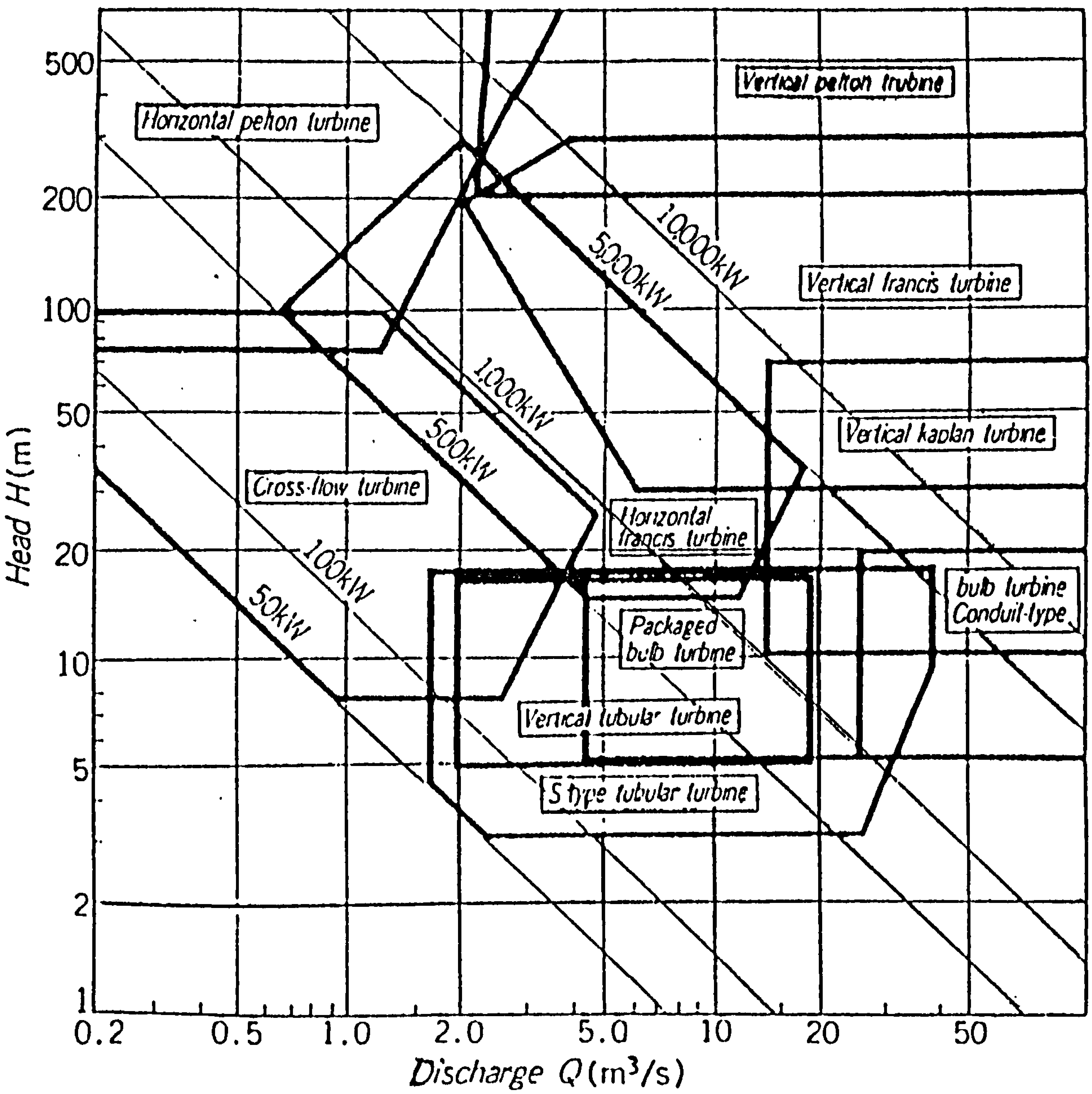



Chart (1.1) selection chart of Fuji standard hydraulic turbine

Below, a list of turbines, designed and manufactured by Fuji is presented :

1 . Cross - flow turbine .
2 . Packaged Bulb turbine .
3 . S- type tubular turbine .
4 . Horizontal Pelton turbine .
5 . Horizontal Francis turbine .
6 . Vertical tubular turbine .
7 . Small vertical Francis turbine .

## Micro hydro in New Guinea

New Guinea is one of the countries which has benefited application of micro hydros in her rural areas (12 ) . Apart from being an end user of these machines , apparently there has been some research work performed on the subject . For instance, R . J . Hothersall, a lecturer at the department of mechanical engineering, university of technology, in New Guinea has performed some work on establishing criteria for selection of microhydro turbines . In an article he has presented a methodology, by which for a given small hydropower potential, suitable turbine or pump can be selected (13) . In this work only turbines with power outputs less than 100 (KW) are considered . Also design features specific to micro hydro turbines are presented . Since, for certain applications , a pump , operating in reverse, as a turbine provides an acceptable option, then data for pump selection are also included .

Table (1.2) , from this article, provides a comparison between impulse and reaction turbines for microhydro schemes. Also figure (1.1) represents a selection chart for water turbines under 100 (Kw), according to the criteria established in above mentioned article .

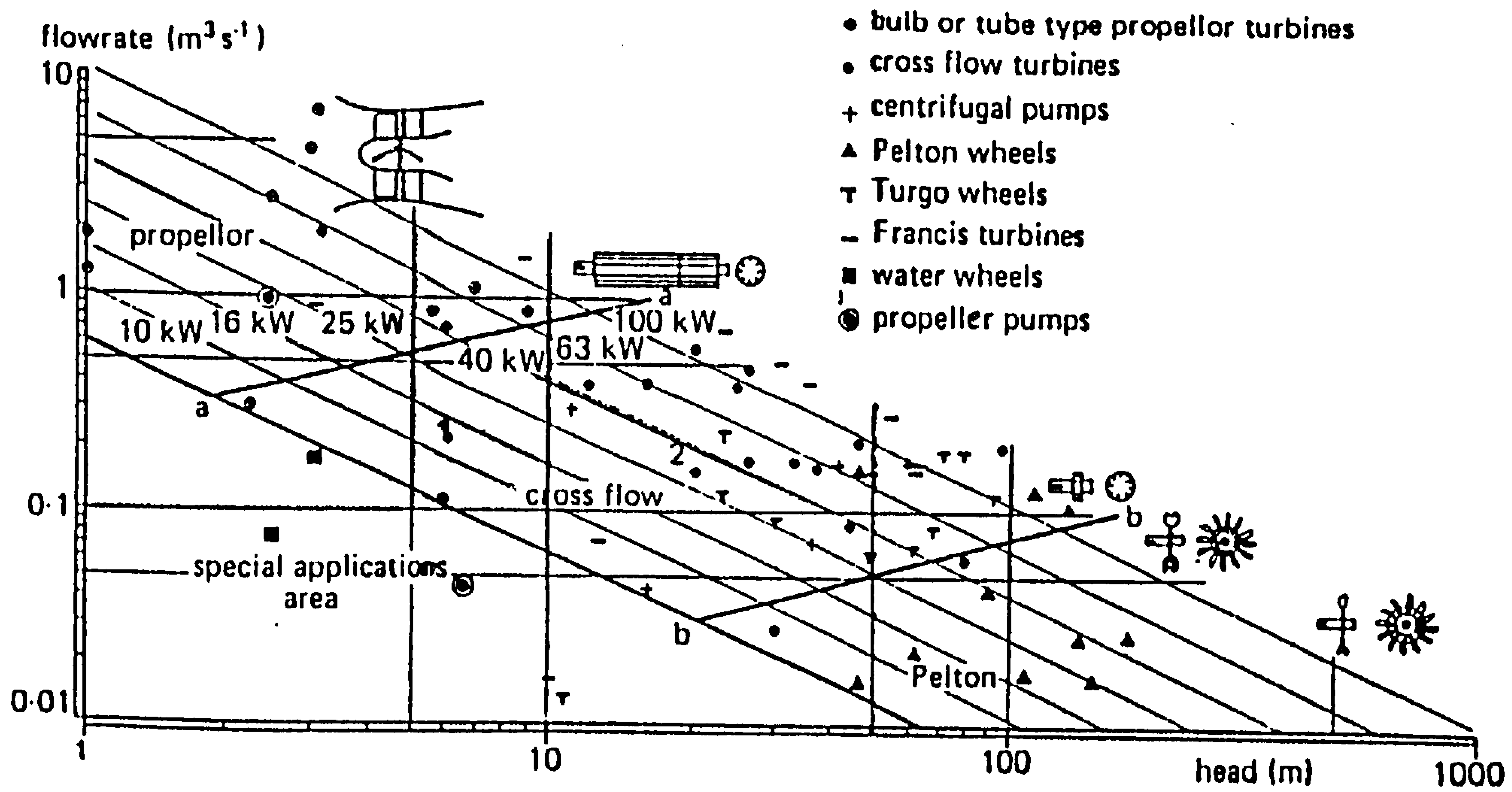



Figure ( 1.1 ) -Selection chart for water turbines under 100 Kw.

## Micro hydros in the U. K .

Altogether, 11 turbine manufacturers were contacted in the U.K. Nine of these companies had either stopped working on micro hydros or were not interested in them any more . Only two of these

Advantages and disadvantages of impulse and reaction turbines

| Turbine characteristics | Suitability for small hydro schemes (< 100 kW) | |
|---|---|---|
| | Advantages | Disadvantages |
| Impulse<br>*In general:* | Reduced likelihood of cavitation.<br>Good part-load efficiencies.<br>Possibility of static mechanical governing. | Low specific speeds limit applications. Lower peak efficiences over reaction turbine. Turbines must operate generally, above tailrace level. |
| Pelton | Erosion damage easily repaired. No axial bearing loads. | Multiple jets required to cope with higher flowrates. |
| Cross-flow | Specific speeds in the range 40-200 SI obtainable by varying turbine width only. | Efficiencies of between 70-80 per cent are slightly lower than Pelton/Turgo/Francis designs. |
| Turgo | Application range similar to multi-jet Pelton wheel designs. | Axial loading on bearings high. |
| Girard | | Poor part-load efficiencies. Outdated design. |
| Water wheels | Simple technology. | High torques/low operating speeds. |
| Reaction<br>*In general:* | High speficic speeds allow for compact design.<br>High peak efficiences.<br>Use of draft tube enables more effective site utilization. | Require elaborate seal design.<br>Maintenance more complex and expensive.<br>Application and operation limited by cavitation. |
| Francis | Standard sizes available from several manufacturers. | Relatively poor part-load efficiencies. Limited, economically, to power outputs of above 60 kW. |
| Kaplan/bulb/tube | Bulb and tube types offer savings when applied to existing dam installations. | High runaway speeds. |
| Centrifugal pump<br>Axial flow pump | Low cost. Stock items. | Poor part-load efficiencies.<br>Lack of application data. |
| Straflo | Ultra-compact design suitable for low-head schemes. | Not commercially available under 100 kW. |

Table (1.2)

manufacturers were still interested and were actually interviewed for possible future work .

One of these companies was " water power engineering turbine " owned by Mr. Osman . M . Goring , a mechanical engineer who had been working on micro hydros for many years . There , they had developed a modular unit comprising a turbine tank and an inlet tank . The prime objective of this design was to meet the requirements of sites with low heads (less than 10 meters) and low flow rates ( less than 2 ($m^3/s$).The turbine itself was a continuous back to back reaction

machine , having an efficiency of 70 % under ideal operating conditions .

The second company contacted , was owned by Mr. P.W. Agnew , a retired lecturer at the Department of Mechanical Engineering , Glasgow university . Mr Agnew had embarked upon designing and manufacturing a micro turbine in mid 1970's . His turbine at the beginning was a fixed blade , propeller axial flow turbine which later was developed to an adjustable blade 45° inclined axial flow turbine.

Discussions with Mr.Agnew showed his enthusiasm for further developments on his turbine . He believed that there had not been enough work performed on development of his turbine , ". . . for lack of money it was necessary to go straight from the drawing board to site testing stage , without any laboratory testing and development work " . (7) . Therefore by studying , performance results of the turbine and considering suitability of its working range it was then decided to use the Agnew turbine as the subject for our future work . Details of the turbine are presented in chapter 4 .

## Iran , Hydropower and microhydros

In order to find out about what had been done towards microhydros in Iran , many ministries , companies , research institutes and universities were contacted . The following, lists out a number of most important ones : -

1 . Ministry of Power
   I. Water resources section .
   II . Rivers and dams section .
   III . Water potentials of western states Department.

2 . Ministry of Jahad .
   I . Water resources section .
   II . Construction deputy, power and energy department .

3 . Power institute , affiliated to the ministry of power . Actually designed and manufactured a Pelton wheel .

4 . Mahab Ghods co . Involved in over-all design of hydro power plants . (Dr . Bakhtari ) .

5 . Ghods Niroo co.

6 . Neer pars co.

7 . Pump Iran factory , which basically is a pump manufacturing company. They ,in a joint venture with Tehran University ,embarked upon production of 35 different sizes of reversed pumps (pump turbines). According to the data released by them , their pump turbines are capable of generating 5 to 250 (KW) , heads over 10 meters and flow rates of 100 to 2000 $m^3/h$ . Figure (1.2), shows a graph of their flow rates against head for different sizes of pump turbines .

8 . Water power company , affiliated to the Ministry of Power . A floating turbine was actually designed, manufactured and successfully tested by them .Needless to mention , that the efficiency of the turbine was not higher than 50% .

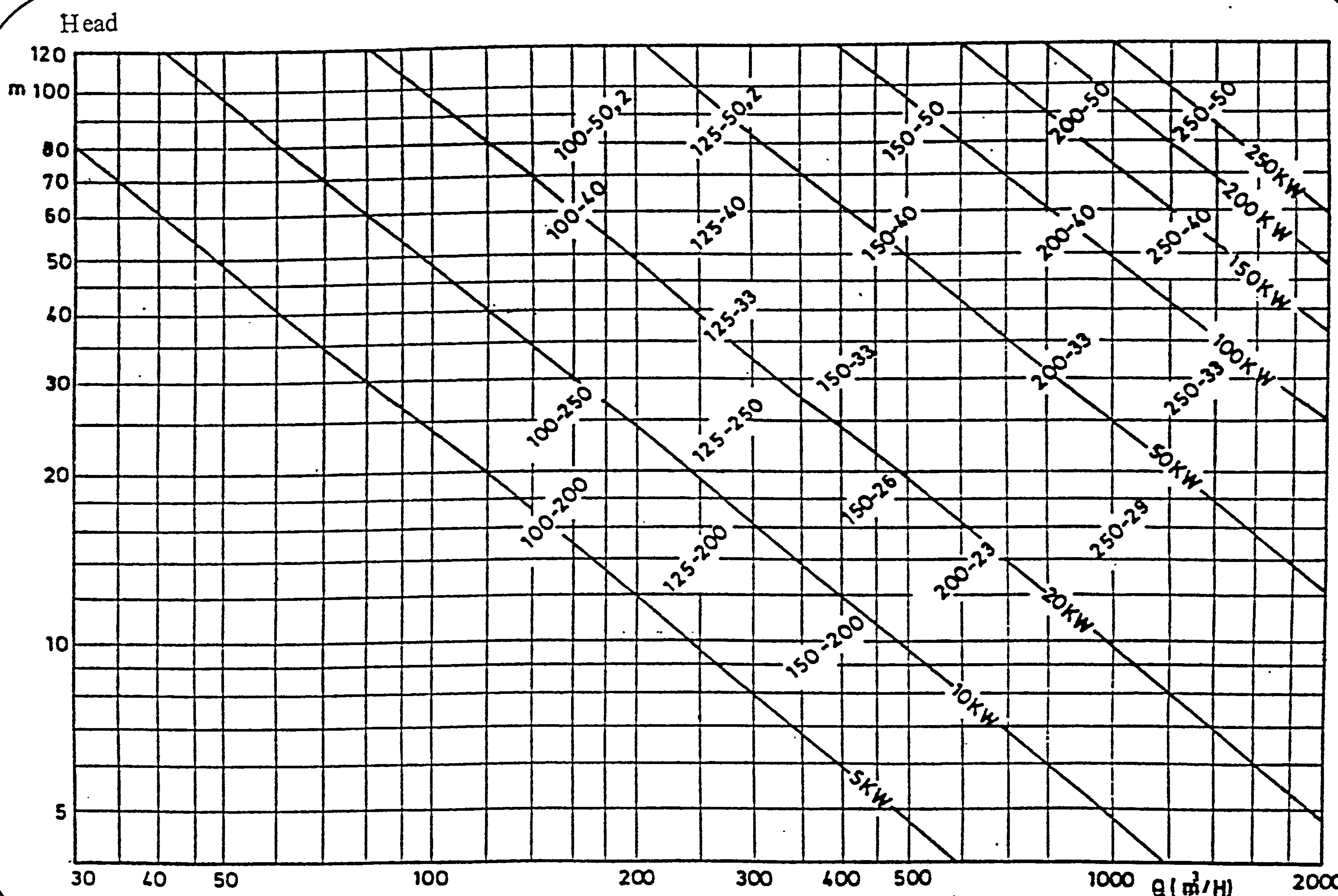

Head
m
120
100
80
70
60
50
40
30
20
10
5
30
40
50
100
200
300
400
500
1000
2000
Q (m³/H)
100-50,2
125-50,2
150-50
200-50
250-50
100-40
125-40
150-40
200-40
250-40
125-33
150-33
200-33
250-33
100-250
125-250
150-26
250-29
100-200
125-200
200-23
150-200
250KW
200KW
150KW
100KW
50KW
20KW
10KW
5KW

Figure 1.2

## I. Resources and Potentials

Iran is a vast country , having an area of 1 , 648 , 000 $Km^2$ . As far as temperature is concerned almost all four seasons may be found at any time of year at different parts of the country . A larger proportion of its 60 million population live in , western and northern parts . Table( 1.3) , divides the country into two major eastern and western sections and provides data on their population , rain fall , surface and underground waters and the area of each section . ( 14 )

| Western Area | Eastern Area | |
|---|---|---|
| 70 | 30 | Population % |
| 60 | 40 | Rain fall % |
| 60 | 40 | Surface waters % |
| 57 | 43 | Underground % Water resources |
| 30 | 70 | Area % |

Table ( 1.3 )

Figure ( 1.3 ) , shows the areas mentioned above on the map of the country . The shaded area represents the eastern area . The annual amount of rain in Iran is around $240 \times 10^9$ $m^3$ of which , $100 \times 10^9$ $m^3$ is evaporated , $60 \times 10^9$ $m^3$ is absorbed by the ground and the remaining $80 \times 10^9$ $m^3$ is actually in either seasonal or permanent rivers . (14)

A look at the geography of Iran shows that , Alborz mountains and Zagross mountains , cover , mainly the north , north west and western parts of the country , exactly those parts which receive more of the rainfall ( western area , on figure 1.3 ).Therefore , most of the permanent rivers , and water falls are situated in the western area .

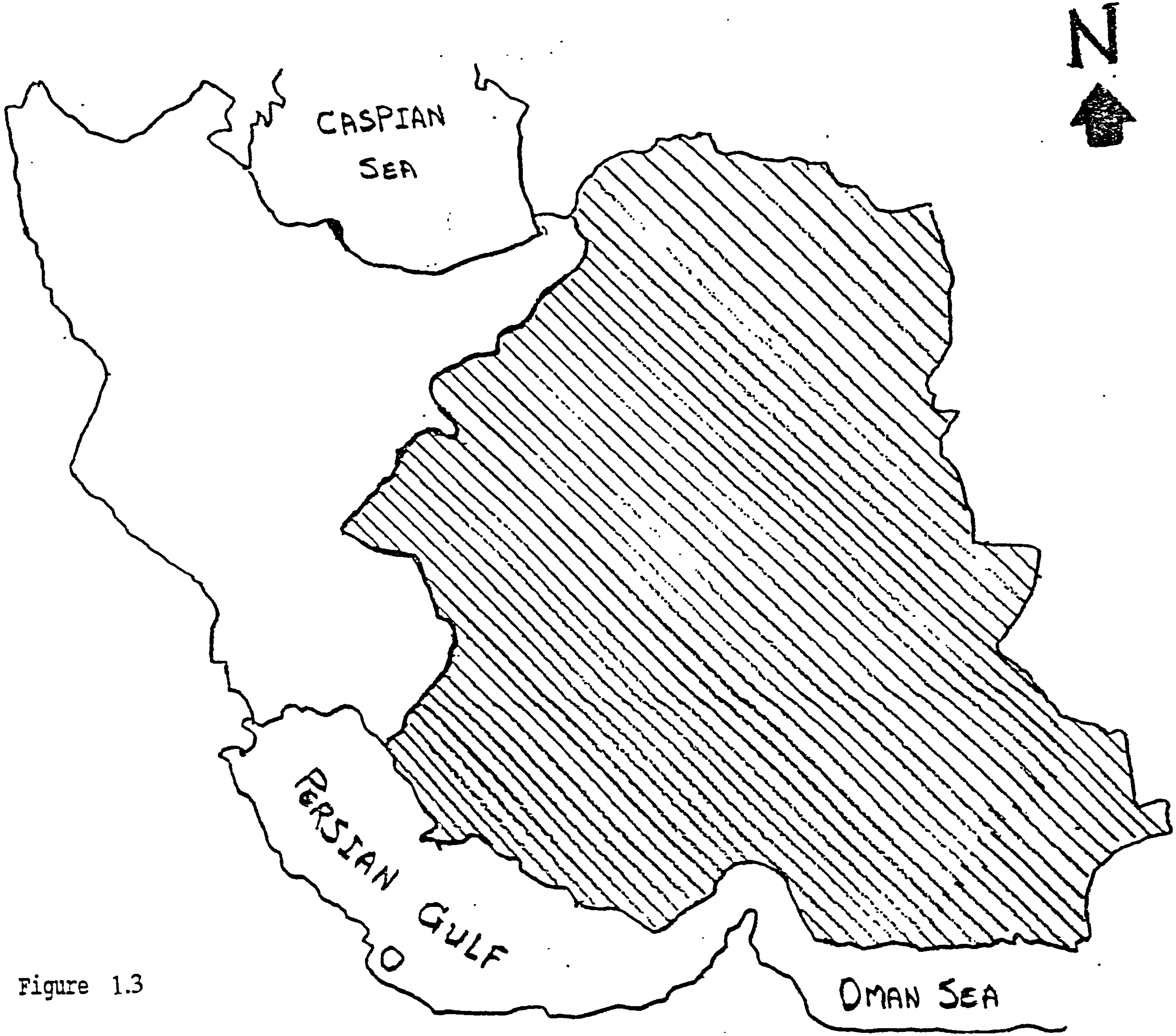



Figure 1.3

Studies led by the ministry of power, show that, the hydro power potential in Iran is around 17000 (MGW), of which only about 2000 (MGW) are being benefited and the rest remain to be used in future. This figure does not include micro hydro potentials. Early studies by the same ministry show that, over 3000 micro hydro potentials exist in rural areas (2).

Also, there have been some studies led by the ministry of Jahad as an attempt to pin point hydro potential sites in the range of, less than 200 KW up to 20000 KW, across the country. They have divided this range into three sections as shown in table (1.4) according to the site's nominal power generating potential. (3)

| Micro hydro sites | Mini hydro sites | Small hydro sites |
|---|---|---|
| Up to 200 KW | From 200 KW to 2000 KW | From 2000 KW to 20000 KW |

Table (1.4)

According to their Study, a total number of 262 microhydro potentials exist in 13 counties, through out the country.

Although, in their study they have not taken total heads less than 10 metres into account .This would justify the discrepancy between their figure and figures released by the ministry of power. Table (1.5) represents an overall view on the potentials selected by the ministry of Jahad.

## II. Needs & Problems

According to a report issued by the ministry of Jahad, there are over 64000 villages scattered all over the country, Less than 30.000 of these villages, at present are enjoying the electric power, whereas, the rest of them (more than 34000) are simply suffering a lack of this

Table (1.5)

| No. of potentials | Total power Small (KW) | No. of potentials | Total power Mini (KW) | No. of potentials | Total power Micro (KW) | Total No. of potentials | Total power (KW) | Name of the county | |
|---|---|---|---|---|---|---|---|---|---|
| 3 | 8732 | 21 | 13679 | 10 | 1303 | 34 | 23714 | West Azerbaijan | 1 |
| — | — | 3 | 898 | — | — | 3 | 898 | East Azerbaijan | 2 |
| 1 | 20000 | 10 | 6599 | 5 | 721 | 16 | 27320 | Esfahan | 3 |
| 1 | 3500 | 1 | 1700 | — | — | 2 | 5200 | Boushehr | 4 |
| — | — | 11 | 11254 | 3 | 436 | 14 | 11690 | Bakhtaran | 5 |
| 10 | 81300 | 35 | 31436 | 2 | 268 | 47 | 113004 | Chahar Mahal | 6 |
| — | — | 1 | 235 | 4 | 449 | 5 | 684 | Khorasan | 7 |
| — | — | 10 | 9947 | 16 | 654 | 26 | 10601 | Zanjan | 8 |
| 5 | 17940 | 20 | 22691 | — | — | 25 | 40631 | Kohkilouyeh | 9 |
| — | — | 15 | 8252 | 28 | 3529 | 43 | 11781 | Gilan | 10 |
| 2 | 8380 | 26 | 19002 | 6 | 785 | 34 | 28167 | Lorestan | 11 |
| 1 | 2388 | 32 | 22102 | 21 | 2429 | 54 | 26919 | Mazandaran | 12 |
| — | — | — | — | 3 | 503 | 3 | 503 | Hamadan | 13 |
| 23 | 142240 | 185 | 147795 | 98 | 11077 | 298 | 301112 | Tptal | |

valuable , basic commodity . Along with electricity , lack of clinical services, telecommunications, etc.. , have caused a mass exodus , towards cities. In one hand , cities do not have enough facilities to cope with this rapid population growth . On the other hand , agricultural activities would suffer heavy damages due to lack of manpower . This is due to the fact that , in Iran , agricultural activities are mainly concentrated at villages , with people leaving villages there will be no one to work on the farms .

The prime objective of the above short discussion is not , to hold "lack of electricity" responsible for all the hardship and shortcomings but its important role in man's welfare and culture can not by any means be denied .

## SELECTION OF HEAD & FLOW RATE RANGE

Before moving any further it would be more appropriate to explain what ranges were selected for the turbine's head and flow rate , and how they were selected .

Referring to early sections of this chapter , it was mentioned that , the ministry of power has spotted more than 3000 sites for micro hydros having heads less than 10 meters , On the other hand , ministry of Jahad 's report indicates presence of 298 micro hydro sites for total heads of over 10 meters . A close look at this report would indicate that more than 130 of these sites have flow rates ranging between 50 l/s to 500 l/s . This figure is about 45% of the total number of sites , which would mean that selecting a flow rate less than 500 l/s can cover a good percentage of the total potential sites . However , if generalization of these figures is accepted then it may be concluded that more than 1300 sites out of 3000 sites located by the ministry of power could have flow ates less than 500 l/s .

Therefore , by consulting data collected , and a number of those in charge in ministry of Jahad and ministry of power and also by consulting both supervisors at I.R.O.S.T. and Glasgow university it was decided that work should concentrate on micro turbines, operating under heads below 10 meters and flow rates less than 500 ( l/s).

# CHAPTER

# ( II )

# THE THEORY

# *Turbomachinery*

This is the term applied to describe the types of machinery in which energy exchanges (transfers) take place between the working fluid and the machine rotating parts (rotor, impeller ) as a result of a change of angular momentum occurring when the fluid passes through the machine .

The exchange of energy may take place in either direction , a turbine being a machine in which the fluid is doing work on the rotor , whilst in the pump , fan , or compressor the fluid has energy transferred to it from the impeller .

The working fluid may be incompressible or compressible and may flow in different paths through the machine ( rotor , impeller ) which leads to a further general classification based on flow direction through the machine rotor :

**( a )** Radial flow
**( b )** Axial flow
**( c )** Mixed flow

Also, variation of conditions of flow through the rotor ( impeller ) leads to the terms;impulse and reaction turbines.

Alternatively an Impulse turbine is one in which the change of the angular momentum of the fluid ( hence, energy transfer ) occurs mainly or wholly as a result of the change in direction of the fluid velocity relative to the blade . No change in pressure occurs in the blade passages and no change in the magnitude of the relative velocity vector .

A Reaction turbine is one in which due to the shaping of the blade passages changes in the magnitude and direction of the fluid velocity relative to the blades occur . This shaping of the passages producing an increase in the relative velocity also results in a pressure drop within the blades .

Turbomachinery development may be approached through fluids dynamics, experiments , dimensional analysis and in practice a combination of all three is used .

The fundamental design relationship for all turbomachinery is derived using the angular momentum law,which states :

The resultant external torque on the matter momentarily occupying a fixed volume equals the rate of change of angular momentum of the matter inside the volume plus the net rate of outflow of angular momentum through the control volume . For steady flow ( or cyclic flow ) such as considered to exist in the rotor of a turbine or compressor the rate of change of the angular momentum of the matter inside the volume is zero , thus ;

| The resultant external torque about an arbitrary fixed axis | = | Net rate of outflow of angular momentum through the volume about that axis |
|---|---|---|

$$\text{i.e.} \qquad T_Z = \dot{m}\,\Delta\,(r \times U)_Z$$

Study of the flow through a rotor is best carried out using velocity vectors , and constructing vector or velocity diagrams . They can be constructed for any position through the rotor passage but usually only for the inlet and outlet positions to give conditions for the inlet and outlet of the blades .

Absolute Velocities are with respect to the machine casing .
Relative Velocities are with respect to the machine rotor .

In general the absolute fluid velocity may be resolved into three mutually perpendicular components at radius r : ( Figure 2.1 )

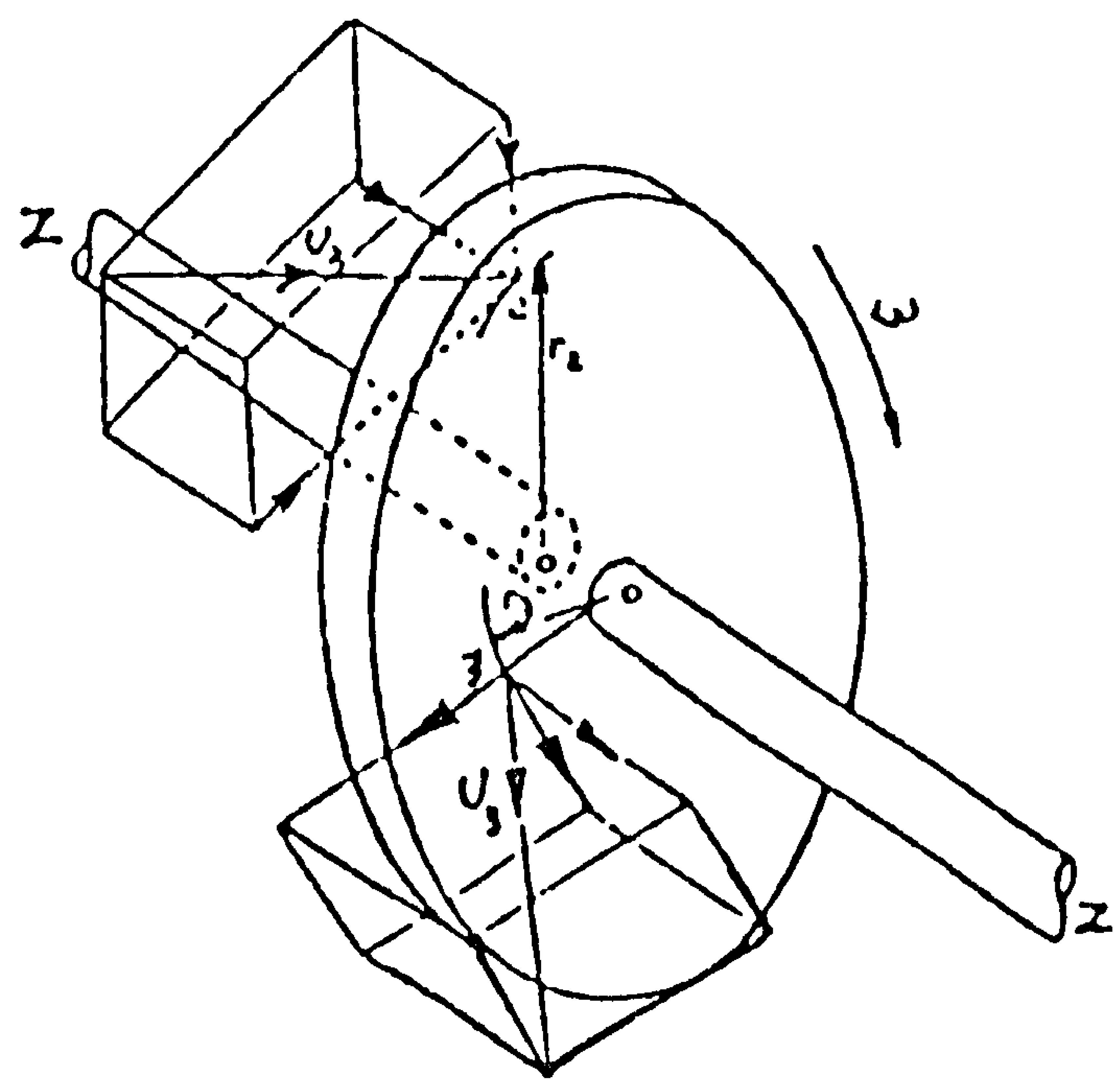



**Figure (2.1)**

*(a)* directed parallel to the axis of rotation of the rotor $U_{Axial}$

*(b)* directed radially through the axis of rotation of the rotor $U_{Radial}$

*(c)* directed at right angles to the radial direction $U_{Tangential}$ ($U_w$)

The absolute velocity of the blade at radius r is given by

$$U_b = \omega \times r$$

Since only the tangential component of the absolute fluid velocity , called the whirl component , $U_W$ , affects the angular momentum change about the axis of rotation , then

$$T_Z = m \; \Delta ( r \times U_W )$$

and between inlet ( 2 ) and outlet ( 3 ) of the blade ,

$$T_Z = m \; ( r_2 \times U_{w_2} - r_3 \times U_{w_3} ) \quad \text{on rotor ,}$$

where conditions are assumed constant across inlet and outlet sections .

The fluid velocity component normal to Uw and indicating the direction of fluid flow through the rotor is called the flow component , Uf ,

<u>ie</u> $U_f = U_{axial}$ , axial flow machine
$U_f = U_{radial}$ , radial flow machine

Any change in the components in the axial or radial directions produce thrusts on rotor shaft taken by bearings .

The construction of vector or velocity diagrams for the blade inlet and outlet sections is carried out using information on the absolute fluid velocity , or its components , the blade velocity , $U_b$ , and the blade angles etc .

The vector triangles are drawn for the plane containing the fluid whirl and flow components . The fluid absolute velocity issuing from the nozzle or guide blades is normally quoted as that value in that plane and is thus the vector sum of the whirl and flow components .

Typical inlet and outlet diagrams for a turbine are shown in figure ( 2.2 )

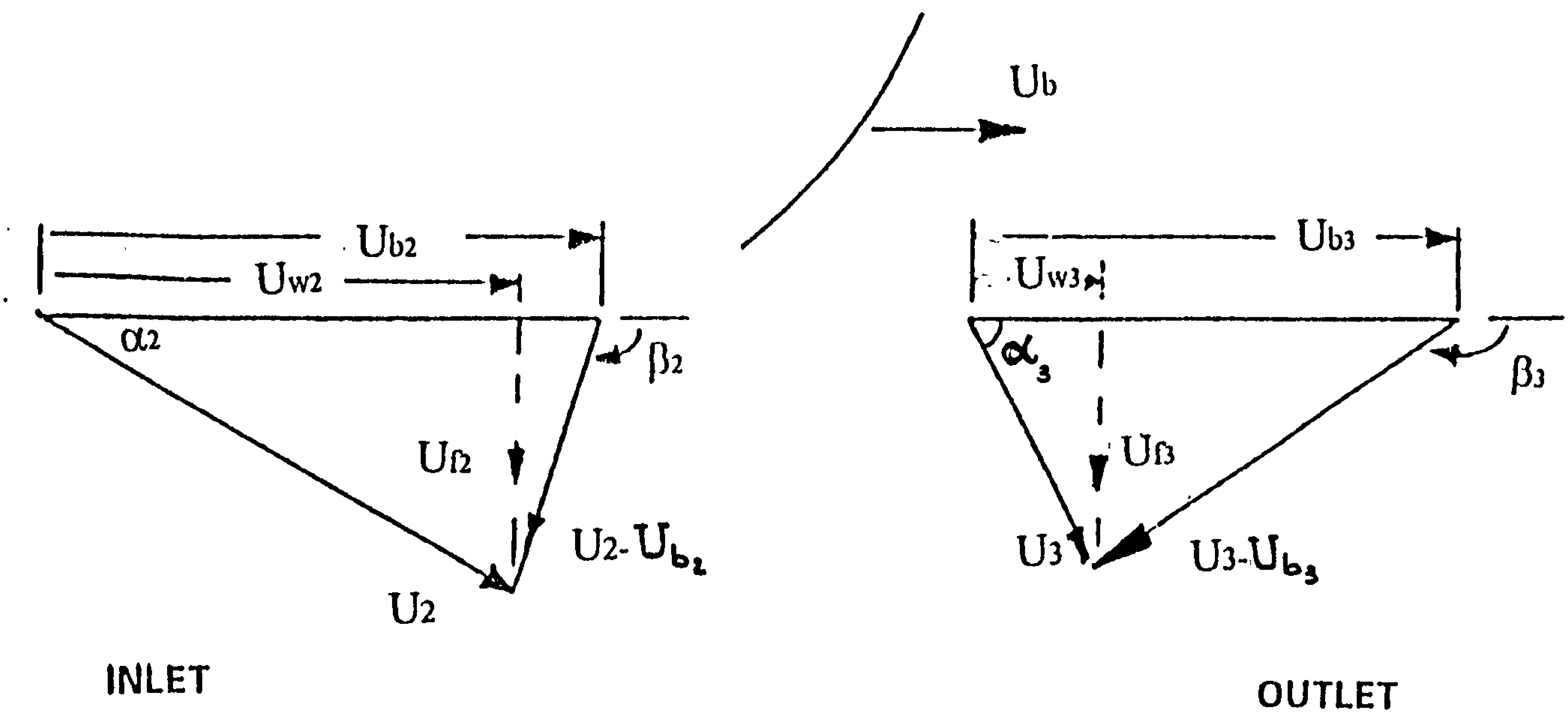


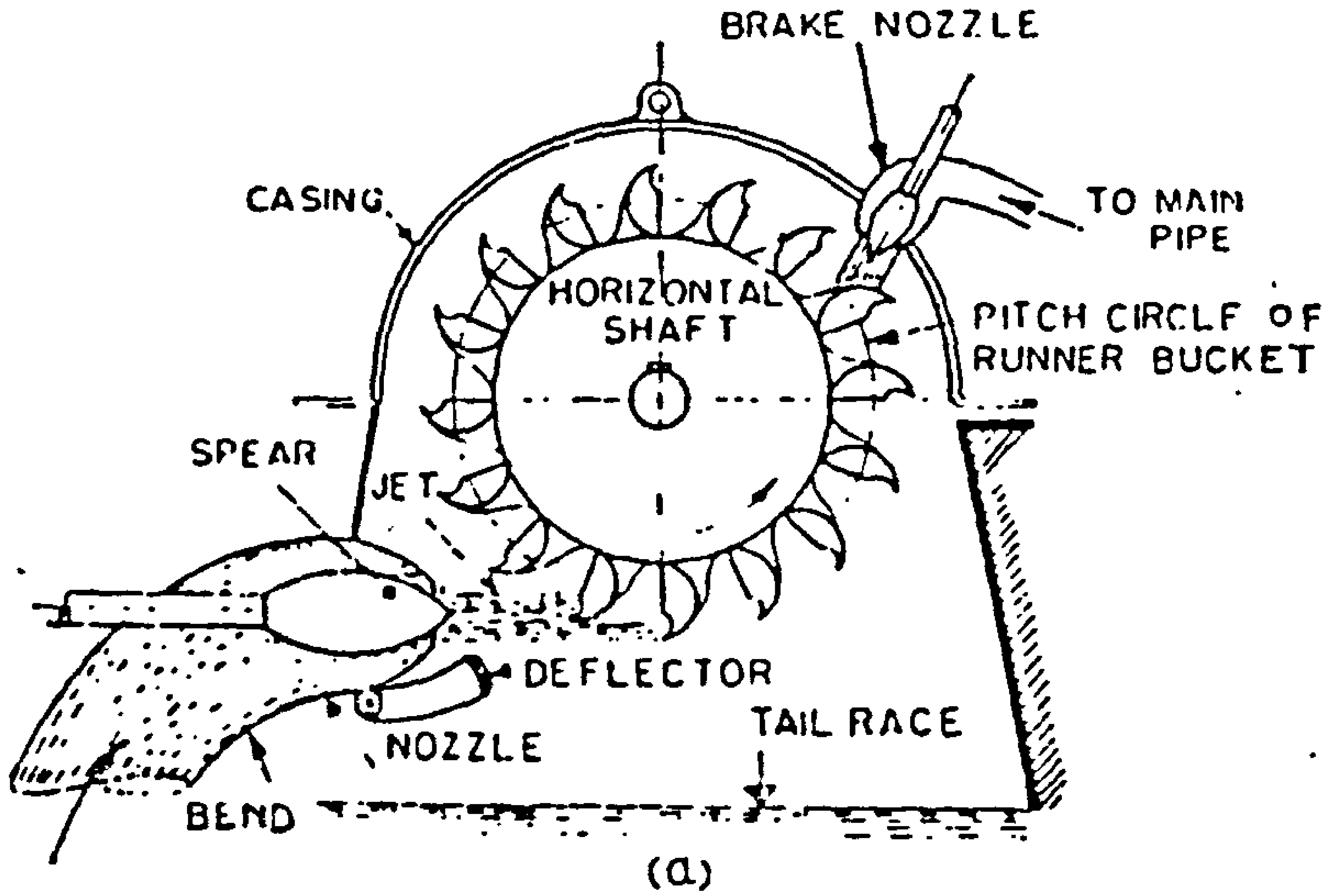

Figure (2.2)

## *Hydro Turbines*

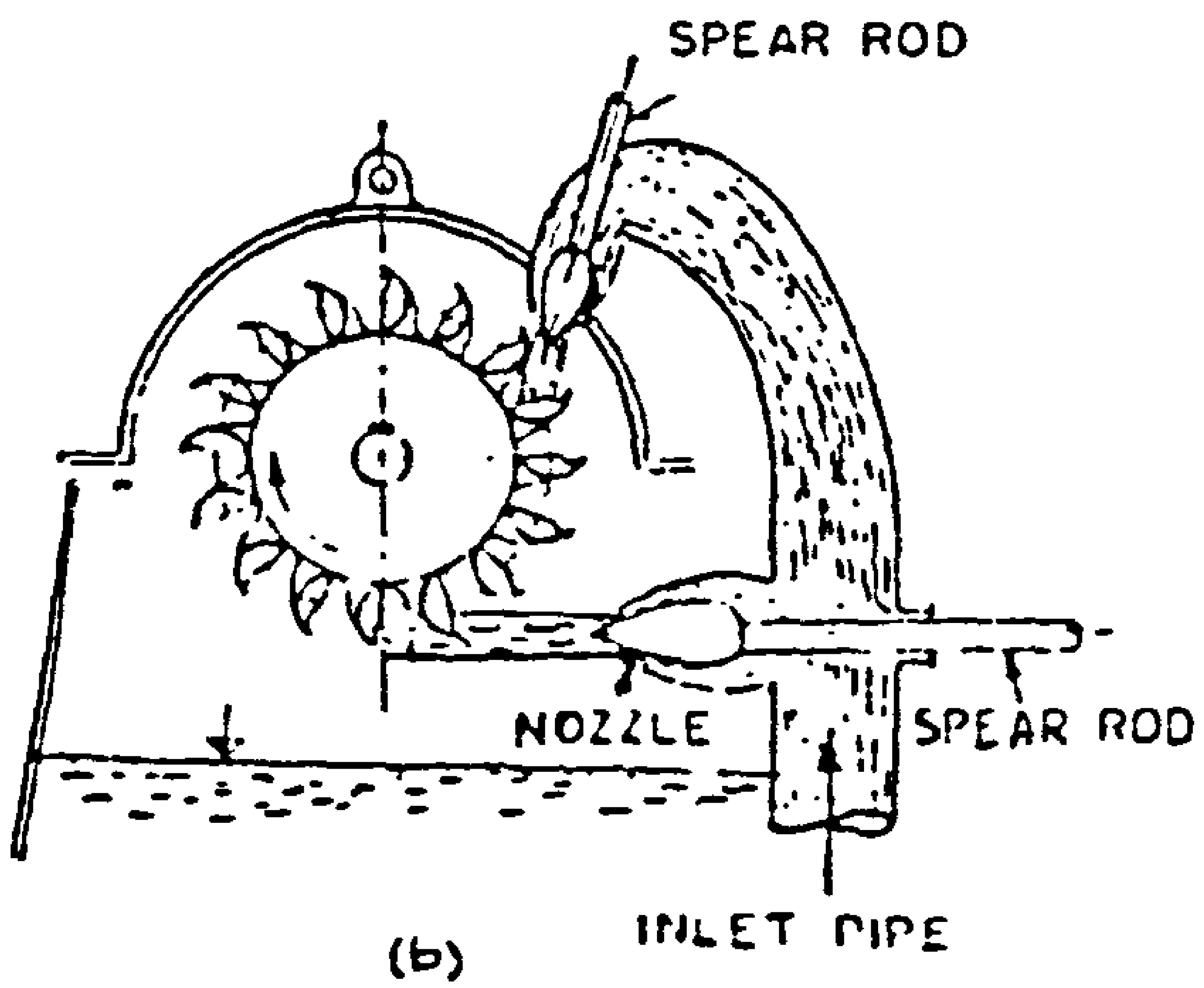
Usually water turbines fall into two categories ;

*1-* Impulse turbines .
*2-* Reaction turbines .

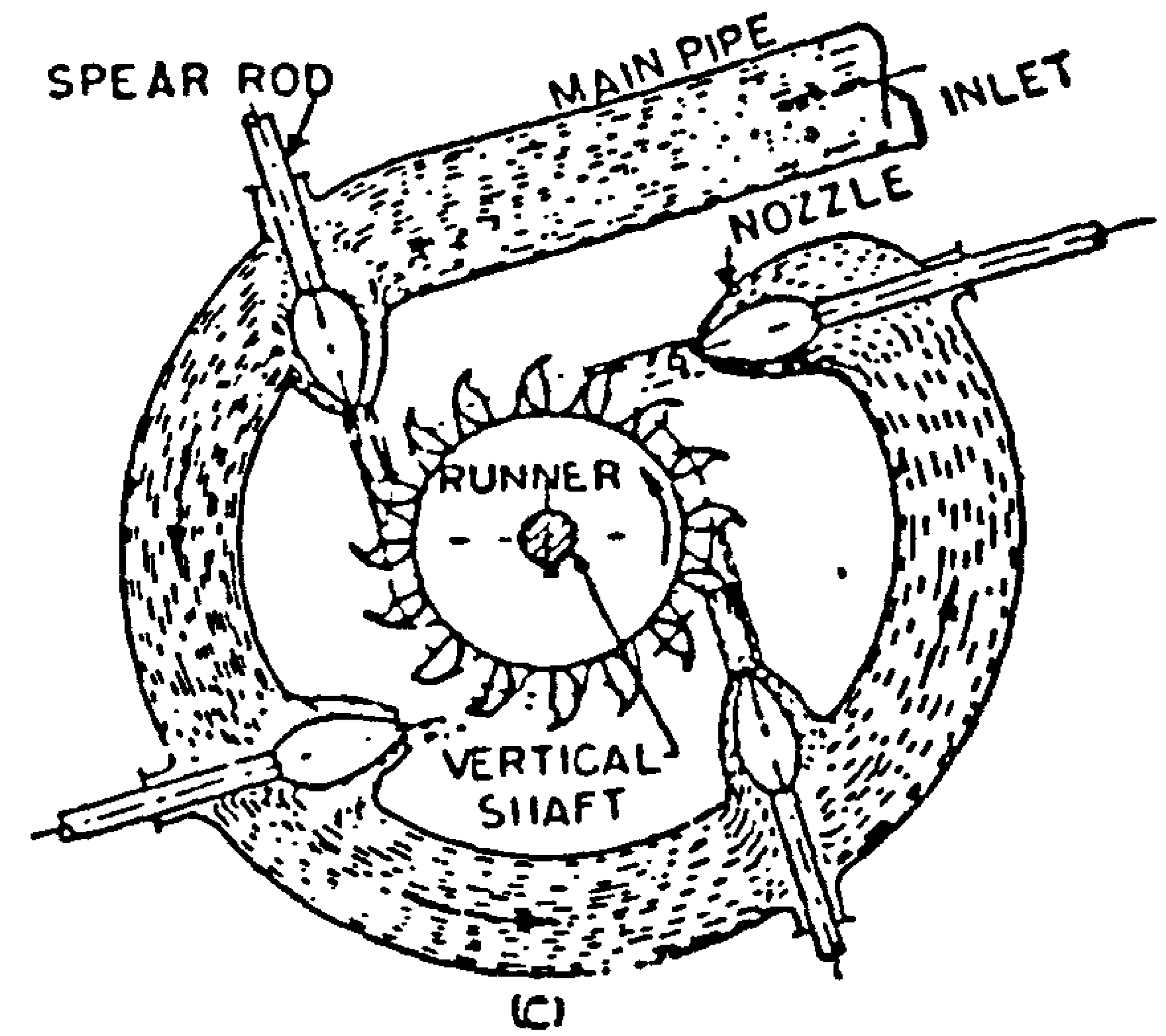
Amongst the well known impulse turbines are Pelton turbines , which are employed under high heads and relatively low flowrates . Banki turbines are also of the same category,but not as widely used as pelton turbines . These turbines are shown in figure ( 2.3 ) .




(a) Single Jet, Horizontal Shaft Pelton Turbine

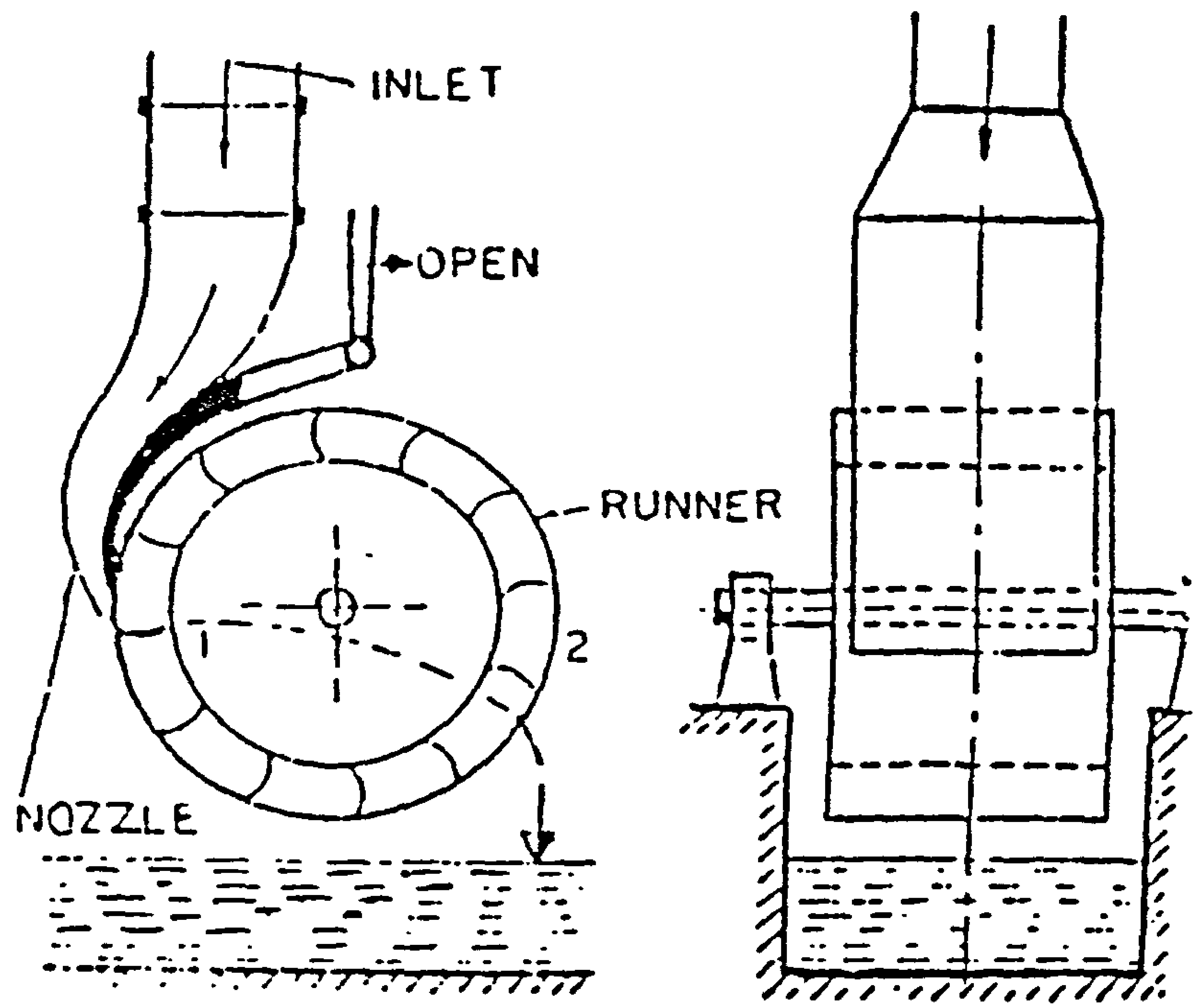



(b)




(c) Four-Jet Vertical Shaft Pelton Turbine

Figure (2.3)




Banki Turbine

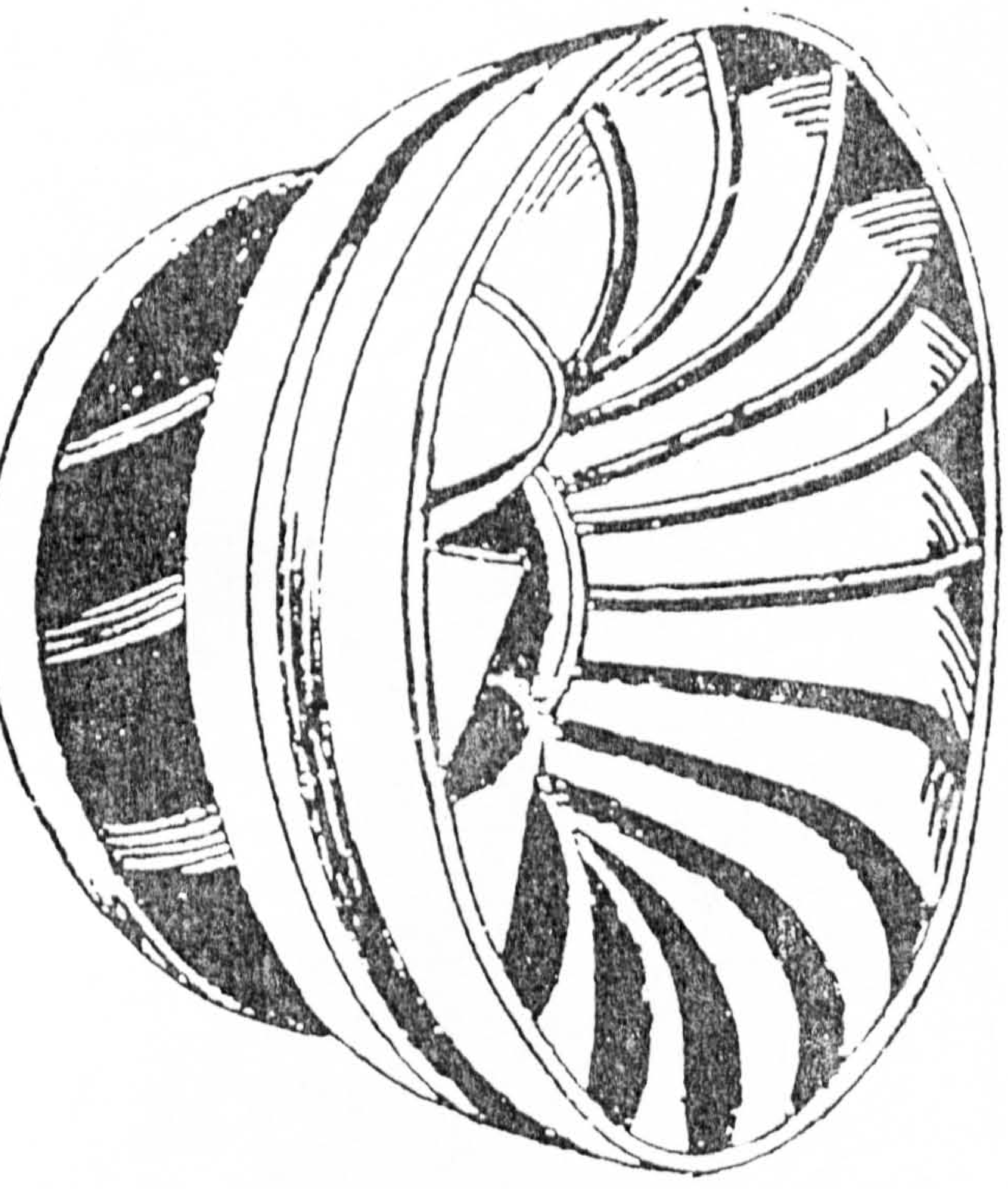
Here, a jet of water through a nozzle is directed toward turbine buckets ( blades ) thus transferring energy from water into the wheel . An important point here , is the fact that water flows into open ( atmospheric conditions ) immediately after leaving the nozzle , so that , the water jet would be impinging the blade under atmospheric conditions , where it freely splashes around .

There are also different types of reaction turbines . These types are mainly as follows :

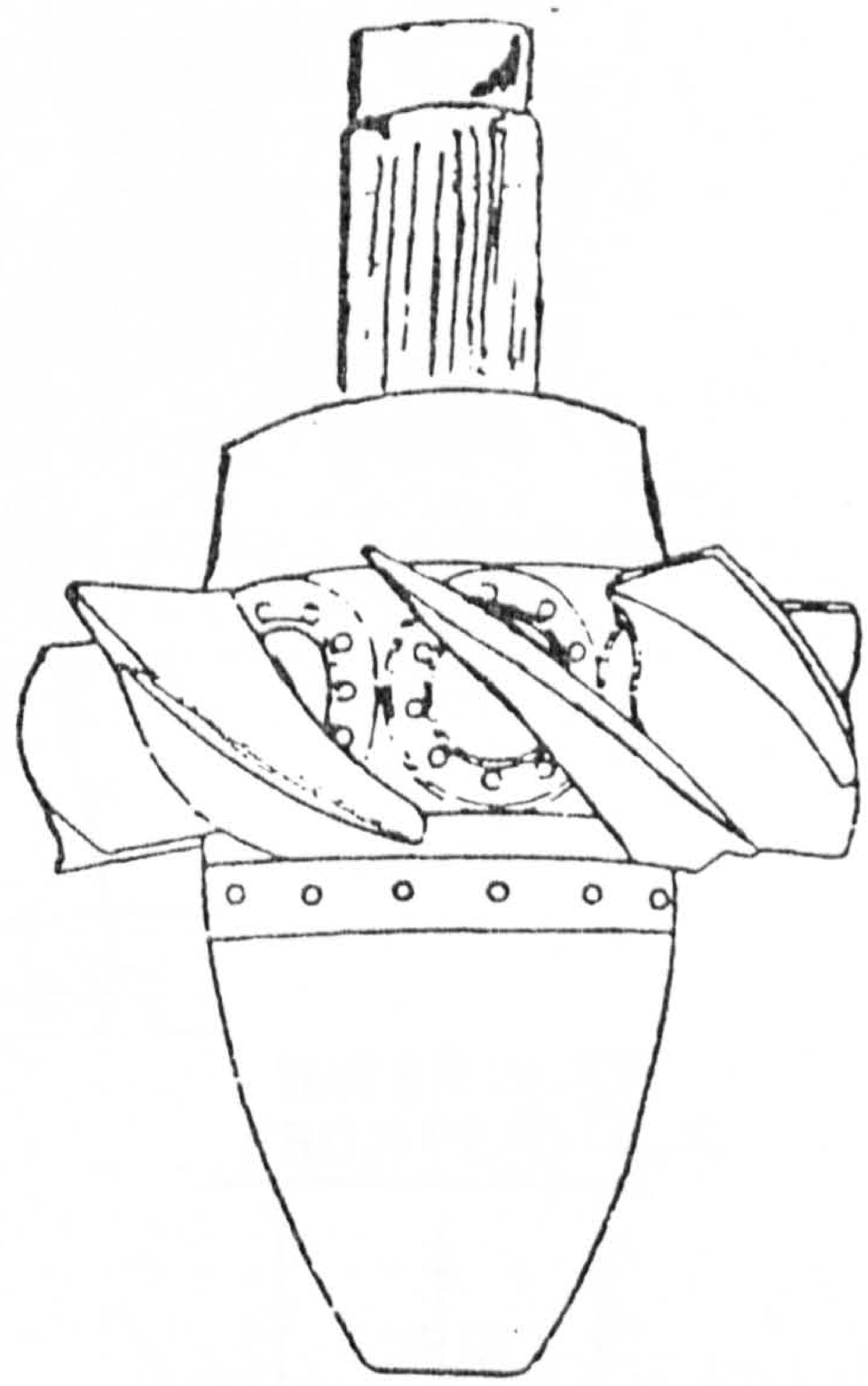
*1-* Radial inward flow , Francis turbine .
*2-* Mixed - flow Francis and Deriaz turbines .
*3-* Axial propeller ( fixed blade ) turbines .
*4-* Axial Kaplan ( adjustable blade ) turbines .

Figure ( 2.4 ) represents , different turbine runners . The main difference of these turbines with those of impulse type is that, water travels through a closed system , from entering the penstock or the turbine down to the exit from the draft tube . This system mainly consists of three parts as follows :

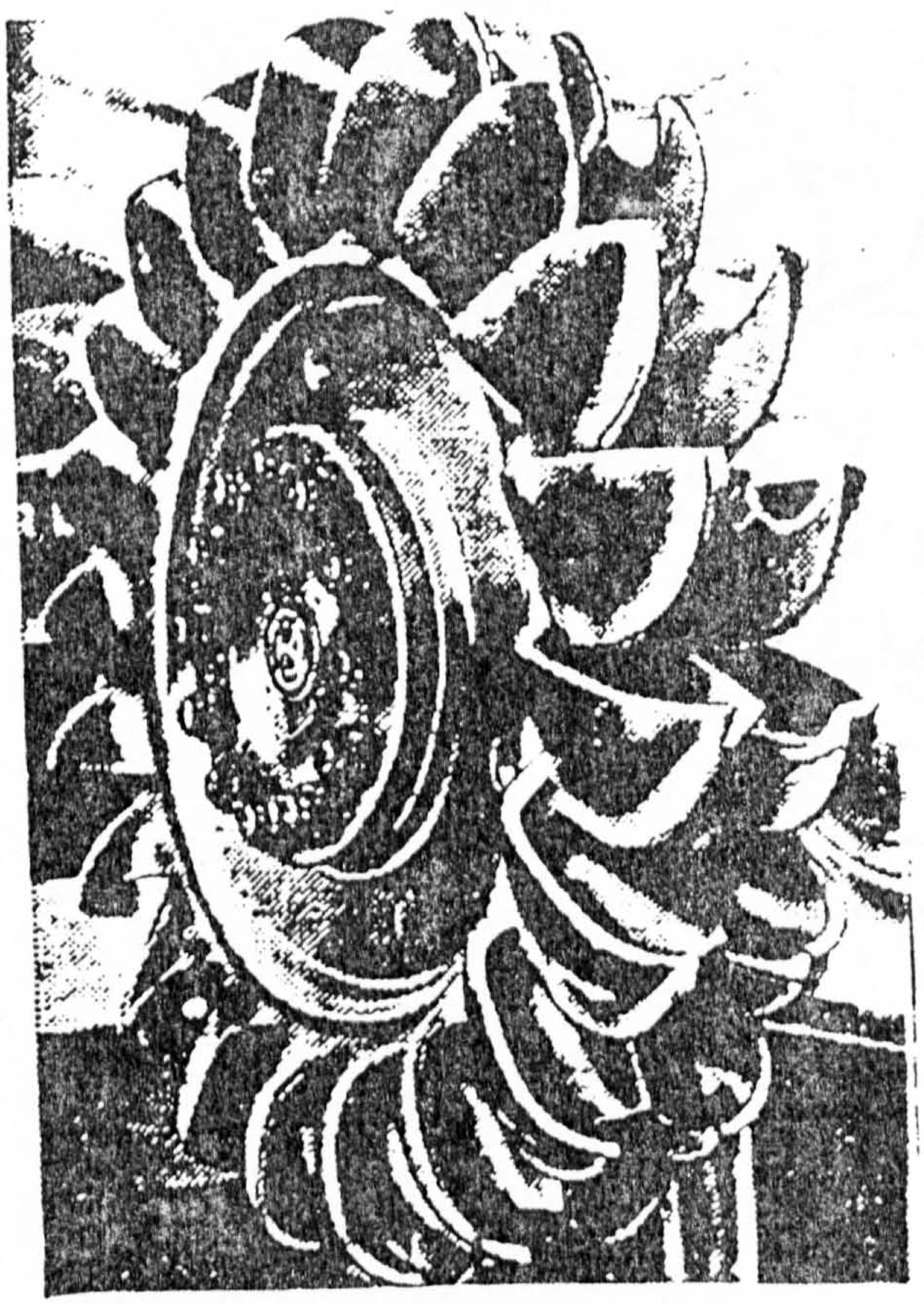
*1.* ***Penstock :*** pipes of relatively large diameter carrying water under pressure from the storage reservoir to the turbine .
*2.* ***The Turbine:*** where a part of the water pressure head is transformed into kinetic energy .
*3.* ***Draft Tube:*** water leaving the runner has a high kinetic head (i.e . high velocity). Draft tube , normally a cone of smoothly increasing cross section , can reduce velocity , thus , converting the kinetic head into pressure head . Figure ( 2.5 ) shows a reaction turbine with its draft tube .



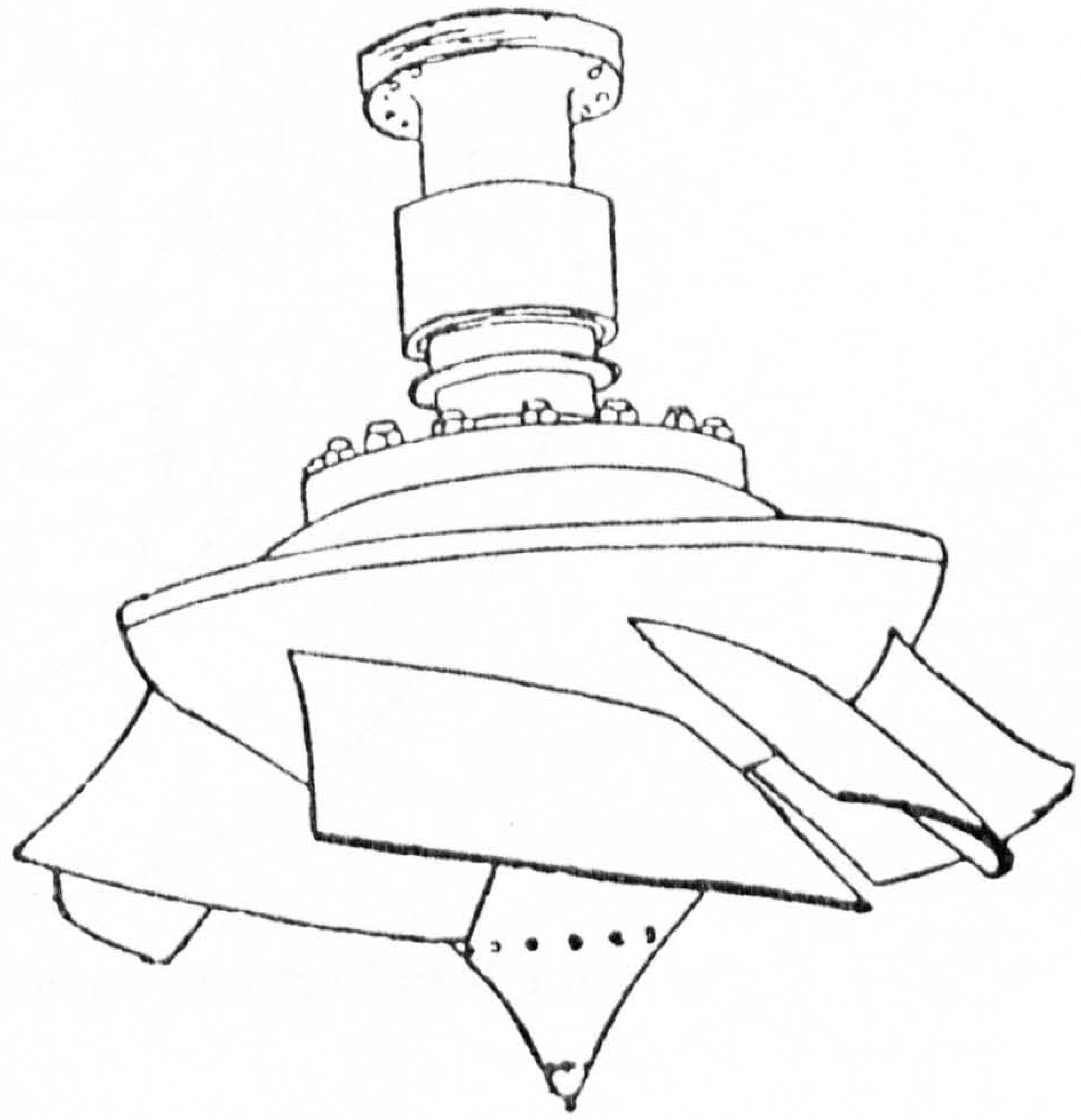
Francis turbine runner



Kaplan Turbine Runner



Pelton turbine runner



**Deriaz Runner**

Figure (2.4)

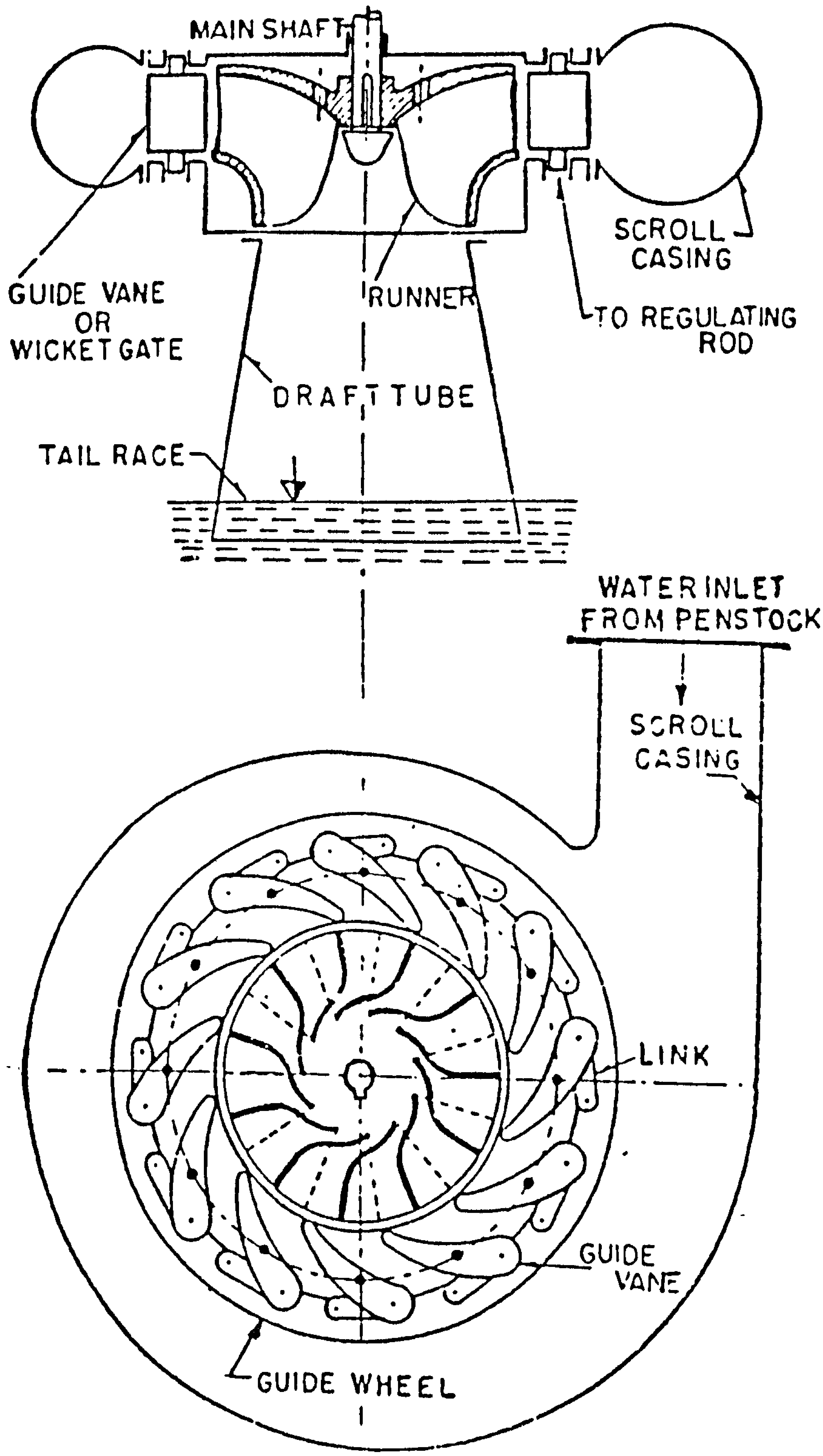



Figure (2.5) Outlines of a Francis Turbine, Vertical Closed Type

As mentioned earlier , water turbines maybe classified as, impulse and reaction turbines . Each of these categories contain a number of different type of turbines . Naturally , these different designs have been developed for a certain range of head and flow conditions . So far as , this work is concerned ; turbines suitable for relatively low heads and moderate flow rates would be of more importance . Figure ( 2.6 ) represents classification of different types of turbines . According to this figure propeller and Kaplan turbines seem more appropriate and applicable for our purpose . Therefore in next section some details about these turbines is presented to justify why such design has been selected for a micro hydro .

## ***Propeller Turbines***

This type of reaction turbine has a rotating element very similar to that of the axial flow pump . The propeller shaped runner is a development of the Francis mixed flow design and deals with large quantities of water . Figure ( 2.7 ) represent a Kaplan turbine .

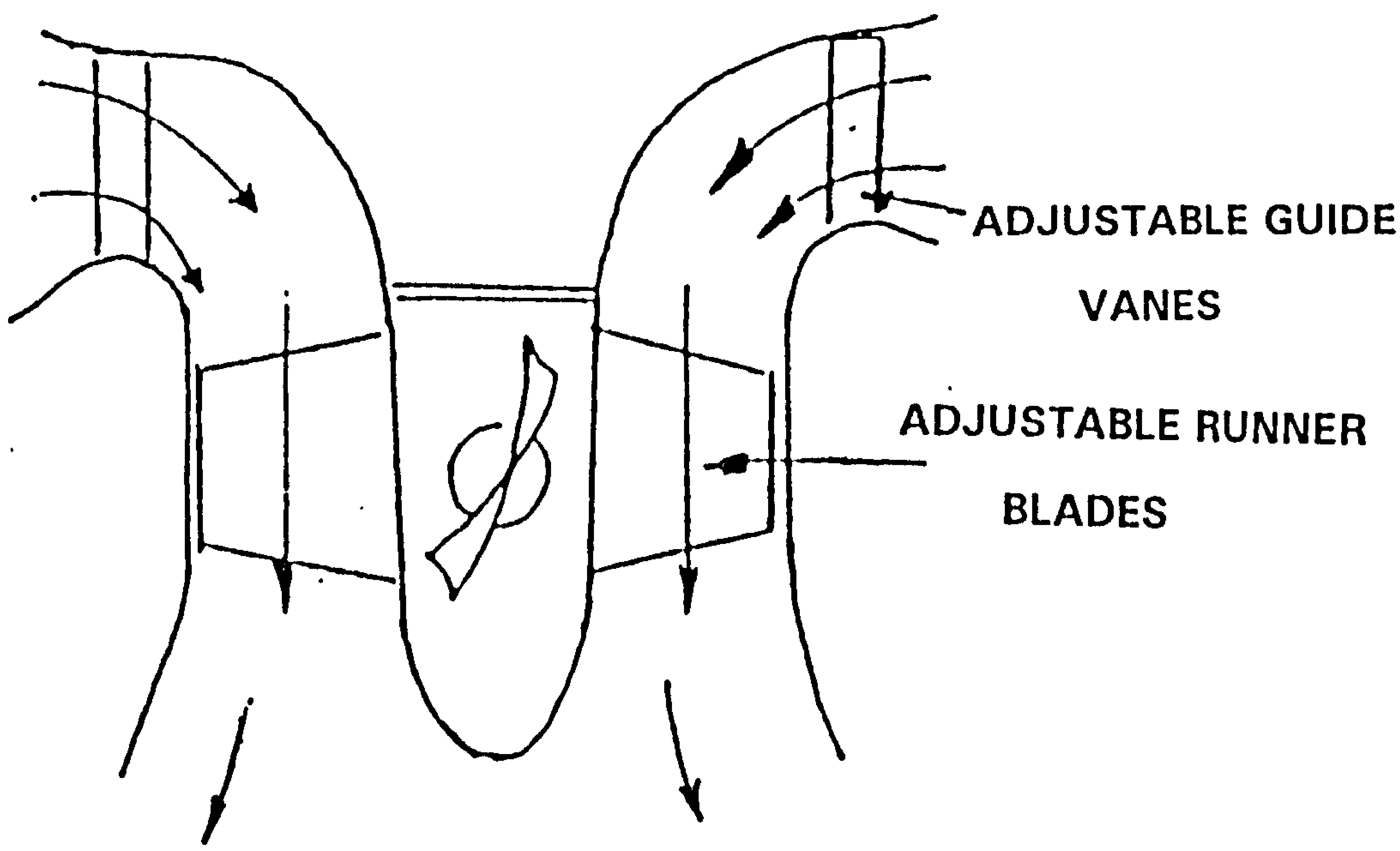



Figure (2.7) KAPLAN TURBINE

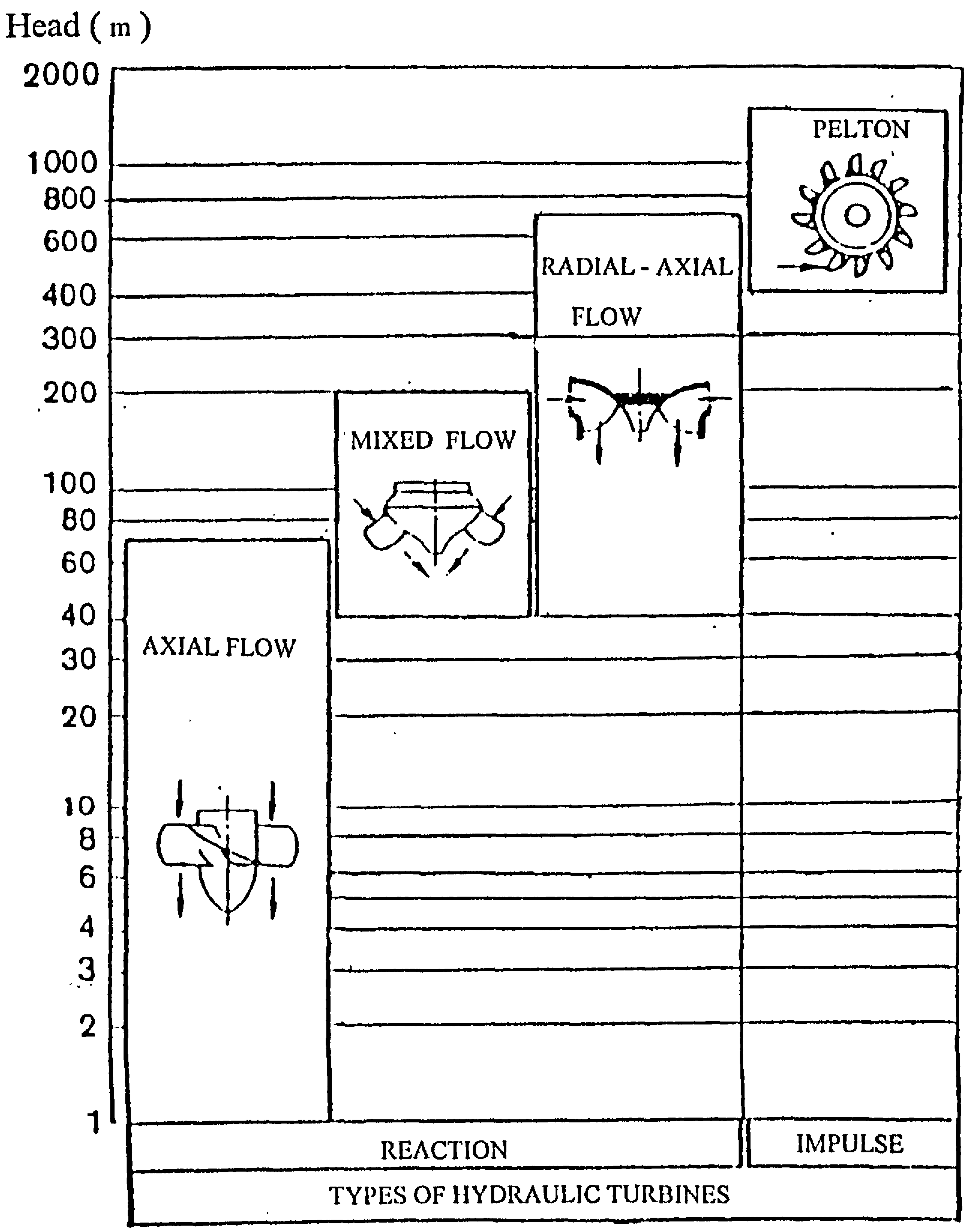



Figure ( 2.6 ) . Application of turbines of different types and heads.

The units operate under relatively low heads ranging from 4 meters to 40 meters, with a maximum of 70 meters for small machines. Separation and cavitation are often unavoidable in the flow passages of propeller turbines operating with high velocities and it is the onset of cavitation that the specific speeds, $N_S$, of these units range from 400 to 840 approx.

$$\text{where} \qquad N_S = \left( \frac{N \sqrt{p}}{(H)^{5/4}} \right) \eta_{max}$$

The power efficiency curve for a fixed blade propeller turbine is very peaked and hence, indicates a poor performance at part load. However, in 1920 Kaplan overcame the problem by developing a turbine in which the pitch of the propeller blades is automatically adjusted by a governor to suit a particular load, the actuating mechanism being accommodated in the boss and the hollow bore of the vertical shaft. The Kaplan turbine is more costly but the improvement at part-load is such that the fixed propeller unit is only installed at sites where the head and load are constant.

Figure ( 2.8 ) represents, a comparative sketch of the performance of Kaplan design with Francis and fixed blade propeller turbine under part loads. Also figure (2.9) shows the curve of efficiency against specific speed for pelton, Francis and axial turbines ( ie. Kaplan and propeller design ).

The bulb, tubular type of turbine installation is a development in the low head field (e.g. tidal power ). The installation consists of a fixed propeller or Kaplan type runner set axially within a short pressure conduit and the generator is close - coupled to the runner and housed in a " bulb " surrounded by flowing water. The runner generally has from 4-6 blades.

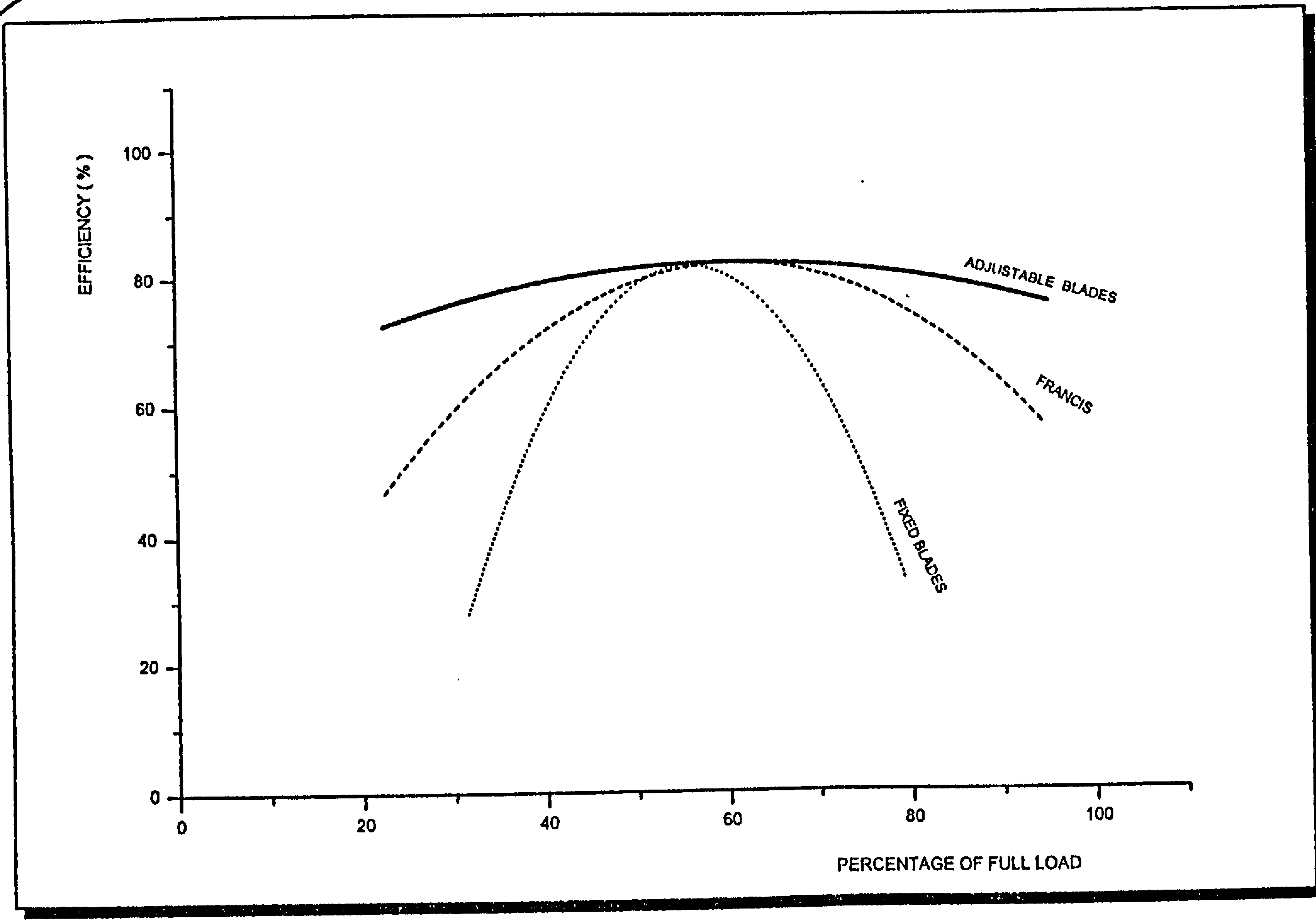



Figure ( 2.8 )

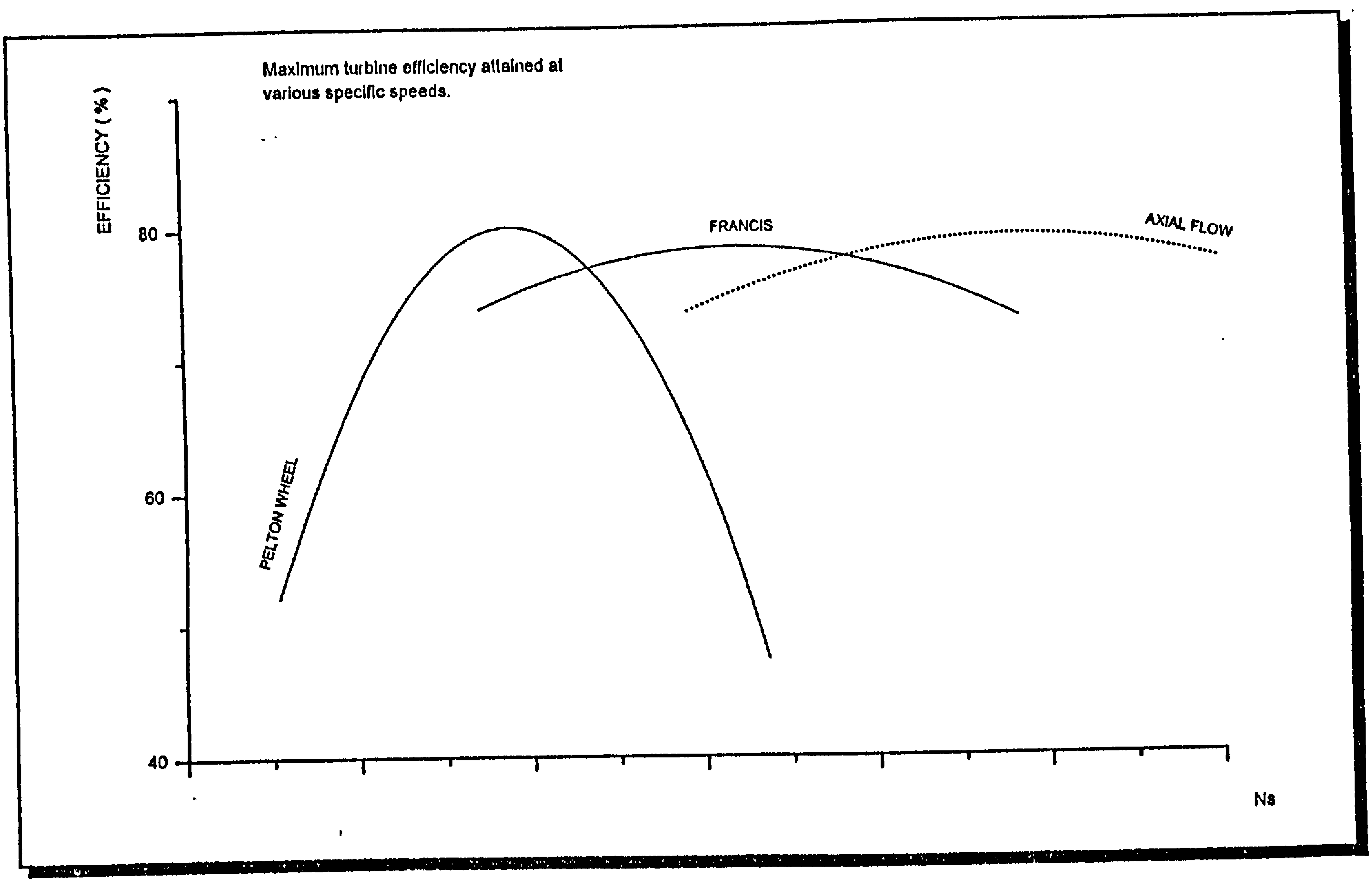



Figure ( 2.9 )

# *Theory of axial flow turbines*

A method of analysis of the axial flow turbine runner blade is based on the following assumptions :

*a)* the total head is constant across a flow section ,

*b)* the flow over the blades proceeds two dimensionally in essentially cylindrical layers which are concentric with the runner axis and with no mixing of adjacent stream - tubes . This implies that radially the change in static pressure balances the fluid centripetal forces and hence that the radial velocity component is zero . $U_r = 0$

*c)* the axial velocity component $U_f$ is constant throughout the runner .

These may be satisfied by suitable design requiring a free vortex flow distribution ( $U_w \times r$ = Const . ) from root to tip before and after the runner blades .

From the angular momentum law

WD / unit mass = $(U_{b2} U_{w2} - U_{b3} U_{w3})$ for a stream- tube

, thus if

$$U_{w2}\, r_2 = \text{Const} \qquad U_{w3}\, r_3 = \text{Const} . \text{ ( preferably - 0 )}$$

then since $U_{b2} = r_2 \times \omega$ and $U_{b3} = r_3 \times \omega$

$$\text{WD / unit mass} = \omega \; (r_2 U_{w2} - r_3 U_{w3})$$

which is constant at all radii

$$\text{WD / unit time} = m \; (U_{b2} U_{w2} - U_{b3} U_{w3})$$

& for $U_{w3} = 0$ WD / unit time $= \rho Q\, U_{b2}\, U_{w2}$

i . e. blade power $= \rho Q\, U_{b2}\, U_{w2}$

If the nett available head $= H$ ( const . across runner )

available power $= \rho g Q H$

$$\text{hydraulic efficiency},\ \eta_h = \frac{\text{blade power}}{\text{available power}}$$

$$\text{i. e.} \qquad \eta_h = \frac{U_{b2}\, U_{w2}}{gH} \qquad \text{(const .at all radial )}$$

$$\text{mechanical efficiency},\ \eta_m = \frac{\text{brake power}}{\text{blade power}}$$

$$\text{overall efficiency}, \qquad \eta_o = \frac{\text{brake power}}{\text{available power}} = \eta_m \times \eta_h$$

## ***Micro hydro Turbines***

Micro hydro turbines have attracted attention during past few decades , after successful application of small and specially mini hydro turbines . These turbines are usually of simple design , light weight , low manufacturing costs and do not require heavy and costly construction for installing and operation. They may specially be applied at sites with low heads and limited flowrates such as rivers , runs off rivers , irrigation canals etc.....

Historical background of hydro turbines show that old Persians were the very first people who made use of hydro power as water mills ( 1 ) . In present days , as far as the micro hydros are concerned , a number of countries have entered this field . Amongst them China is the country which has mostly benefited micro hydros . About 60000 of micro turbines in four types have been designed , manufactured and installed across China ( 4 ) . Japan is another country which has worked on small and micro hydros . Fuji electric Co. is one of the Japan's companies which has tried to standardize these turbines . Their work involves 7 different types of turbines covering a power range of 20 MW to 50 KW (5 ). New Guinea is another country which has benefited application of micro hydros at her rural areas . In the U . K . also a number of companies have embarked upon design and production of micro hydros (6 ) . But lack of government back up has almost paralyzed the work . Agnew turbine is one of the micro hydros developed in the U . K ., a number of which were actually manufactured and installed at different sites in Scotland . However , due to the reason mentioned above no research was done on this turbine to improve its efficiency ( 7,8 ) .

Apart from that , some research work has also been undertaken on establishing criteria for selection of micro hydros according to the available head and flow rate (9). As far as Iran is concerned , no work has yet been undertaken on micro hydros . According to a report by the ministry of power of Iran (2 ) , around 3000 natural sites suitable for micro hydros have been spotted around the country . However , this figure does not include runs off rivers , irrigation or any other man made canals . Due to ever increasing demand for electricity and daily reduction of fossil fuel resources , Iranian government has given priority to renewable power generating resources such as hydro power . In accordance to the governmental programs work on hydro power began at I . R . O . S . T . by establishment of a turbo machinery group at its mechanical engineering research center . Design and establishment of a turbo machinery laboratory . and improvements on Agnew micro hydro turbine are the first assignments of this group . Prime objective of this project may be summarized as making improvements on Agnew turbine to achieve the highest possible efficiency . However , study of the work by different sources on micro hydros show that , apart from Agnew turbine , non of other turbines have been of Kaplan design . Therefore it was appropriate to concentrate efforts on this type of turbine .

## The draft tube

Usually, in reaction turbines water leaves the runner into a draft tube .Then through this tube, water would be discharged into the tail water pool . However , if application of draft tube is avoided , then for avoiding any loss of head the turbine should be placed below the tail race level . Such a turbine should always be immersed in water and not easily accessible . In order to eliminate this drawback, a draft tube would be used . Here, the turbine is located above the tail race level, where the runner is surrounded by an air tight draft tube which extends down below the tail race level .In this way the draft tube would serve two purposes :

**1.** Generally, if an airtight draft tube connects the runner to the tail race, then the workable head would be increased by an amount equal to the vertical distance between the tail race level and the runner outlet . Due to the presence of the draft tube, a suction ( negative ) head is established at the runner outlet . This phenomenon may be explained by the pressure difference present inside the draft tube, between the tail race level and the runner outlet . This pressure diffrence is due to the fact that the pressure in the draft tube at the tail race level is amospheric . If the cross section of the draft tube is kept uniform, then the pressure at the runner outlet would be equal to the atmospheric pressure minus the vertical distance between the tail race level and the runner outlet . Therefore the available head from the head race level would be the same as if the turbine were installed at the tail race level, discharging into atmosphere .

In this way, the turbine would be installed above the tail race level thus accessing to it for repair work purposes or any other reasons would be much easier than otherwise .

Sometimes the above is referred to as "the hydraulic characteristics" of draft tubes, and it may be explained in more detail as follows :

### 1.1 Hydraulic characteristics

Referring to the figure (2.10), let sections 2-2 and 5-5 be at the exit of the runner and at the end of the draft tube in tail race pool respectively.

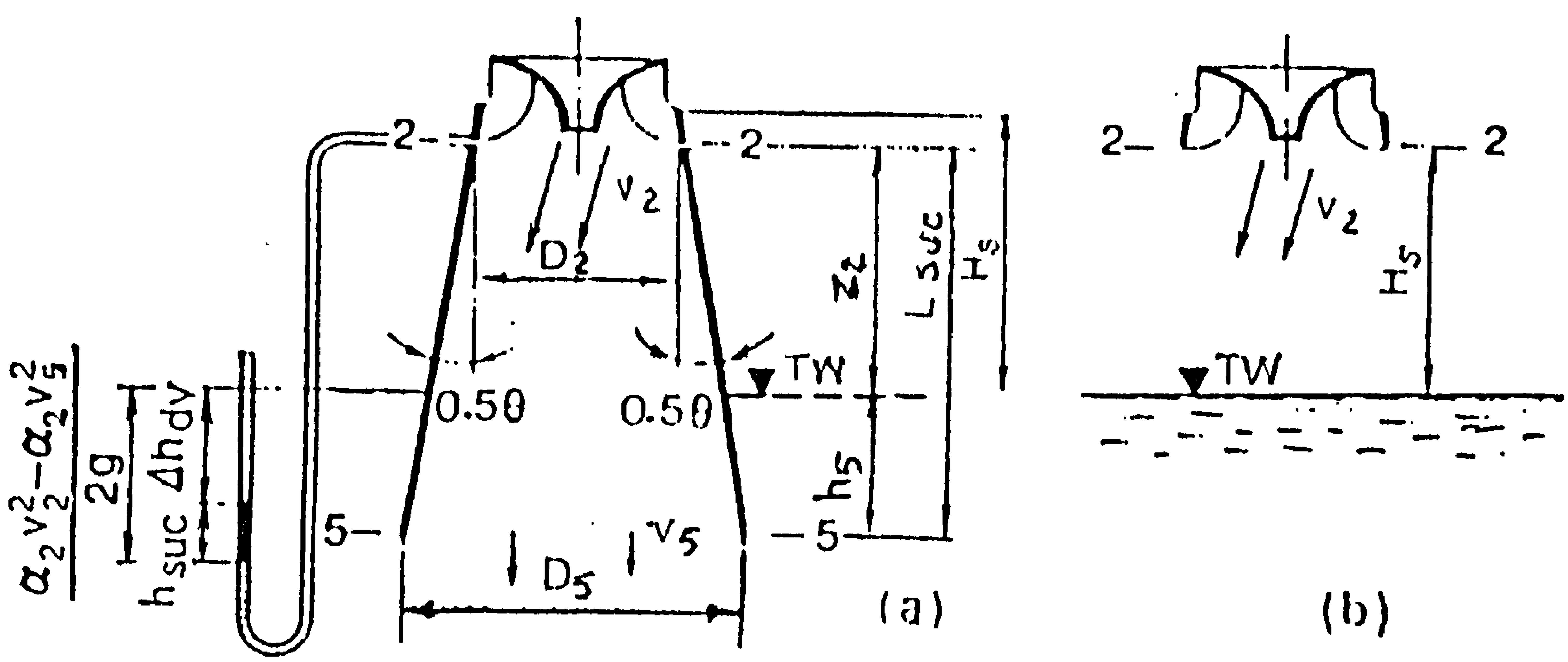



**Figure (2.10) . Hydralic characteristics of draft tube**

By applying Bernoulli's equation between these two sections it can be shown that :

$$\frac{P_{a2}}{\rho g} + \frac{k_2 U_2^{\,2}}{2g} + Z_2 = \frac{P_{a5}}{\rho g} + \frac{k_5 U_5^{\,2}}{2g} + Z_5 \qquad (1)$$

Where $\frac{P_{a5}}{\rho g} = h_5 + \frac{P_{atm}}{\rho g}$ and $h_5$ is the depth of section 5-5 below the level of tail water pool. $P_{atm}$ is the atmospheric pressure, $Z_5 = - h_5$, and $h_f$ represents the hydraulic losses in the draft tube . Also $H_s$, known as suction height, stands for the height at which the turbine is installed above the tail race .

K is corioli's coefficient allowing for non-uniform velocity distribution over the section . (15 )

Through the above equation and substitution of some values, the amount of pressure at section 2-2 would be given as : -

$$\frac{P_{a2}}{\rho g} = \frac{P_{atm}}{\rho g} - \left( H_s + \frac{K_2 U_2^{\,2}}{2g} - \frac{K_5 U_5^{\,2}}{2g} - h_f \right) \qquad (2)$$

It may be observed from equation (2 ) that pressure 2-2 ( i .e . $P_{a2}$ ) would be below atmospheric pressure . Obviously this negative pressure (or vacuum) is caused by presence of the draft tube without which water would have been discharged into atmosphere ( i .e . $P_{a2} = P_{atm}$ ) . This vacuum is equal to the sum of two parts as follows :

**i** . Static fall of pressure ( $H_s$ ) which is actually independent of the discharge .

**ii** . Dynamic fall of pressure

$$\Delta_{hdy} = \frac{K_2 U_2^{\,2}}{2g} - \frac{K_5 U_5^{\,2}}{2g} - h_f$$

2 . If as in a pelton turbine, water from a reaction turbine is discharged into atmosphere,then a considerable amount of kinetic energy would be lost . This kinetic energy is as a result of a high velocity which water possesses when leaving the runner . Reduction of this velocity may be accomplished by application of a draft tube of smoothly increasing cross-section area . The discharge velocity reduction would then result in a pressure head;which would in turn cause an increase in the negative pressure head at the runner outlet . This would lead to an increase in the working head of the turbine, causing a higher output and thus raising the efficiency of the turbine .

The above may be referred to as "*The power generating characteristics*" of the draft tube ; which is explained in more detail as follows : -

## 2.1 Power generating characteristic

At section 2-2 (figure 2.10 ) the average specific energy of water ($e_2$) is given by the following equation :

$$\frac{P_2}{\rho g} + \frac{k_2 U_2^{\,2}}{2g} + Z_2 = e_2 \qquad (3)$$

Where : $$\frac{P_2}{\rho g} = \frac{P_{a2}}{\rho g} - \frac{P_{atm}}{\rho g}$$

As mentioned above this energy ($e_2$) cannot be utilised by the turbine and therefore is regarded as a loss .

By applying expression (3) in the equation (2), behavior of ($e_2$) in presence of a draft tube may be studied :

$$e_2 = -\left( H_s + \frac{K_2 U_2^2}{2g} - h_f - \frac{K_5 U_5^2}{2g} \right) + H_s + \frac{K_2 U_2^2}{2g}$$

This would result to :

$$e_2 = \frac{K_5 U_5^2}{2g} + h_f \qquad (4)$$

From expression (4) it is concluded that $e_2$ is equal to the sum of outlet losses $K_5U_5^2/2g$ and internal losses, $h_f$ .

To reduce the overall losses , thus increasing the turbine's efficiency, $e_2$ , should be reduced as much as possible . This may be achieved by increasing the outlet cross-section area (at section 5-5), since an increase in the area would lead to a reduction in velocity . Also, $h_f$ which consists of wall friction losses and vorticity losses due to diffusivity should be minimized to lowest possible level .

## Efficiency of a draft tube

Let us assume that there is no draft tube, and water is discharged into atmosphere, straight from the runner into the tail water pool . Then $e'_2$ will be given by :

$$e'_2 = \frac{K_2 U_2^2}{2g} + h_f \qquad (5)$$

A comparison of expressions (4) and (5) would establish the fact that the draft tube increases the energy utilized by the turbine by

$$\Delta H_f = H_s + \frac{K_2 U_2^2}{2g} - \frac{K_5 U_5^2}{2g} - h_f \qquad (6)$$

The efficiency of the draft tube would then be given by :

$$\eta_D = \frac{\dfrac{K_2 U_2^2}{2g} - \dfrac{K_5 U_5^2}{2g} - h_f}{\dfrac{K_2 U_2^2}{2g}}$$

Generally, for a well designed and sufficiently long draft the value of $\eta_D$ could be as high as 80-85 percent .

However, dependence of the kinetic energy loss on the head for complete turbine power may be observed from figure (2.11 ) .

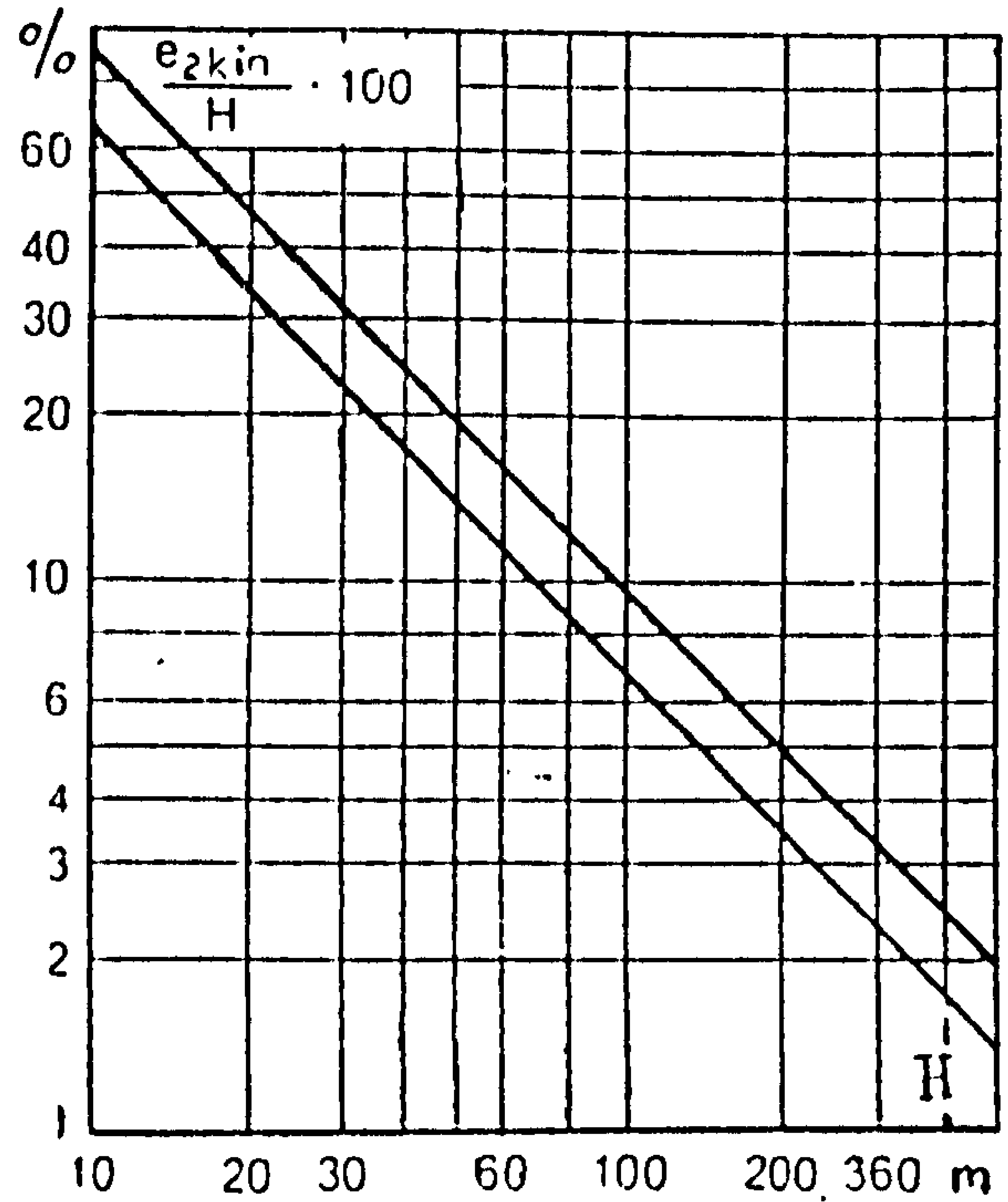



**Figure ( 2.11 ) . Relative kinetic energy downstream of runner versus turbine head.**

According to this graph, water leaving the runner in the low head turbines can possess as much as 50-90 percent of the total energy,H. As the effective head rises this value decreases, as seen, for values of H larger than 100 (m) it falls even below 10% .

Therefore, it can be concluded that application of a draft tube ( i.e. utilizing the kinetic energy of water, leaving the runner, $e_2$ ), especially in low head and high speed turbines, can be of paramount importance to raising the effectiveness of the turbine (10,15,16) .

## DIFFERENT TYPES OF DRAFT TUBES

As with turbine design, there are also many different types of draft tubes . However, there are two main categories of draft tubes; straight divergent draft tubes,and elbow type draft tubes .

Although at present discussing all details of various designs of draft tubes is out of the scope of this report, a summary of some of these designs would be presented as follows :

### I . Straight divergent draft tubes

#### i . Simple straight draft tubes

This is the simplest type of straight divergent draft tubes . Its shape is that of a frustum of a cone . It usually possesses good power generating characteristics.However for the purpose of avoiding flow separation from the tube walls, it is essential to keep the cone angle to at the most 8° ( i .e . $\theta = 4°$ figure 2.11 a ) . (10) . This is due to the fact that this flow separation would lead to an unfavorable loss of head . Since the draft tube must discharge sufficiently below the tail race level, then for obtaining high power generating characteristics, it must be of a considerable required length .

The following is an expression recommended for selecting the cone angle:

$$\frac{\sqrt{F_5} - \sqrt{F_2}}{L} = \frac{1}{5} \text{ to } \frac{1}{8} \qquad (7)$$

Where $F_2$ and $F_5$ are cross-sectional areas of the draft tube at entrance and exit respectively, and L is the length of the draft tube .

Expression (7) ensures that the cone angle would stay between 7° to 11° (16). However experiments indicate that, for a conical diffuser, head losses tend to increase rapidly for cone angles over 10° (17).

**ii.** Curved wall straight draft tube

As it is seen in figure (2.11b), a curvilinear line has been used in the design of the draft tube. This could increase the effectiveness of the tube and decrease its length (15).

**iii.** Flare type draft tubes

This type of draft tube is, as a matter of fact, an improvement to the curved wall design, having a cylindrical outlet. (figure 2.11c).
To improve the efficiency of such a draft tube it is recommended that a conoid be constructed at the center of the tube. The idea of using a conoid has first been put forward by Moody, known as Moody's spreading type draft tube or "hydrocone". (10)

## II. ELBOW DRAFT TUBES

One of the drawbacks of straight conical tubes, especially in high speed turbines, is their lengths. Their long length would require extra excavations at the site. This could prove to be very costly, especially at rocky sites. To avoid such costly operations, elbow draft tubes are employed. There are many different designs of elbow draft tubes, of which only a couple would be mentioned.

**i.** Simple elbow tube

These draft tubes have circular entrance and exit cross-sections. The area of cross-section increases onwards from the entrance (from runner) to the exit (tail race) . Their efficiency is rather low, around 60% . (figure 2.12 a)

**ii .** <u>Elbow type, circular inlet, rectangular outlet</u>

This design ensures a higher reduction of velocity at the outlet by having larger outlet cross-sectional area with lower hights . However, in some designs the transition from circular to rectangular takes place at the bend, and in some others it takes place in horizontal part over a relatively great length .

Figure (2.12 b) shows an elbow type draft tube with rectangular cross-section at the outlet, transition taken place at the bend . Figure (2.13) also shows the same design with transition from circular cross-section taken place at the horizontal section .

To improve the efficiency of the elbow type draft tubes sometimes guide walls at the bend and piers at the exit are used . Figure (2.14) and (2.15 a) and (2.15 b) show different designs of elbow type draft tube with guide wall and piers respectively .

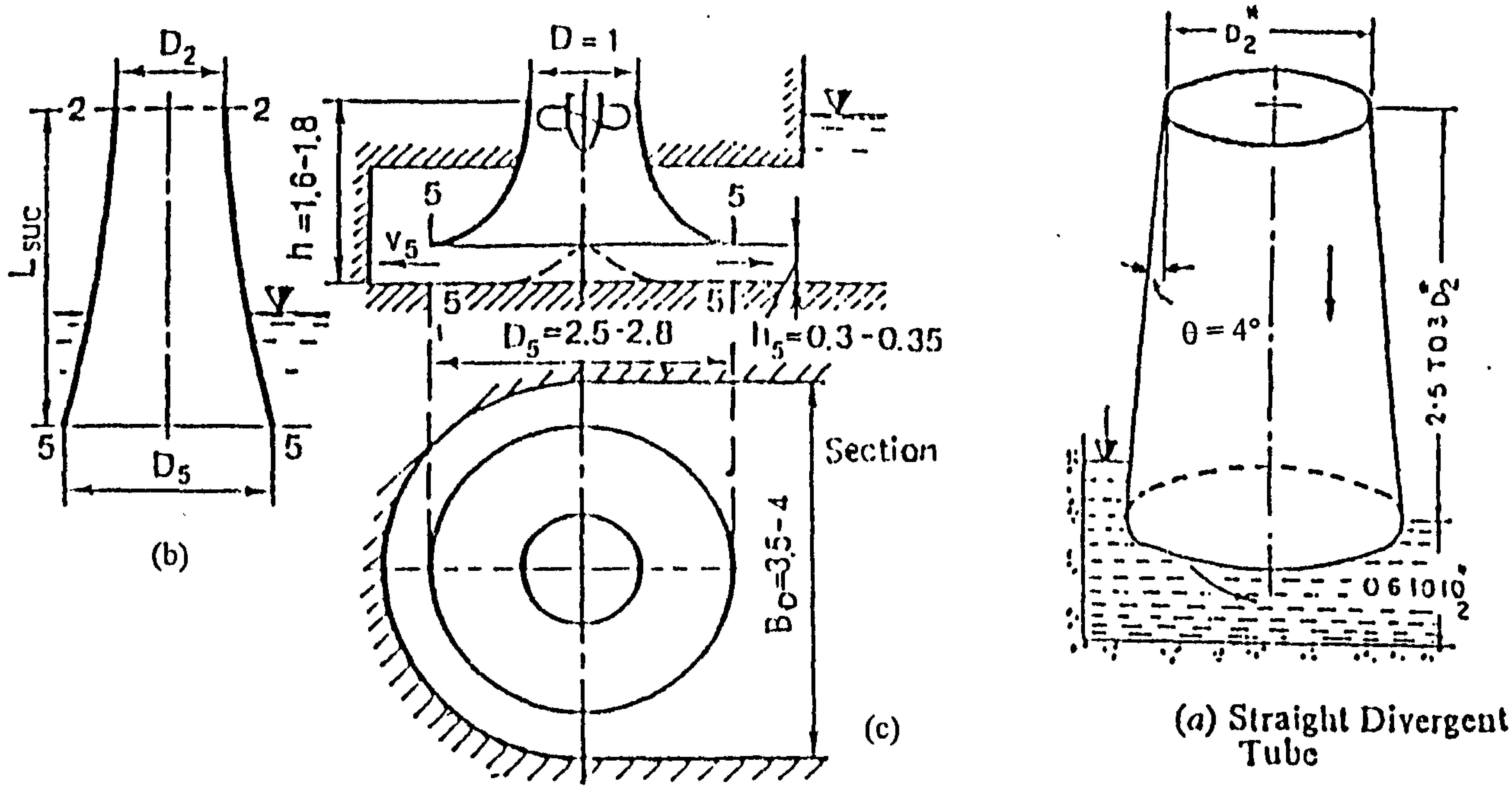



Figure 2.11 - Straight draft tubes

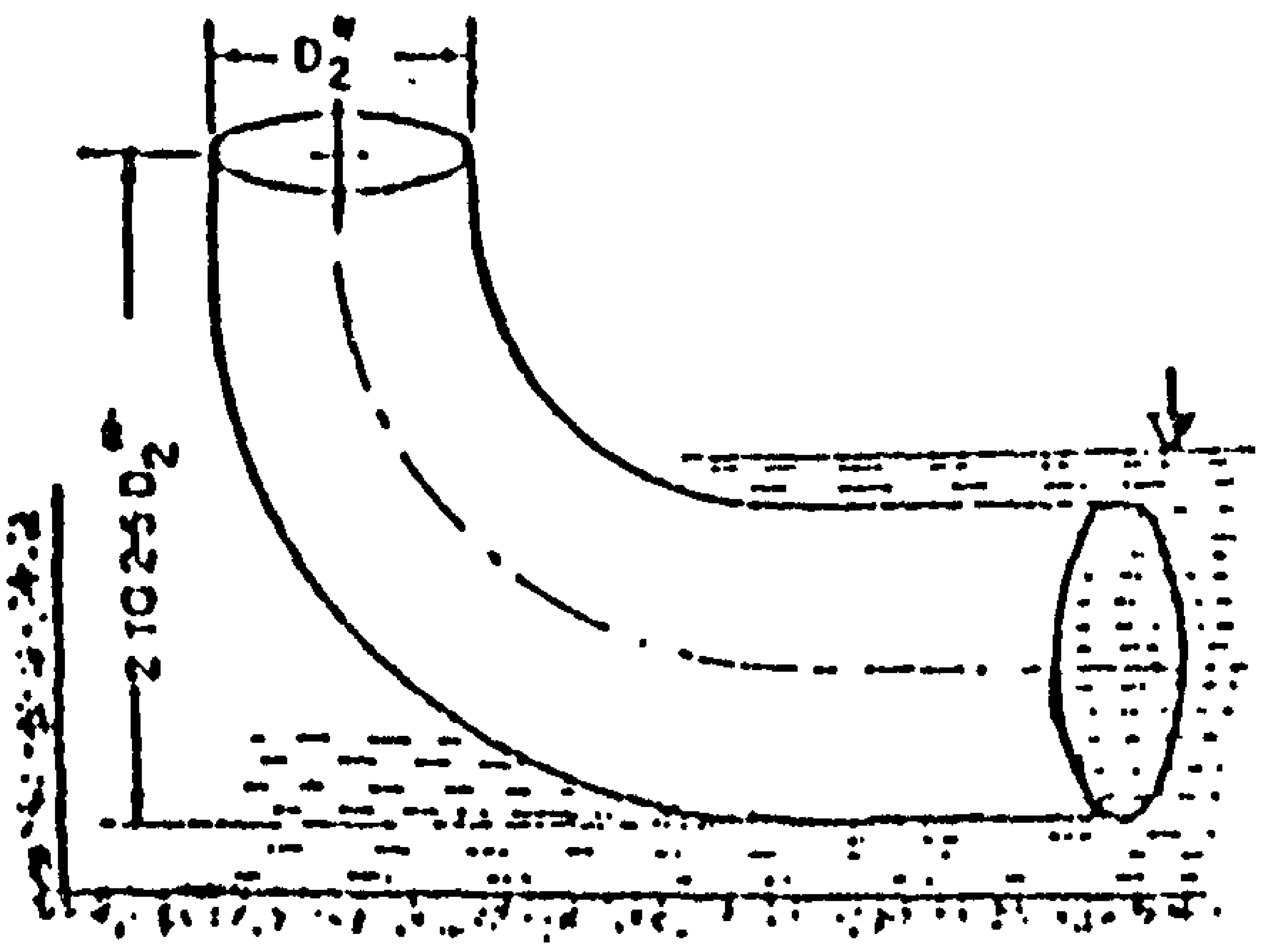



Figure 2.12 a - Simple Elbow Tube

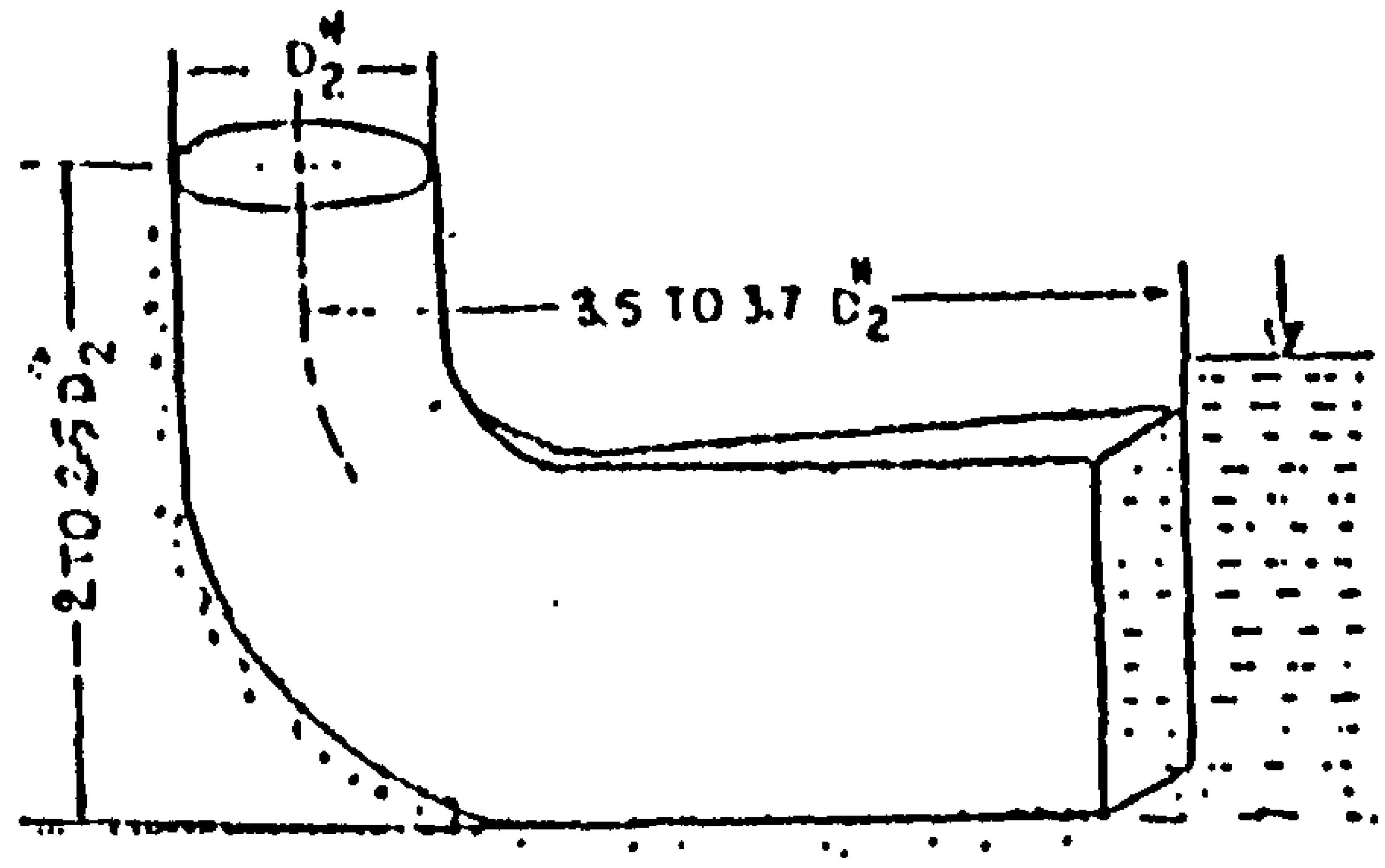



Figure 2.12 b - Elbow tube having circular cross - section at inlet but Rectangular at outlet.

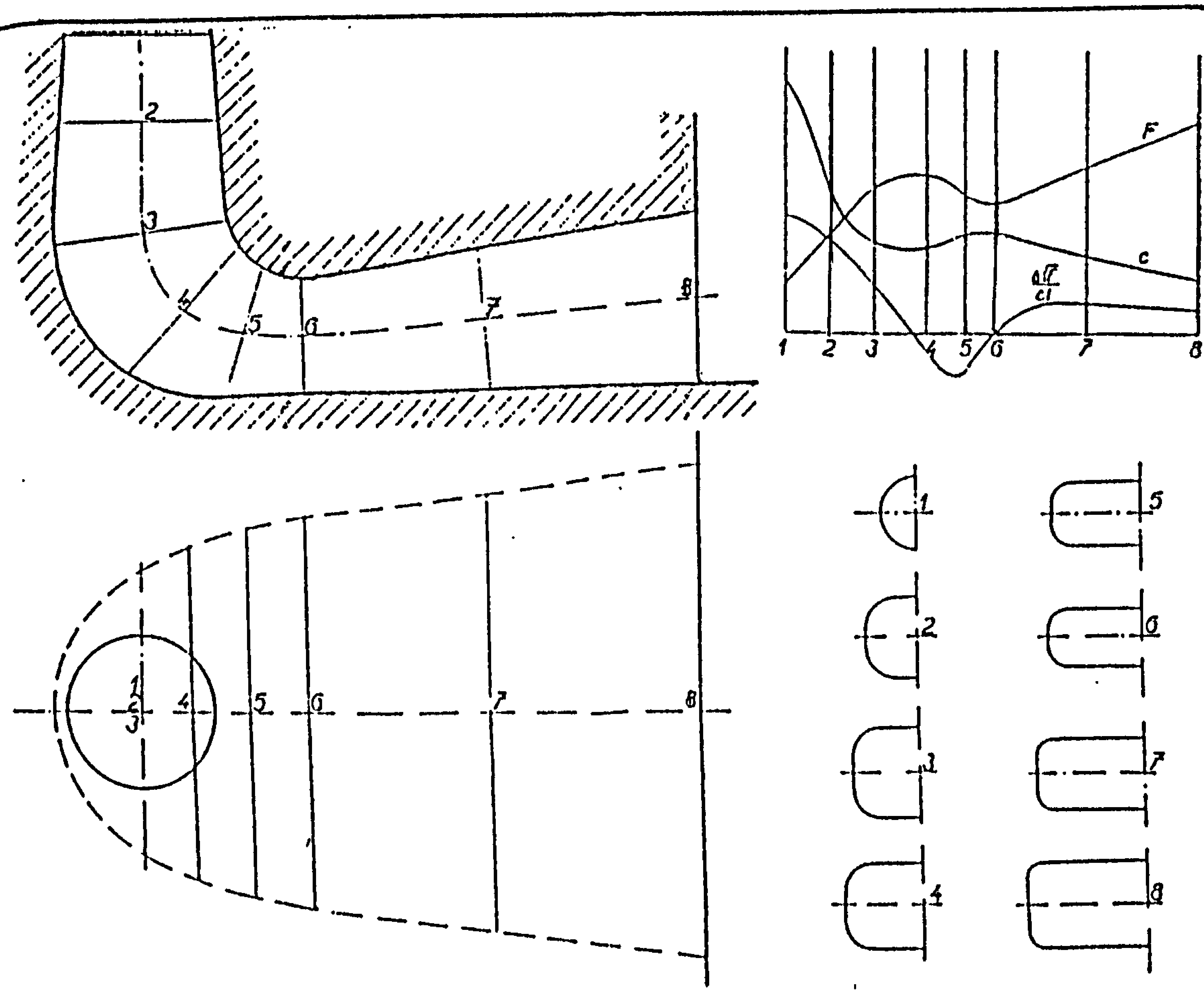



Figure 2.13

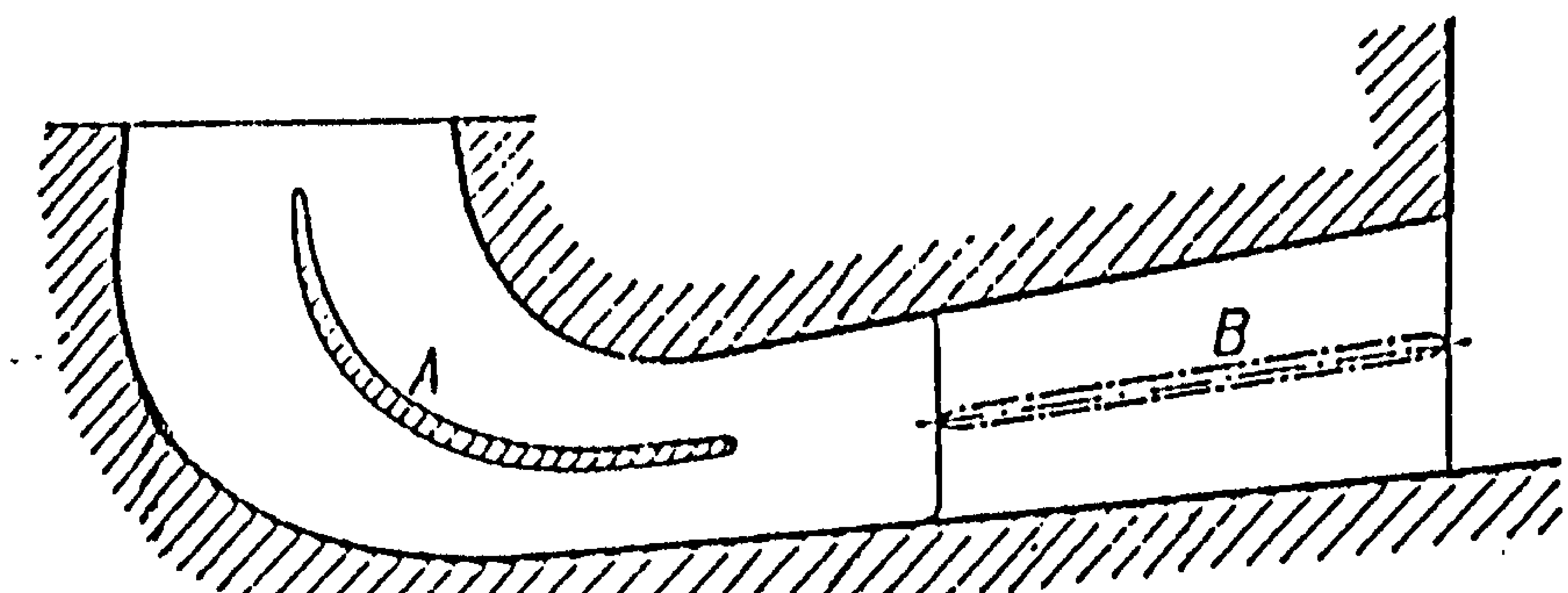



Figure 2.14

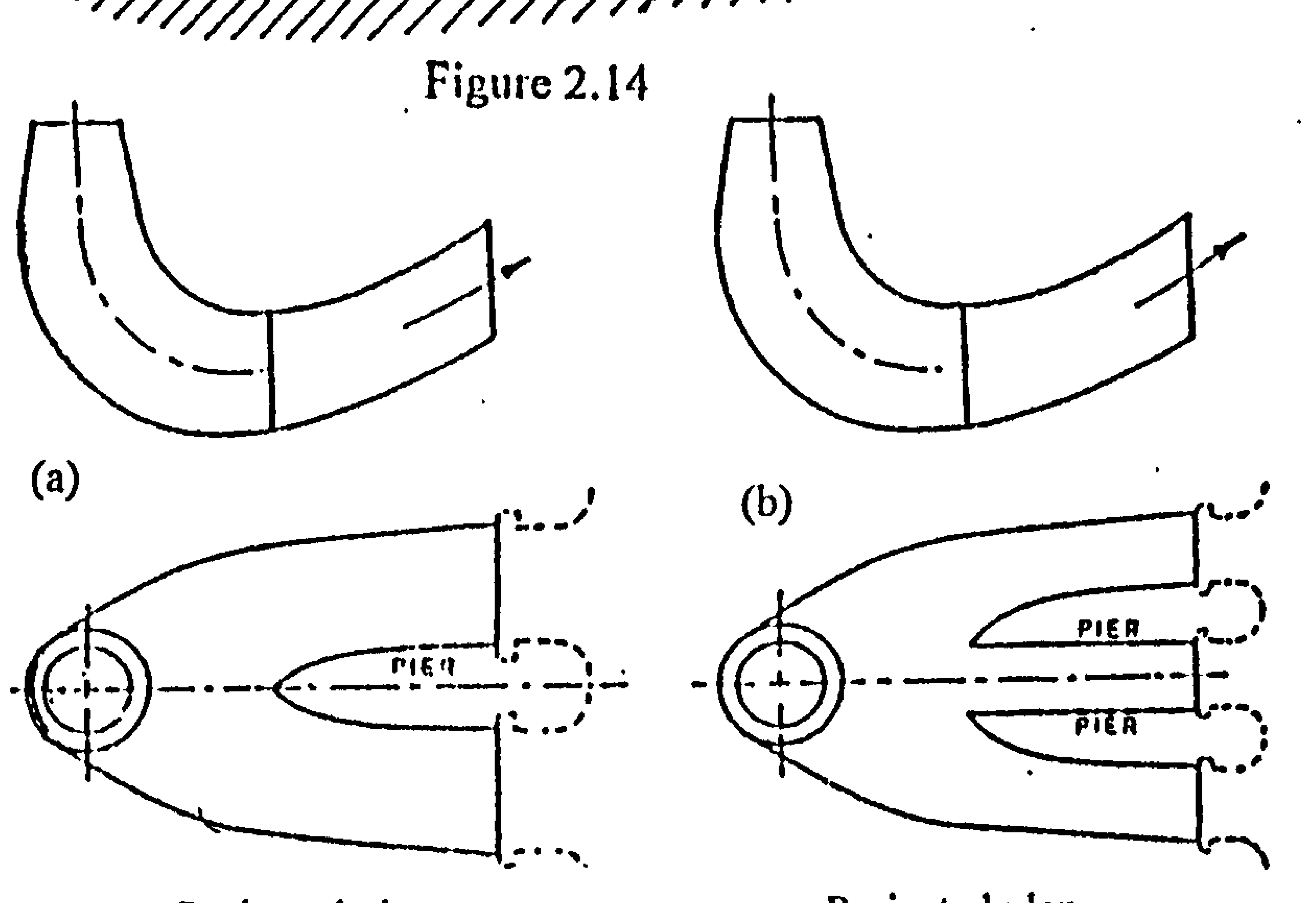



Figure 2.15 - Elbow type tube (a) Single pier type (b) Double pier type

## BLADES

As mentioned earlier, the design of blades of Agnew turbine was made as simple as possible . Considering relatively low efficiency of the turbine, it seemed to be appropriate to work on a blade design and manufacture one according to the scientific and technical data .

The following presents steps relating our work on blade design which would be discussed in depth in later sections : -

**I .** Theoretical introduction of propeller.

**II .** Circulation, lift and drag .

**III .** Application of airfoils to the blade design.

**I V .** Design procedure of the runner of the Agnew turbine.

### I . Theoretical Introduction of Propeller and Agnew Turbine

It is believed that propeller shaped runners have evolved from the Francis mixed flow runner, mainly suitable for low heads and relatively moderate to high flow rates . The difference between propeller turbine and Agnew turbine is in the adjustability of blades of Agnew turbine . By adjusting the blade settings, the problem of variations in flow is easily circumvented .

Generally, the number of blades on propeller turbines is 3 to 6 and in exceptional cases it may be as many as 8 to 10 . With regard to the number of blades, their spacing in relation to their length is relatively wide . Again with regard to small number of blades it may be deduced that the driving force of the water is naturally divided between a small number of blades, therefore there would be a great difference between the pressures on the pressure side of the blade and on its suction side. This would result in different fluid velocity on either side of the blade, with higher velocity being encountered on the suction side. Needless to mention that the fluid velocity difference as a consequence of pressure difference is in accordance to Bernoulli's equation . Since the turbine operates in an entirely closed duct from inlet to tail race. However these differences tend to increase as the number of blades decrease. Therefore, velocities (relative & absolute) of water leaving the runner would not be the same even within the range of two successive blades. It is, however, due to these considerable differences that a three dimensional approach may be employed for designing the blade.

One of the major obstacles in the way of calculations would be the fact that most of the coefficients related to the blade profile are not determined . To circumvent this problem it would be advisable to select a suitable blade profile ,all the coefficients of which are known.

## II. Circulation & the lift force

Generally, any fluid flow which does not have a whirl component, would have a circulation equal to zero. As a matter of fact there is only one type of potential flow known as "potential whirl" which possesses circulation other than zero. (16)

According to kutta-Joukowski theorem, if a cylinder of infinite length and radius r is assumed to rotate in a stationary "medium" with an angular velocity of $\omega$ , it will generate a potential whirl whose circulation is given by : -

$$C = 2\,\pi\,r\,\omega \qquad (1)$$

Now if the same cylinder rotates with the same angular velocity in a medium flowing at a velocity V , then the circulation would be given by :

$$C = \frac{F_L}{LV\rho}$$

Where :

$C$ = Circulation in flowing medium
$F_L$ = Lift force acting upon the cylinder
$\rho$ = Fluid density
$L$ = Lenght (span) of the airfoil

The above may be re-written as : -

$$F_L = \rho L V C \qquad (3)$$

There is another expression which yields the lift force and that is :

$$F_L = 1/2 \; (C_L V^2 S) = 1/2 \; (C_L V^2 L 1) \qquad (4)$$

Where :

$C_L$ = Coefficient of lift
$S$ = Airfoil supporting surface
$1$ = Depth of the airfoil

## III. Circulation and the blade design

Consider runner and its blade as shown in figure (2.16)

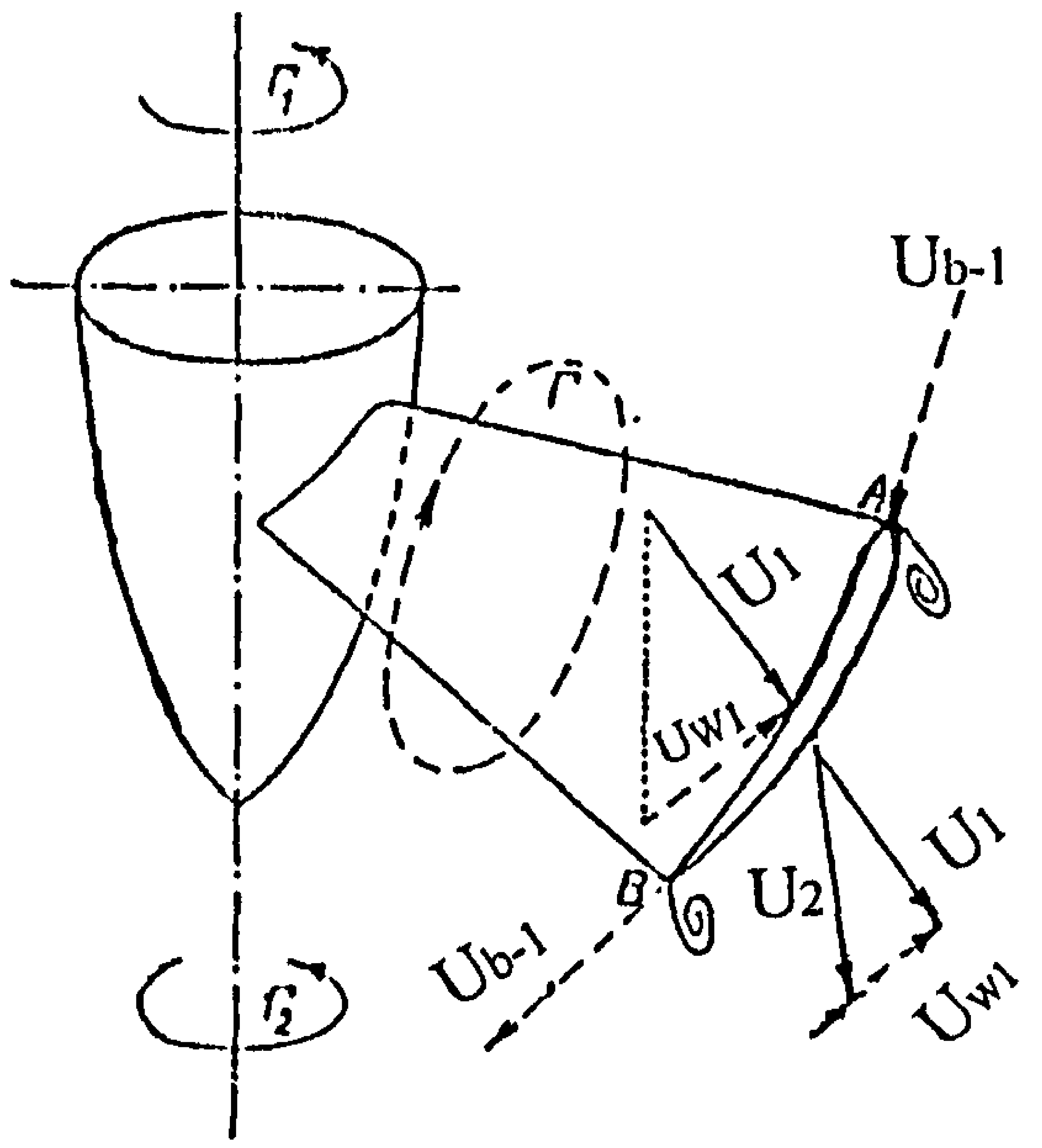



**Figure 2.16**

Water approaches the blade with velocity $U_1$, having a whirl component of $U_w$, with axis of whirl being the same as turbine s axis. Circulation generated at the face of the blade would be equal to :

$$C_{L1} = 2\,\pi\, r\, U_{w1} \qquad (5)$$

Also, there would be circulation at the back of the blade given by :

$$C_{L2} = 2\,\pi\, r\, U_{w2} \qquad (6)$$

In this case the induced drag may be neglected.

From expressions (5) and (6), the circulation around a blade may be calculated as follows :

$$C = \frac{2\,\pi\, r\,(U_{w1} - U_{w2})}{Z} \qquad (7)$$

Where :

Z = Number of blades

Let : $T = \dfrac{2\,\pi\, r}{Z}$ = Spacing of the blades

Then expression (7) becomes :

$$C = T\ (U_{w1} - U_{w2}) \qquad (8)$$

On the other hand, for axial flow hydro turbines the following equation holds :

$$U_b = \ (U_{w1} - U_{w2}) = \eta_h\, g\, H$$

Where :

$U_b$ = Peripheral velocity (blade velocity)
$\eta_h$ = Hydraulic efficiency
g = Acceleration due to gravity
H = Net head

In order to ascertain the lift force due to circulation around the blade, determination of the relative velocity of water ($W_u$) would be essential.Referring to figure (2.17) , and assuming an unlimited area as indicated, the force on the blade from the momentum equation maybe determined.

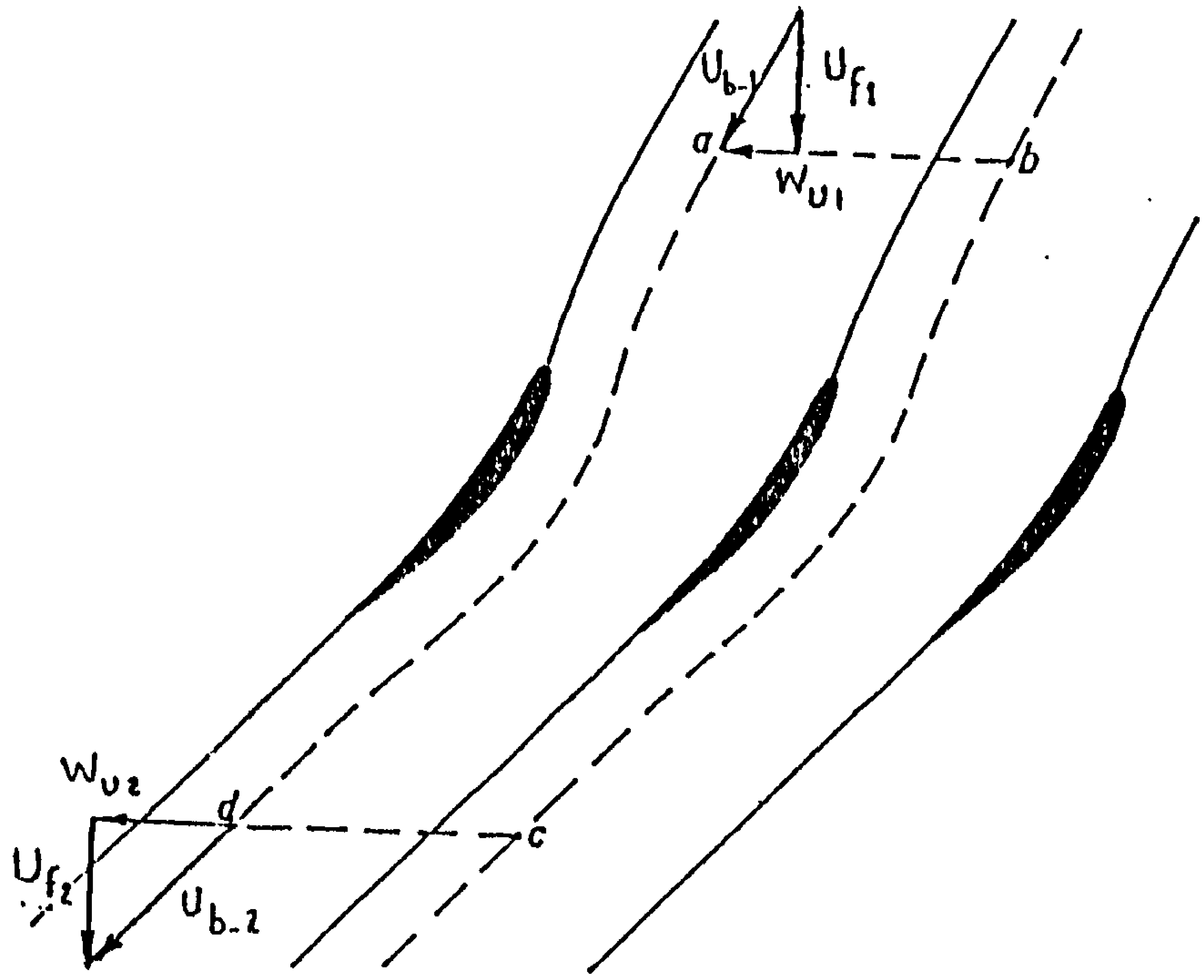



**Figure 2.17**

As shown on figure (2.17) water approaching the blades has a relative velocity of $w_{u1}$ and when leaving the runner its velocity would be equal to $W_{u2}$ The mass per second of this flow would be given by $T\rho U_f$, where $U_f$ is the flow or meridional component of the velocity.Therefore the horizontal component of the force acting upon the part of the blade of the radial length equal to one is given by:

$$P_{h1} = T\rho\, U_f\,(W_{u1} - W_{u2}) \qquad (9)$$

Also in the vertical direction the same part of the blade is acted up on, considering the fact that the velocity $U_f$ does not change its magnitude, then the vertical component of the force would be given by :

$$P_{v1} = T\,(P_1 - P_2) \qquad (10)$$

Where :

$P_1 - P_2$ = Over pressure of the runner

Since in propeller turbines $U_{b1} = U_{b2}$ and $U_{f1} = U_{f2}$, with $h_z$ representing the head loss within the runner, then the Bernoulli equation for relative motion would be:

$$\frac{P_1 - P_2}{\rho g} = \frac{U^2_{(b-2)} - U^2_{(b-1)}}{2g} + h_z = \frac{W^2_{u2} - W^2_{u1}}{2g} + h_z \qquad (11\text{-a})$$

where :

$U_{b-1}$ = Blade velocity relative to inlet.
$U_{b-2}$ = Blade velocity relative to outlet.

Also;

$$\frac{P_1 - P_2}{\rho g} = \frac{W^2_{u2} - W^2_{u1}}{2g} \left( 1 + \frac{2g\, h_z}{W^2_{u2} - W^2_{u1}} \right) \qquad (11\text{-b})$$

From expressions (9), (10) and (11) the resultant force of the two components ( i.e. $P_{h1}$ and $P_{v1}$ ), $P_{z1}$ may be found:

$$F_{L1} = T\ (W_{u2} - W_{u1}) \sqrt{U_f^2 + \left[ \frac{W_{u2} + W_{u1}}{2g} \left( 1 + \frac{2g\, h_z}{W^2_{u2} - W^2_{u1}} \right) \right]} \qquad (12)$$

But since $C = T\ (W_{u2} - W_{u1})$ and the expression under the radical sign dimensionally is equivalent to velocity (to be represented by $W_\infty$) then expression (12) would become:

$$F_{L1} = \eta\, C\, W_\infty$$

Taking into account the full width of the blade (L) then:

$$F_L = \eta\, C\, W_\infty\, L \qquad (13)$$

Assuming a loss free flow through the runner (i.e. $h_z=0$) then a velocity diagram for inlet and outlet of the runner may be constructed..From this diagram values for $W_\infty$ and its inclination $\beta_\infty$ may be obtained (figure 2.18).

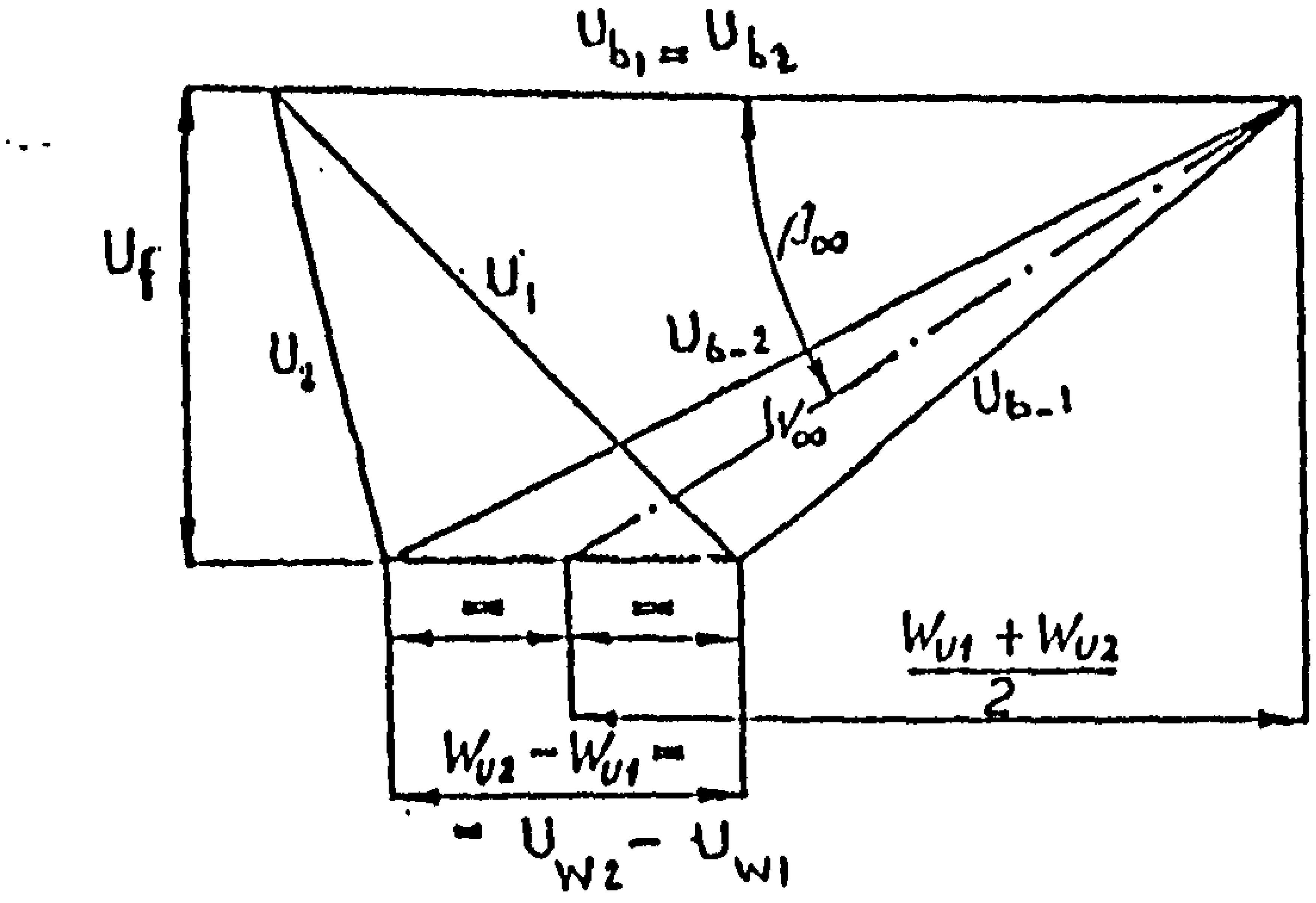



**Figure 2.18**

Facing the reality, $h_z$ actually exists and its existence, according to expression (13), would cause an increase of $g\, h_z / (W_{u2} - W_{u1})$ to the horizontal component of $W_\infty$. The vertical component of $W_\infty$

would, however, remain unchanged since the flow component of the velocity through the runner ($U_f$) is assumed constant. Changes in $W_\infty$ would result in $W'_\infty$ and $\beta'_\infty$ . (figure 27)

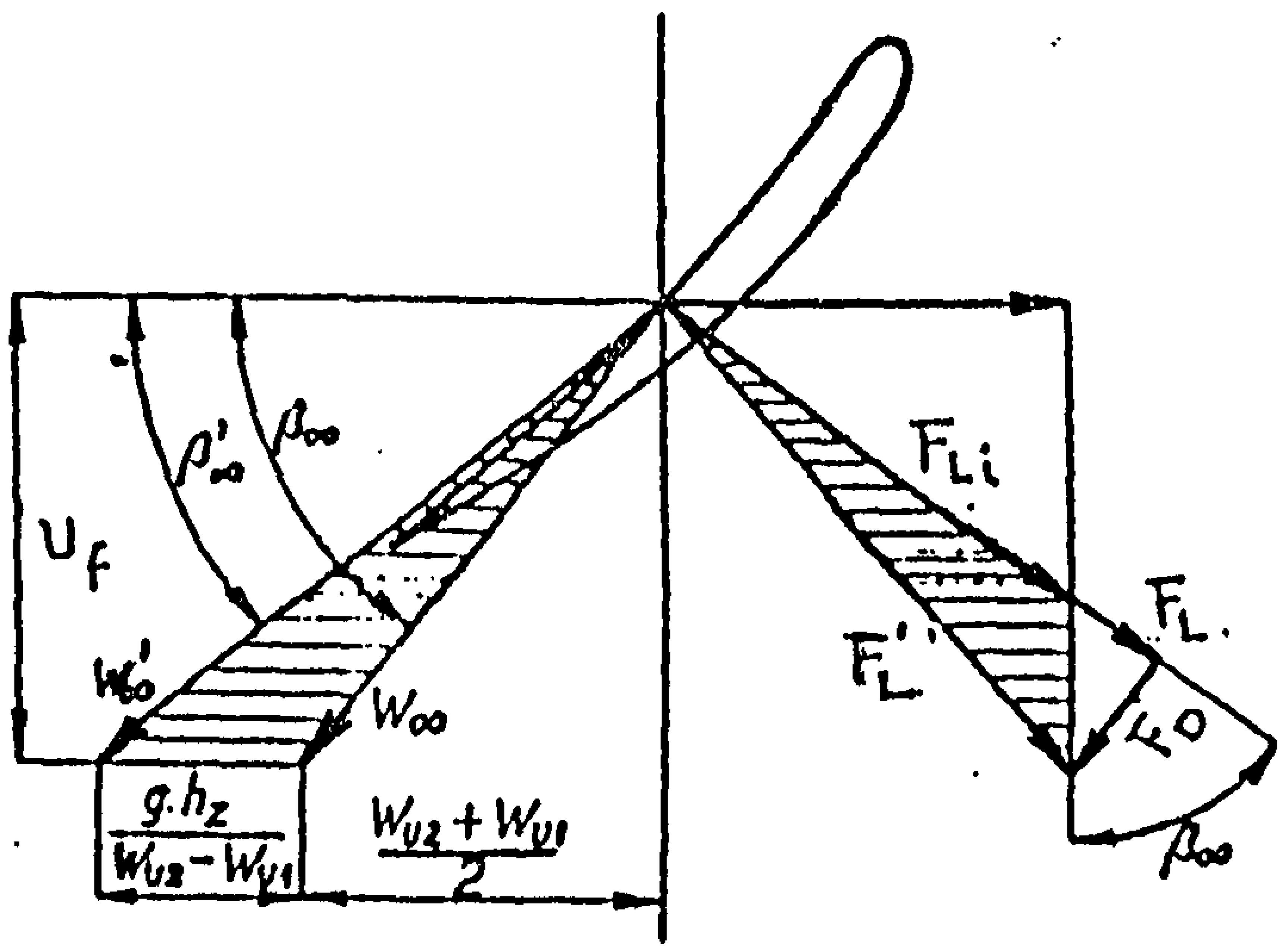



**Figure 2.19**

Considering the fact that the resultant force is always perpendicular to its respective velocity direction (i.e $F_L \perp W_\infty$ and $F_L \perp W'_\infty$) then

$$F'_L = \rho \, L \, C \, W'_\infty \qquad (14)$$

From figure (2.19) it may also be observed that $F'_L$ is the resultant of lift force $F_L$ and drag force $F_D$ components. Using similarity of the two hatched triangles, in figure (2.19), the following will hold :-

$$\frac{F'_L}{F_L - \dfrac{F_D}{\tan(\beta_\infty)}} = \frac{W'_\infty}{W_\infty} \qquad (15)$$

Since

$$F_L = 1/2\ (C_L V^2 S) = 1/2\ (C_L V^2 L\ 1)$$

and

$$F_D = 1/2\ (C_D W^2{}_\infty L\ 1)$$

Then

$$F_L - \frac{F_D}{\tan(\beta_\infty)} = ½\,\rho\, C_L L\, 1\, V^2 - ½\,\rho\, C_D L\, 1\ \frac{W^2{}_\infty}{\tan(\beta_\infty)} \quad (16)$$

and From expressions (14) , (15) and ( 16 ):-

$$\rho L C W'_\infty W_\infty = ½\,\rho\, 1\, C_L W'_\infty W^2{}_\infty - ½\,\rho\, C_D L\, 1\ \frac{W^2{}_\infty}{\tan(\beta_\infty)} W'_\infty$$

or

$$C = ½\ \left(C_L - \frac{C_D}{\tan(\beta'_\infty)}\right) 1\, W_\infty \qquad (17)$$

From expression (17) it may be deduced that, for the above calculations for C, a knowledge of $W'_\infty$ and $\beta'_\infty$ is not required. Values of $W_\infty$ and $\beta_\infty$ may be obtained from the velocity diagram (figure 2.18). To relate $W_\infty$ and $\beta_\infty$ to the whirl velocity expressions (8) and (17) may be used, so that:-

$$T\ (U_{w1} - U_{w2}) = ½\ \left(C_L - \frac{C_D}{\tan(\beta_\infty)}\right) 1\, W_\infty$$

whence

$$C_L - \frac{C_D}{\tan(\beta_\infty)} = 2\ \frac{U_{w1} - U_{w2}}{W_\infty} \cdot \frac{T}{1} = \frac{2C}{1\, W_\infty}$$

Applying specific velocities in the above:-

$$C_L - \frac{C_D}{\tan(\beta_\infty)} = 2\,\frac{U_{w1} - U_{w2}}{W_\infty} \cdot \frac{T}{1} \qquad (18)$$

But Euler equation maintains that:

$$U_b = (U_{w1} - U_{w2}) = \eta_h / 2$$

Using Euler equation in expression (18):

$$\eta_h = U\,W_\infty \frac{1}{T} \quad \left(C_L - \frac{C_D}{\tan(\beta_\infty)}\right) \qquad (19)$$

Where U and $W_\infty$ are specific velocity values.

From the above it may be concluded that in order to maintain a certain value of hydraulic efficiency, any increase in the value of U or $W_\infty$ should be met by a reduction in the length of the blade1 , or reduction in the number of blades , causing an increase in the value of T.

Now expression (19) can be used for determining a suitable airfoil for designing a propeller blade. Bearing in mind that corrections should be made to the values of $C_L$ and $C_D$, ascertained at laboratories . This is due to the fact that these values are related to individual airfoils and not airfoils of infinite span as it is in expression (19).

Design procedure of blade according to the properties of airfoil in accordance with the N.A.C.A. data is presented in appendix **( II )** .

## DESIGN PROCEDURE

According to the data presented upto now,and the properties of airfoil mentioned in appendix ( I ) , an attempt to design the blade of the turbine, may be made, through the following procedure. This procedure, however, has been prepared according to the data available on Agnew turbine, for other cases more general form of this procedure may be used.

**I** .According to the available data , the following are determined:

1. Blade diameter = D
2. Hub diameter = d
3. Net head = H
4. Desired hydraulic efficiency = $\eta_h$
5. Approximate output angular velocity = $\omega$
6. Approximate flowrate = Q
7. Number of blades = Z

**II** . In the space of the runner a number of cylindrical sections of equal areas are determined as shown in fig (2-20-Section I-V) .

**III** . Having diameters of blade (tip to tip) and the hub and the flow rate, it would then be possible to calculate the flow (meridional) component of the velocity, $U_f$ , which is constant through the runner (i.e. $U_{f1} = U_{f2}$) by the expression:

$$U_f = \frac{Q}{\text{Area of flow}}$$

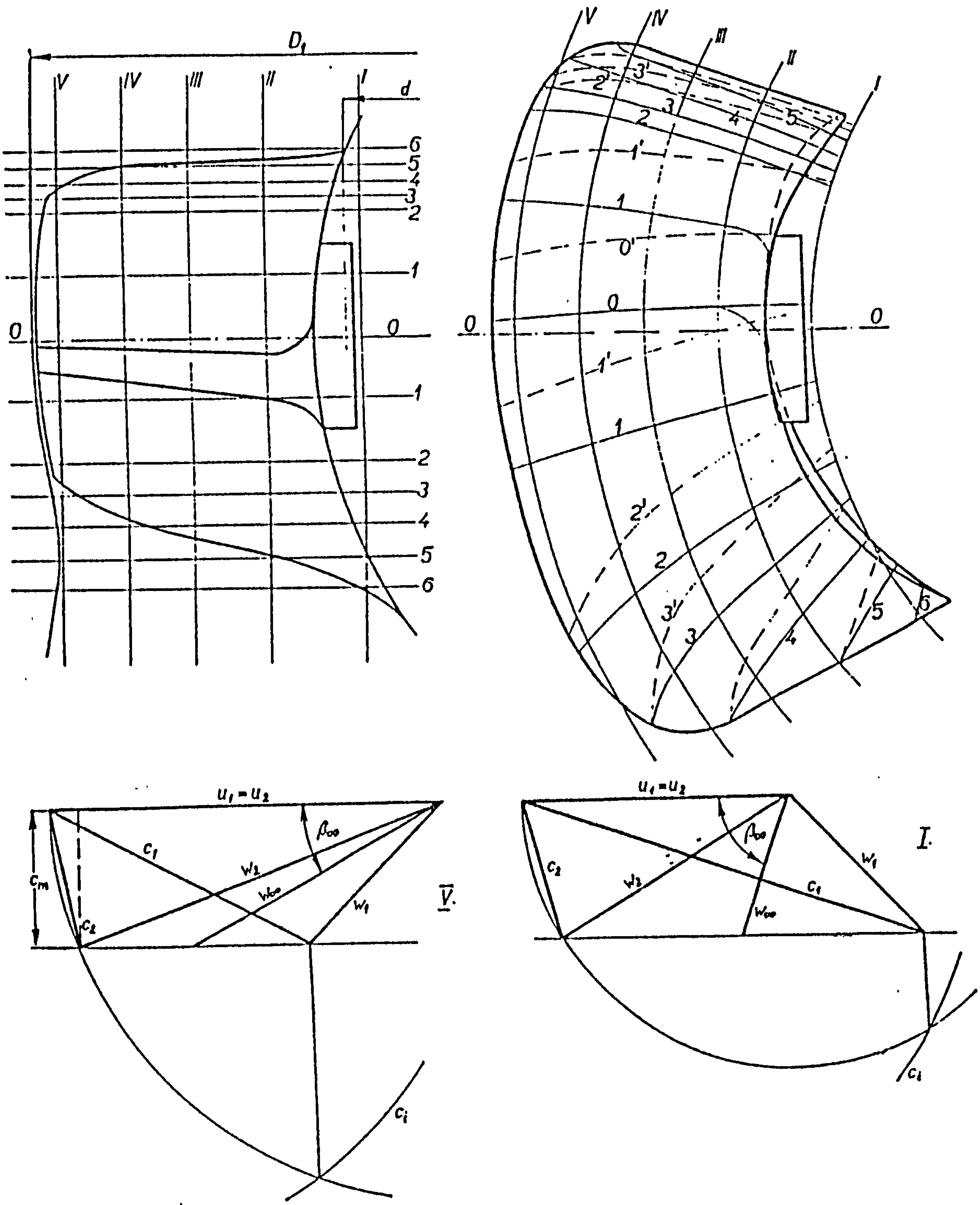



**Figure 2.20**

Also, having the angular velocity, $\omega$ , the blade velocity at any radius may be calculated by, $U_b = ½ D \omega$. Here, also for a propeller turbine, $U_{b1} = U_{b2}$.

**IV.** As mentioned before one of the conditions for obtaining maximum efficiency, is to have no component of whirl at the runner's outlet. (i.e. $U_{w2} = O$). Also, $U_{w1}$ may be obtained by the efficiency expression:

$$g.H.\eta_h = U_b ( U_{w1} - U_{w2} )$$

Now, having the above data on different velocity components, the velocity diagrams may be constructed. However, there may be cases, that, the whirl component of the outflow velocity is not equal to zero. In such cases or any other case where enough data for construction of velocity triangles is not available, application of Braun's diagrams is recommended (16). By using this method through calculation of the indicated velocity from the following expression;

$$Ui^2 = U^2 + \eta_h . g.H$$

Where :

$U_i$ = specific indicated velocity
U = specific velocity of flow in turbine

And then by construction of Braun's diagram, the vertical axis on which the vertex of inlet triangle should be located is determined. By constructing this diagram the rest of the data required for further steps in blade design is obtained. However, it was mentioned earlier on, that because of special case of axial propeller turbines and available data, construction of Braun's diagram was not required.

Having sufficient data, then construction of velocity diagrams may be proceeded. (figure 2.20 )

**V.** Referring to figures ( 2.18 ) and ( 2.19 ), values of $\beta_\infty$ and $W_\infty$ can be obtained. Then from expression (18)

i.e. $$\eta_h = U\, W_\infty \frac{1}{T} \left( C_{LL} - \frac{C_{DL}}{\tan(\beta_\infty)} \right) \qquad (23)$$

the value for $C_{LL} - C_{DL} / \tan(\beta_\infty)$ is calculated. To find this value, the chord length for the section in question would be required. Usually, a range of 0.9 to 1.05 is recommended for the ratio $l/T$ where, $T = \pi D/2$ , known as the blade pitch, and $l$ is the chord length ( 10 ). (Care should be taken to avoid confusion between the blade pitch and the airfoil thickness).

**VI.** Since the above values are related to an airfoil of infinite span in a lattice system, then we can apply expressions (20) and (21) in expression (23), then the following would be arrived at:

$$C_L\, J \tan(\beta_\infty) - C_D + \frac{C^2_L}{6\pi} = \frac{\eta_h}{U\, W_\infty (l/T)} \tan(\beta_\infty) \qquad (24)$$

The value of J may be ascertained from the paskuara diagrams (i.e. figures **II**.6 and **II** .7 , appendix **II** ). Also the lattice angle may be calculated by :

$$\beta = 90° - \beta_\infty + \alpha$$

Where $\alpha$ is the angle of attack, to be selected at this stage to determine an approximate value for the lattice angle, $\beta$ . Accuracy of the value of $\alpha$ is not of great importance at this stage, since the diagrams for different angles of attack in figure ( **II.6** ), do not show significant differences within the range of generally used lattice angles. Therefore the diagram in figure ( **II.7** ) (i.e. $\alpha = 0$) may also be used for other angles of attack (1). Next step would involve selection of a profile section according to N.A.C.A. data. Having ratios m/1, L/1, and t/1 for the selected profile,where:

m = Maximum deflection of the central curve of the airfoil.

L = Distance of the maximum deflection of the central curve from the leading edge.

t = Maximum thickness of the airfoil.

$$C_L = \frac{dC_L}{d\alpha}(\alpha - \alpha_0)$$

$$C_D = C_{DV} + \frac{dC_D}{dC^2_L} C^2_L$$

Values of $C_L$ and $C_D$ are then obtained in terms of $(\alpha - \alpha_0)$ substituting these values in expression (24) would yield a quadratic in terms of $(\alpha - \alpha_0)$.The follow ing represents this quadratic.

$$\left(\frac{dC_D}{dC^2_L} - 1\right)\left(\frac{dC_L}{d\alpha}\right)^2(\alpha - \alpha_0)^2 + \frac{dC_L}{d\alpha}(\alpha - \alpha_0)\,J\tan\beta_\infty - \left(C_{VD} + \frac{\eta_h}{UW_\infty\frac{1}{T}}\right)\tan\beta_\infty = 0$$

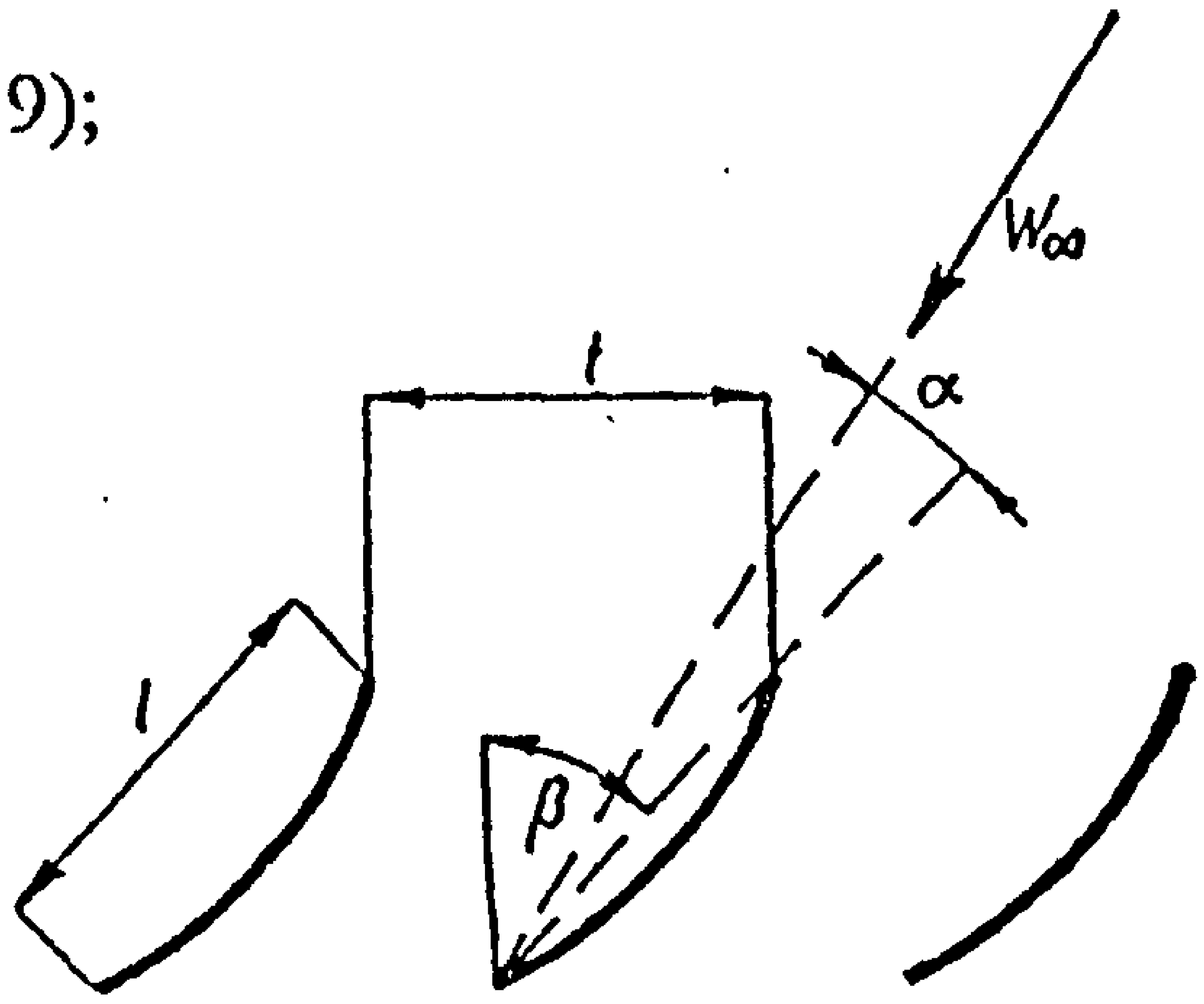
Solving this quadratic and having the value of $\alpha_0$ from figure ( **II.2** ) would result in the value of the angle of attack, $\alpha$ .

Now, applying $\alpha$ , to expression (19);

$$\alpha_\infty = \alpha - 57.3 \, \frac{C_L}{6\pi}$$

then through the expression

$$\beta = 90° - \beta_\infty + \alpha_\infty$$




The lattice angle, $\beta$, may be calculated. Now, having all the required data and the technique mentioned in appendix (II) the airfoil section may be constructed.

This procedure could be employed for all other cylindrical sections. Figure ( 2 .21 ) represents an airfoil section constructed by the above procedure.

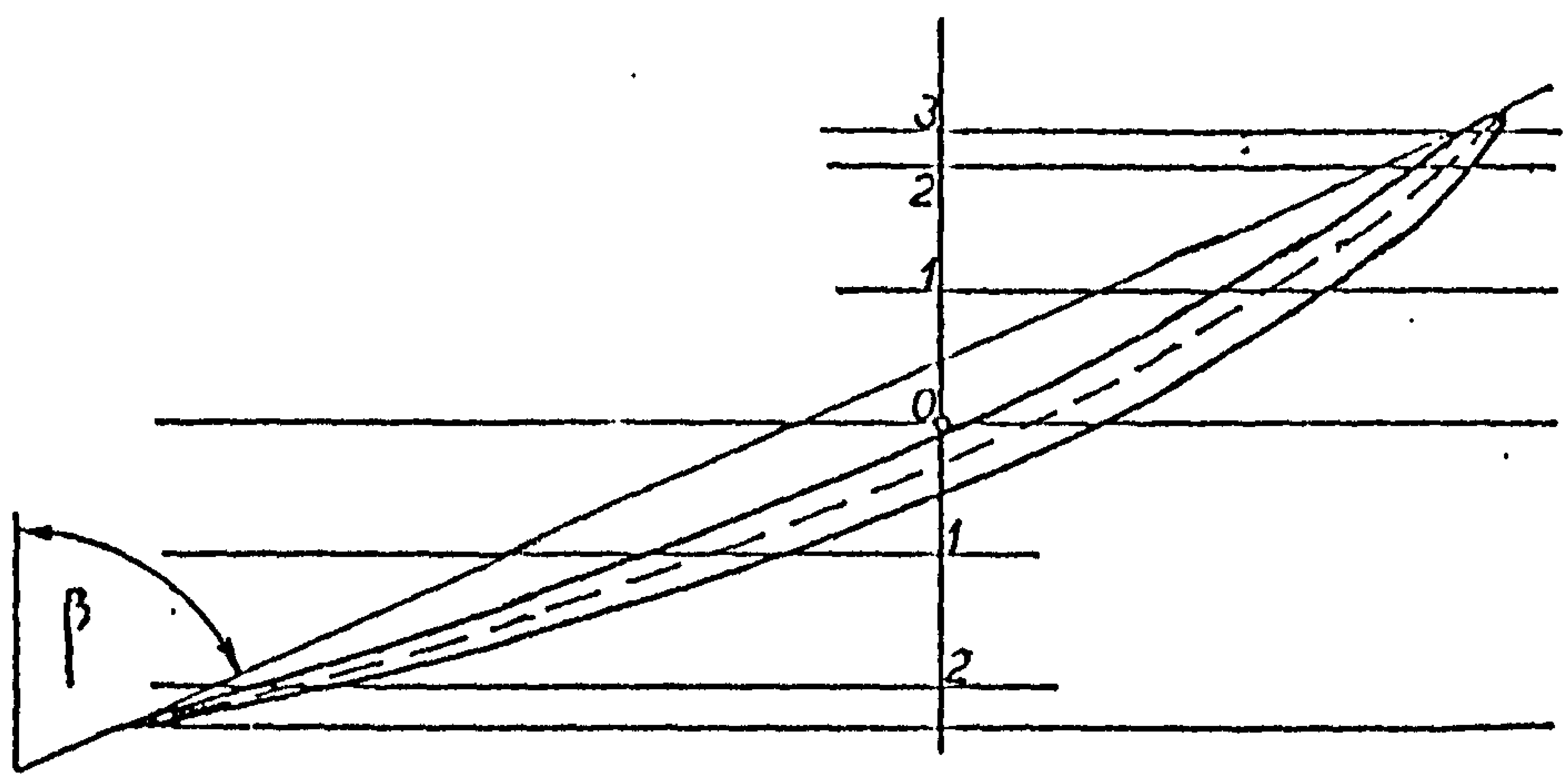



*Figure 2.21*

# CHAPTER

# ( III )

# APPARATUS

# Apparatus

For the purpose of testing the turbine a suitable test circuit was required. Since such a circuit was not available in Iran therefore it was decided to design a circuit. On the other hand, as the work was being carried out at I.R.O.S.T, (a reference research organization) therefore it was decided to prepare a laboratory for testing water turbines with a view of its extension to a turbo machinery laboratory.The lab. is actually situated at the Asr-E-Enghelab research complex at the western suburbs of the city of Tehran.Photos (3.1) and (3.2) show the outside and inside of the lab. respectively at the beginning of the work.

# Laboratory design

In general, there are two types of circuits for testing hydraulic machines.

i. Closed circuit

ii. Open circuit

According to availability of different resources, such as money, space, time etc., the second option was selected.

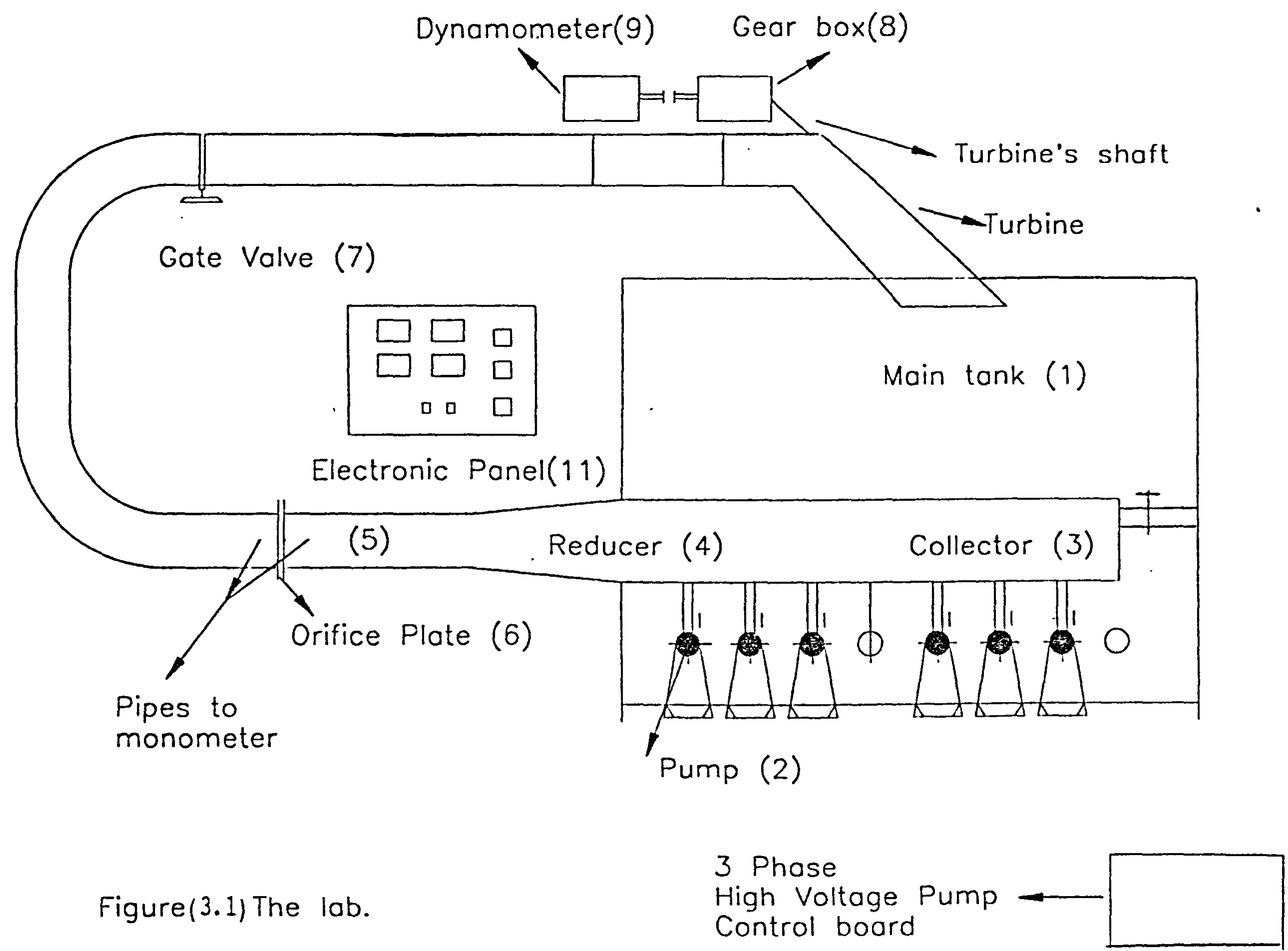
## Open circuit design

After completion of primary studies, a few alternative designs were prepared for the circuit. To finalize the design, a visit was made to the "turbo institut" in Slovenia (Report of this visit is presented in appendix III ).

During this visit consultations were made over the subject and later in Iran an open circuit was designed according to these consultations and according to the ISO standards. Figure (3.1) provides an overall sketch of the laboratory, and the following presents the details according to this figure. Photos at the end of this chapter represent different views of the laboratory.

**Photo (3.1)**

**Photo ( 3.2 )**




Figure(3.1) The lab.

## I. The reservoir

This consists of a (20 $m^3$) cubical open metal container. Photo (3.3) shows a view of the tank.Also photo (3.4) shows a view of the tank at the beginning of its manufacture.

**Photo (3.3)**

The internal volume of the tank has been divided into three sections by interconnected metal sheets to reduce the effects of shock waves of incoming water from turbine to the walls of the tank.Photo (3.5) represents a view of the shock absorbing metal sheets.Eight holes have been made on one of the side walls, close to the bottom of the tank for the purpose of installation of pumps.

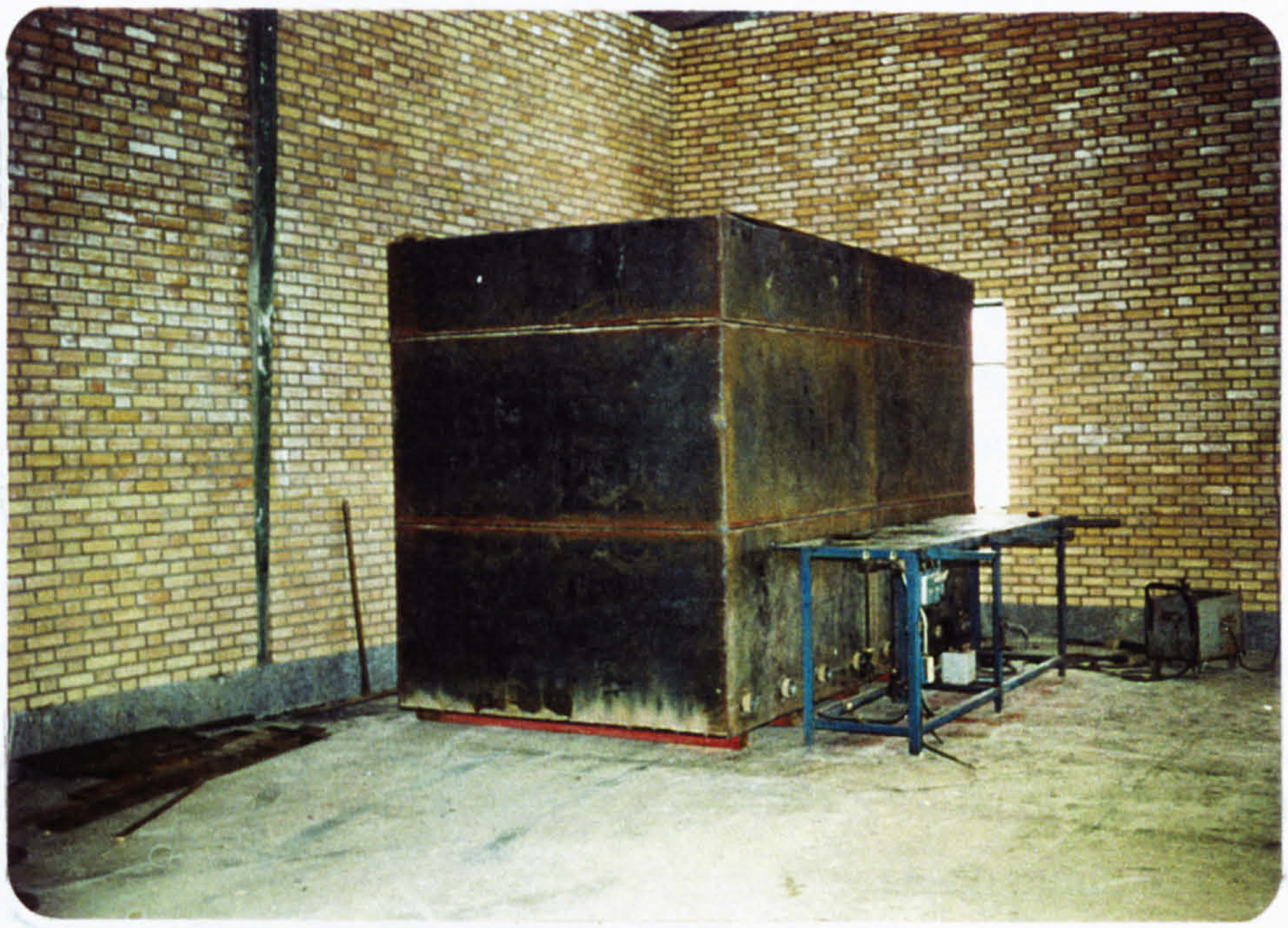


**Photo (3.4)**

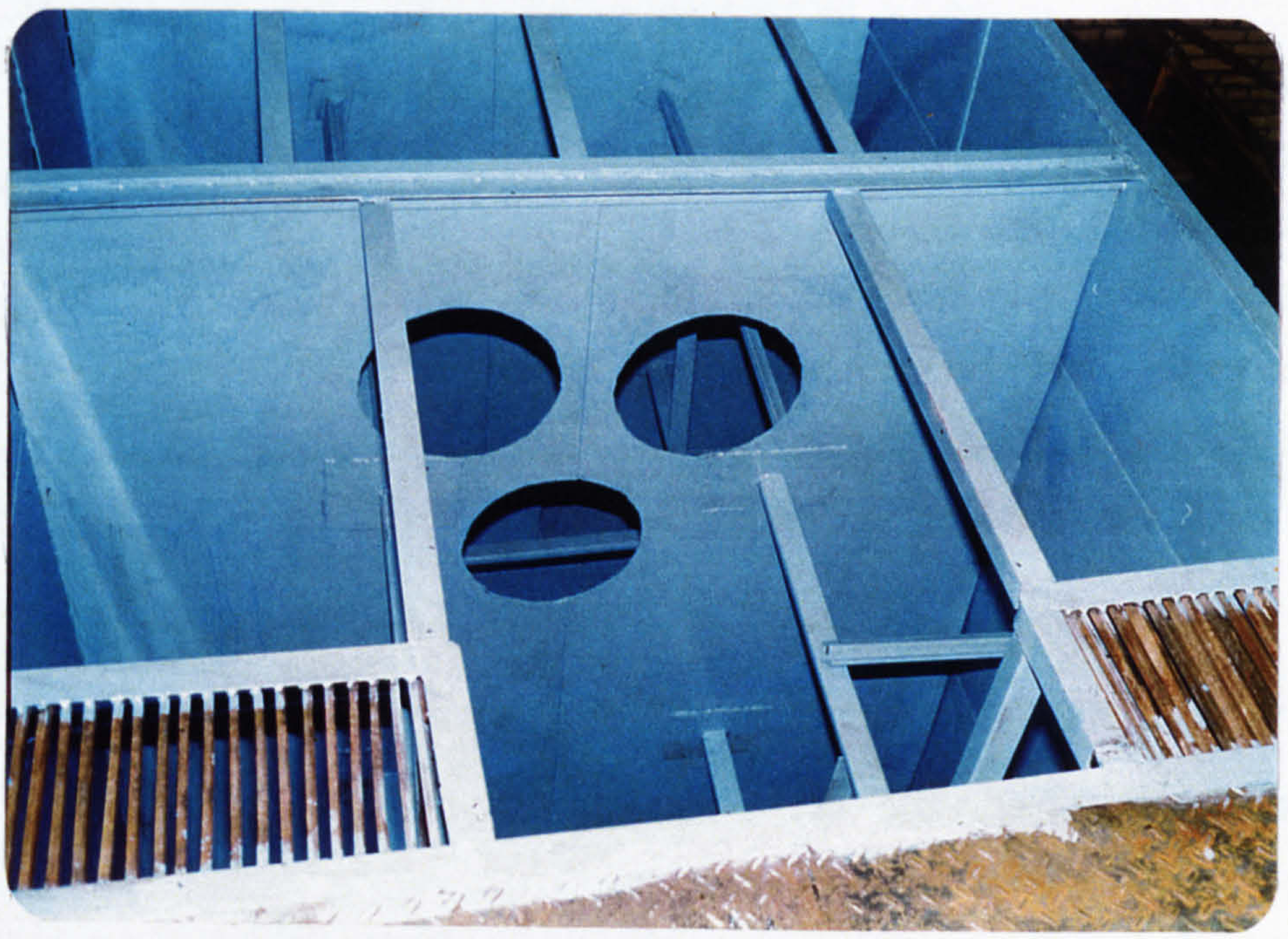


**Photo ( 3.5 )**

## II. Pumps

To produce the necessary heads and flow rates to the turbine, six pumps were installed, connected to the 11(Kw) electromotors.Photos (3.6) and (3.7) represent the installation process of the pumps .Also photo (3.8) shows all six pumps after installation. A high voltage control panel was designed and made for operation of the pumps.Photo (3.9) shows the control panel. However, since enough power for the operation of the electromotors was not available at the site, therefore, 250 meters of ( 200 Amps) power line was laid from the main power house to the site.

Gate valves were installed at inlet and outlet of each pump.Also, one way valves were installed at the out let side of each pump to avoid returning of water pumped by other operating pumps, back into the reservoir. Also shock absorbers were installed at the outlet side of the pumps to reduce the effects of vibration on the circuit.

## III. Collector

Collector, is a 4 meter long cylindrical volume, which actually connects the pumps to the circuit. Its diameter is twice larger than the diameter of the pipings. Water is first pumped into the collector through inlets connected to the pumps outlets; after which water runs into the pipings. The collector may be visualized on photo (3.8).A by pass outlet of 125( mm) diameter along with a gate valve are installed at the other end of the collector which can lead water back into the reservoir. This is normally used for smaller adjustments of flow rate and head.

## IV.Diffuser (Reducer)

This is of semi conical shape and is connected to the other end of the collector to provide a smooth reduction of diameter from collector to that of the pipings. In this way , the losses due to cross sectional changes would be mimimized.Photo (3.3) also shows the diffuser.

Photo (3.6)

Photo ( 3.7 )

Photo (3.8)

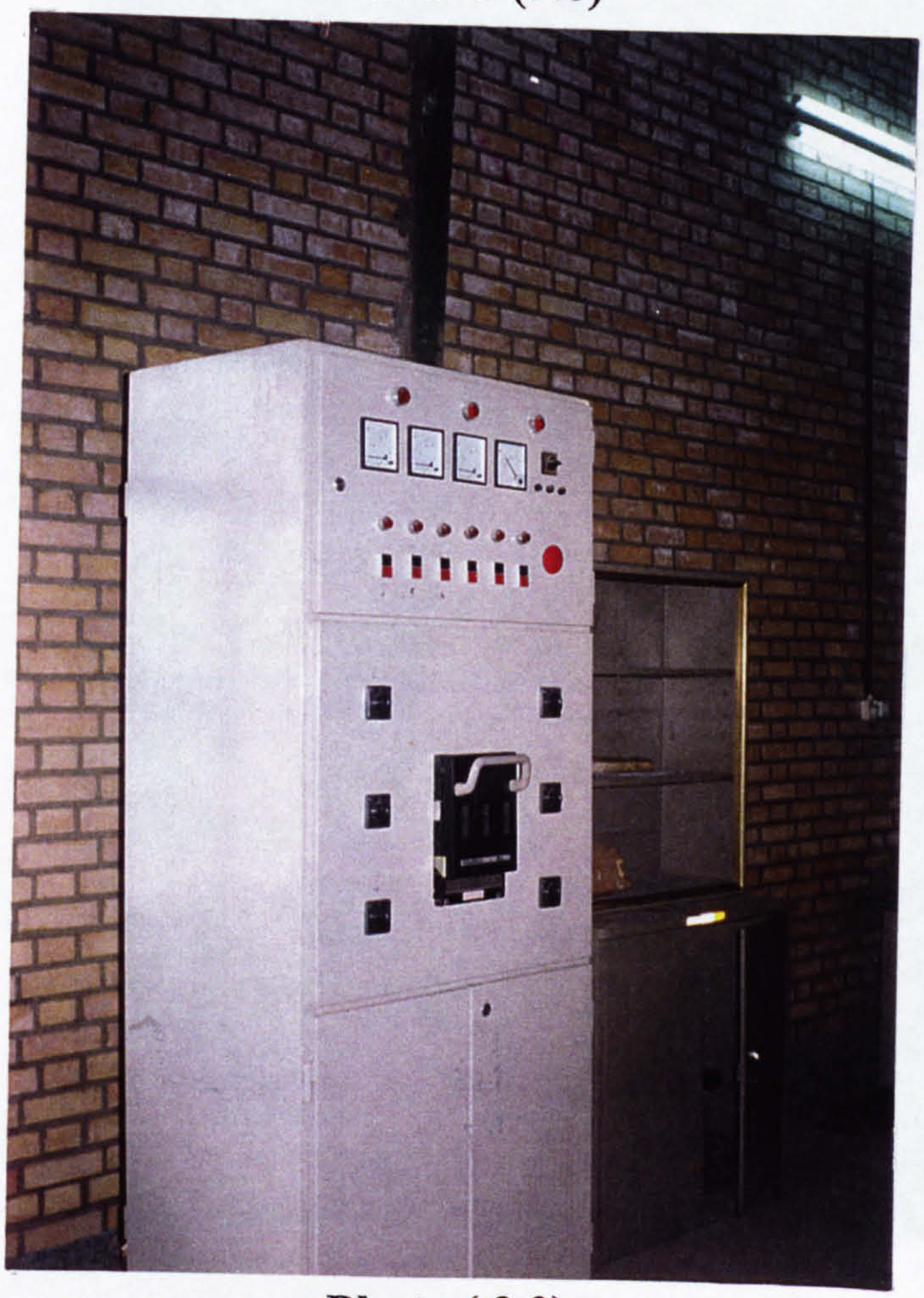


Photo ( 3.9)

## V.Pipings

The piping consists of 10 inches pipes all through from the diffuser into the turbine inlet.

## VI. Main gate valve

A gate valve of the same diameter of the pipings has been installed at the return brunch of the circuit, (Photo 3.10). This is used for introduction of major losses to the system to achieve required heads and flow rates.

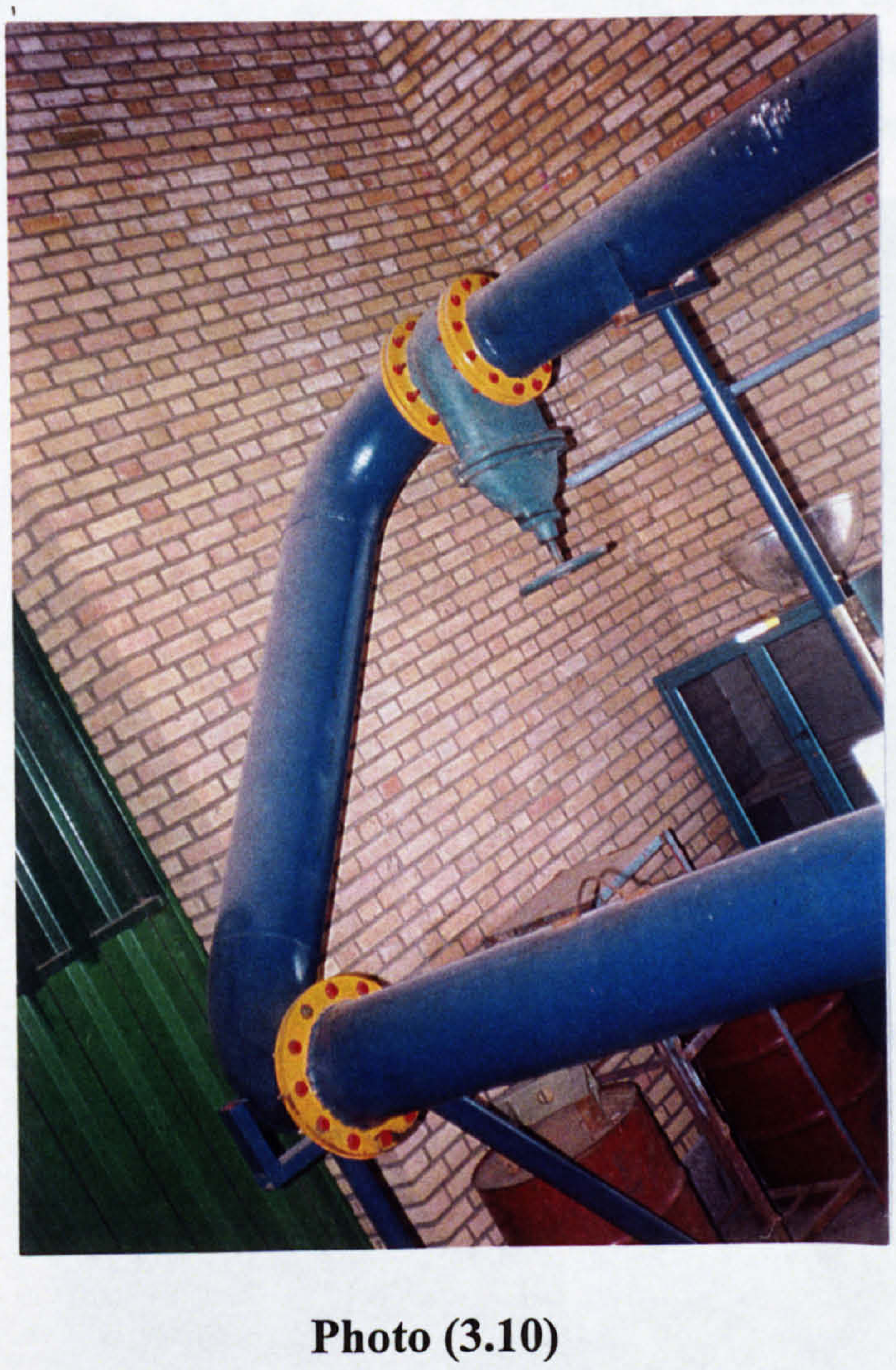


**Photo (3.10)**

# Circuit's measuring devices

## I. Orifice plate

For the purpose of measuring flow rate an orifice plate is used. These plates have been designed and manufactured according to ISO standards . These standards are provided in appendix (IV). Also, standards for proper location of installation of orifice plates are presented in the same appendix.

## II. Manometers

Altogether, there are three manometers installed at different locations for various purposes. These manometers are as follows:

**1**. Differential manometer, for obtaining pressure difference between two faces of the orifice plate. With the help of this reading, calculation of flow rate would be possible (11).Photo(3.11) shows one of these manometers.

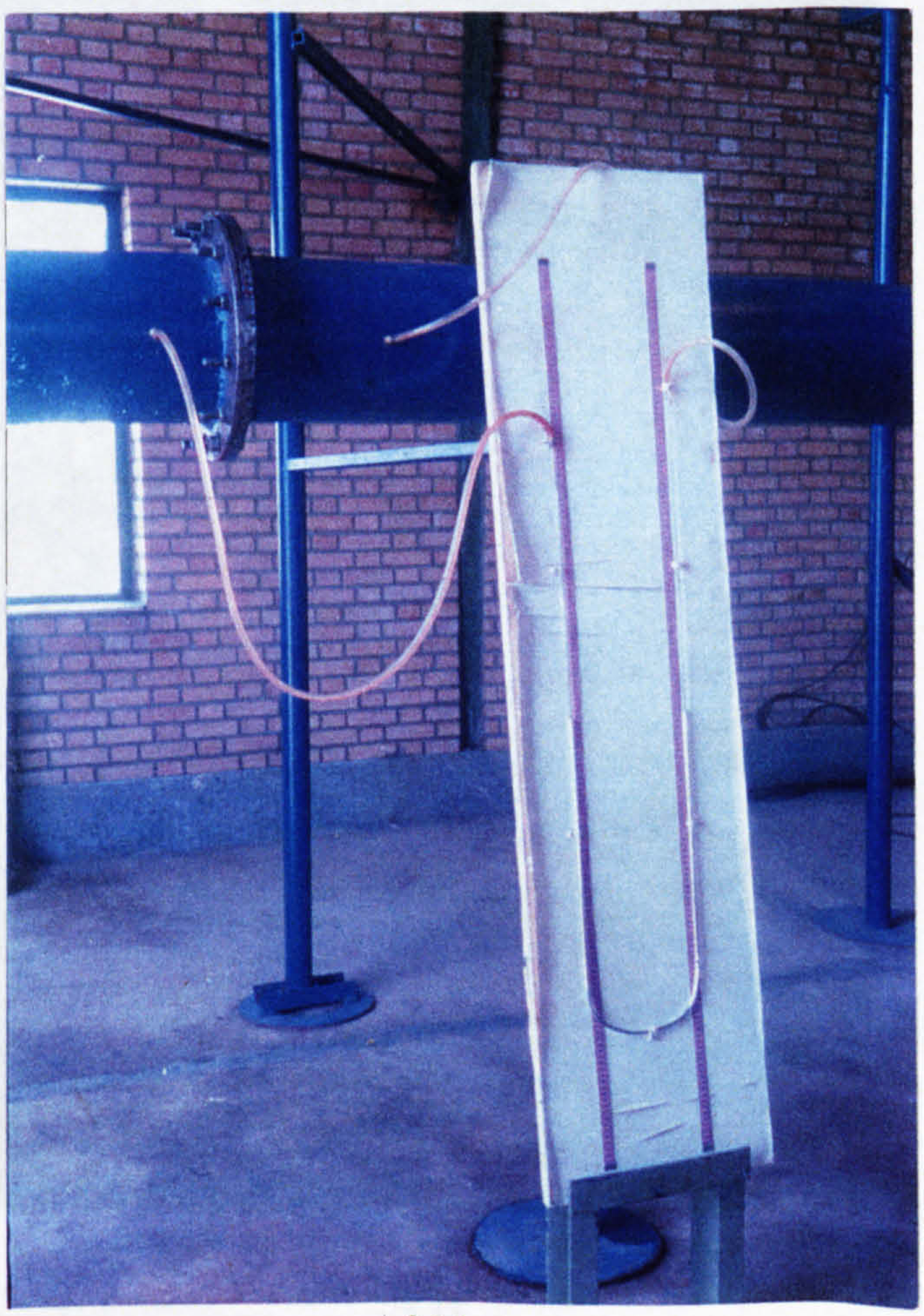


**Photo ( 3.11 )**

**2**. Another manometer is placed at the entrance of the turbine to read the actual head of the incoming water to the turbine.

**3**. The third manometer is placed at the entrance of the draft tube, in order to show the vacuum head generated by it. This reading may also be used to calculate the efficiency of the draft tube.

However, these manometers were later substituted by pressure sensors, so that, the pressure readings could be read directly on the electronic panel.

## III. Dynamometer

To measure the torque produced by the turbine shaft, a hydraulic dynamometer was employed.Photo (3.12) represents a view of the hydraulic dynamometer.

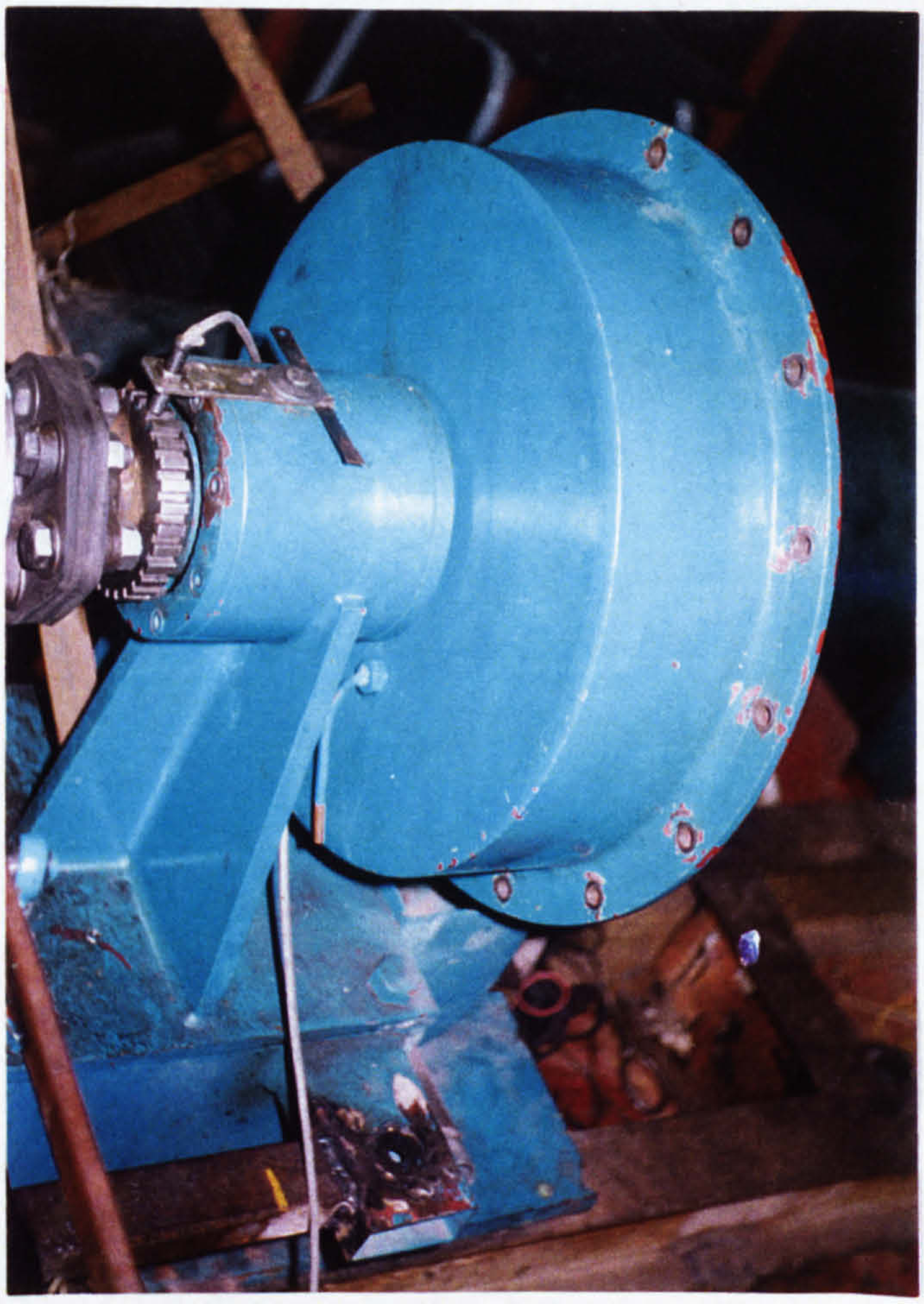


**Photo ( 3.12 )**

However, for avoiding the vibrations caused by the unbalanced rotors of the dynamometer, a Pronney dynamometer was designed and manufactured later.

Because of the nature of the hydraulic dynamometer, it was impossible to align it with the main turbine shaft. This is due to the fact that, the turbine shaft is subtended 45° to the line of horizon, and installation of the dynamometer at such an angle because of keeping the level of water inside the dynamometer at a desired level to obtain enough brake power, was impossible.To connect the two, angular motion of the turbine shaft was transferred via a pair of toothed wheels and a toothed belt to a 22.5°, one to one gear box. In this way the 45° motion of the turbine shaft was reduced to 22.5°. The remaining angle ( another 22.5° ) was recovered by employing a 22.5° universal joint; connected to the horizontal dynamometer input shaft.

**Pronney Dynamometer**

Basically Pronney dynamometer is a completely mechanical device and mainly uses friction forces to determine torque. The design was tried to be kept as simple as possible and the following presents details of the parts of the dynamometer.

*Casing*, which is a hollow cylindrical volume and closed at one end. At the center of the closed end a relatively wide roller bearing is installed and therefore the casing can freely rotate round the bearing and it would then be installed on the output shaft of the 45° gearbox. This shaft rests in the gear box in one end and is held by another bearing system at the other end.

*BRAKE SHOES* : Two brake shoes are hinged to the inner side of the casing. The braking mechanism operates by means of exertion of pressure on brake shoes by driving two screws through the casing.

Two springs pull the shoes back to avoid any chance of unwanted braking . Figure (3.2) represents a schematic front view of the dynamometer .

*Rotating drum* which is a solid cylindrical volume and is keyed to the main output shaft of the gear box and in fact rotates with it . Brake shoes are actually pressed against the drum and therefore the generated torque is transferred to the outer casing . A load cell which is connected to the wing of the dynamometer ( welded to the casing ) would sense the torque and show its amount on the electronic panel in terms of (N m ) . Photo (3. 13) and photo (3.14) show two different views of the Pronney dynamometer .

**<u>IV. Tacho meter</u>**

An electronic tachometer was connected to the dynamometer's shaft to read the angular motion of the turbine shaft. in r.p.m.

**<u>V. Electronic reading panel</u>**

On this panel readings of the manometers, torque and the angular velocity could be read.Figure (3.15) represents a front view of the panel.

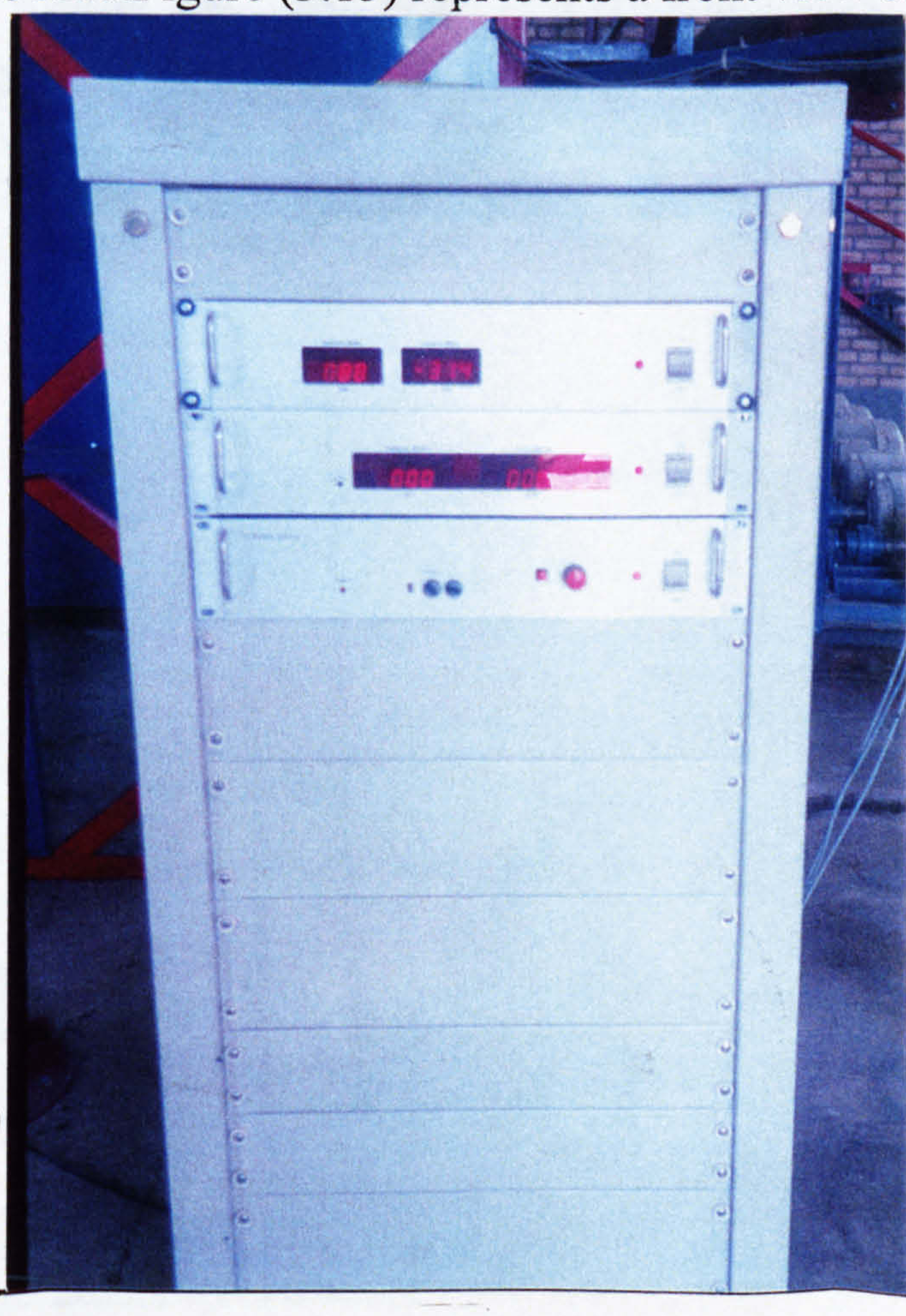


**Photo ( 3.15 )**

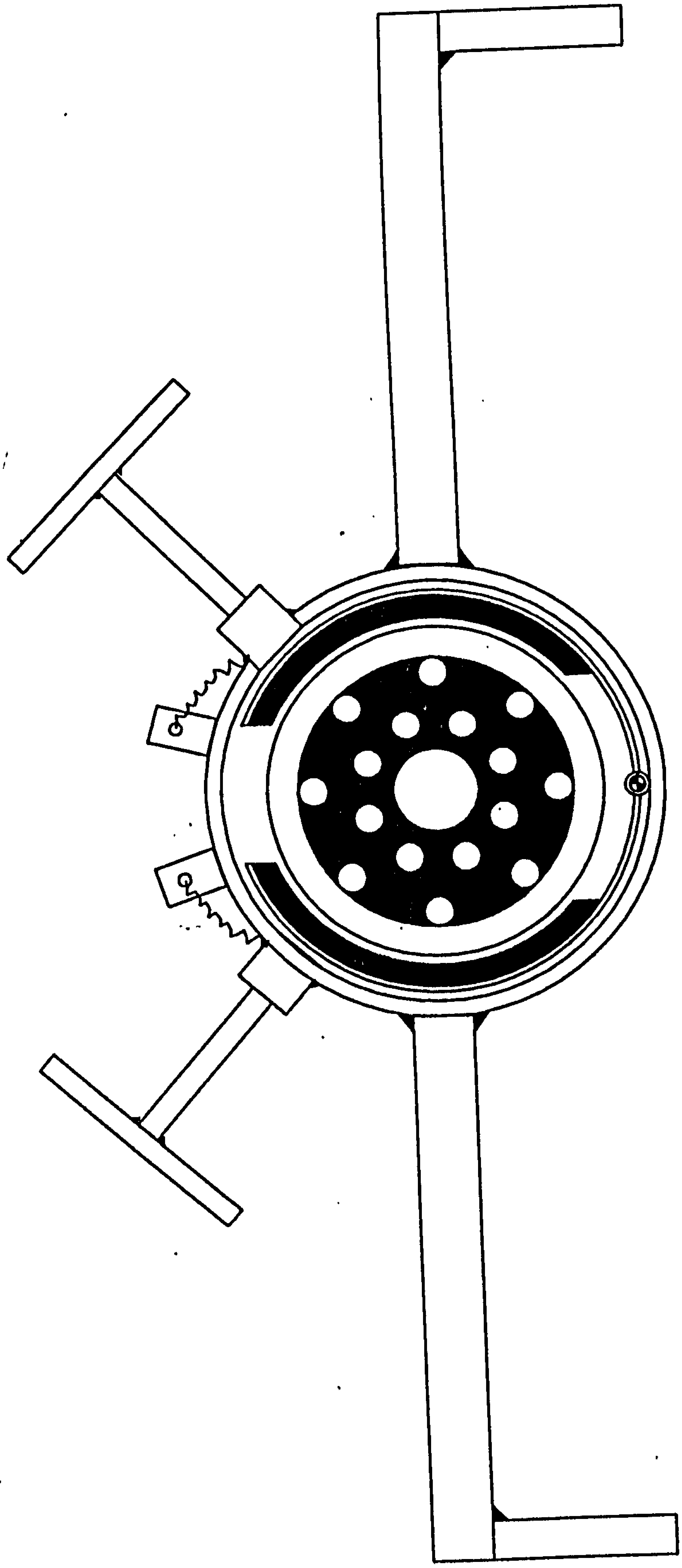


Figure 3.2

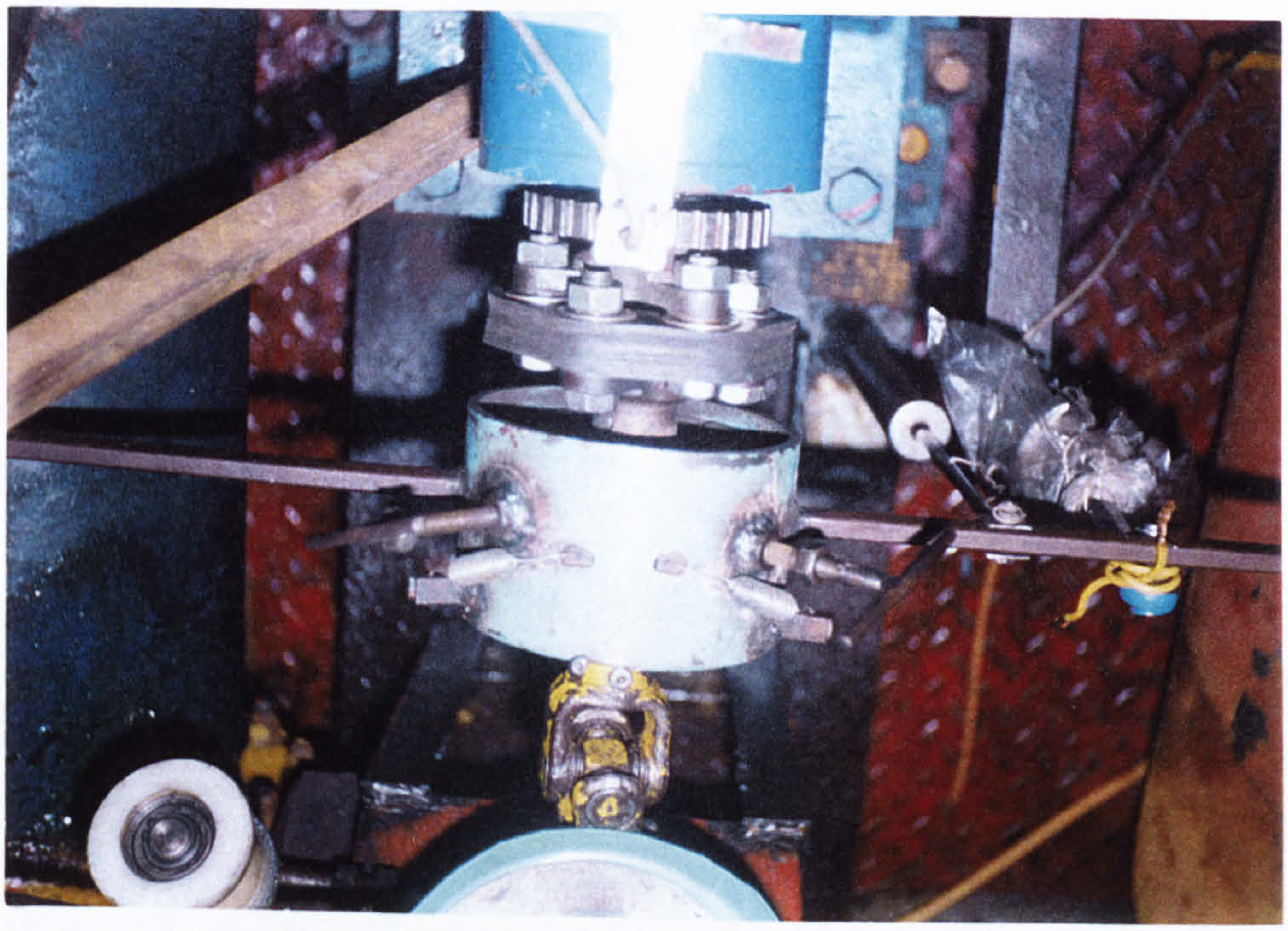


Photo ( 3.13 )

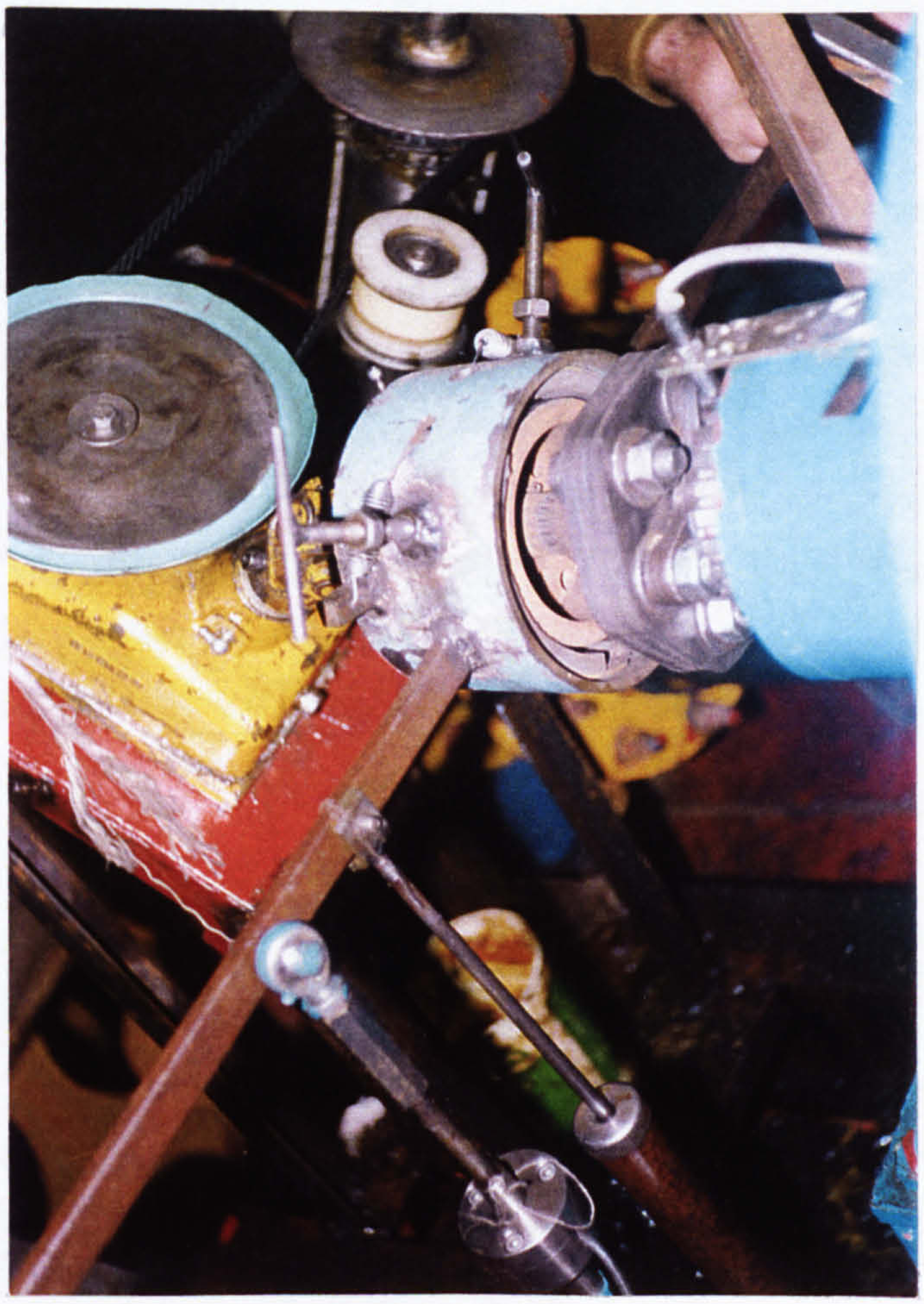


Photo ( 3.14 )

# Design details of the circuit

The following would present some details on design of the test rig.

## I.Pipings

Since the entrance of the turbine is of 254 mm diameter, to reduce propagation of turbulence at the entrance, 10 inch diameter pipes were selected for the piping system.

To reduce costs and speed up the rig building operations, it was first decided to use high pressure P.V.C. pipes and joints. This could also help, the circularity, and. cylindricality required by the ISO standards for the pipes. Also the surface softness of the pipes could reduce turbulence and losses. On the other hand chance of wear of pipings due to rust could have completely been eliminated.

However, after completion of the piping system, water was run through the system to check its performance. After a few times of testing the circuit, breakage of joints, specially at flanges, were observed, which could cause ever increasing leakage. After contacts, made with many suppliers of P.V.C. pipes and joints it was learnt that , joints (i.e. flanges, bends, etc.) of high pressure P.V.C. did not exist in Iranian market. Although many attempts were made to repair the circuit and prepare it for test purposes, but no success was achieved . Therefore it was decided to switch to Iron piping system.

## II. Design requirements of the circuit

As it may be seen from figure (3.1) the arrangement of the circuit after water leaves the pumps is as follows:

1. Collector
2. Reducer
3. Straight run
4. Orifice plate
5. 90° bends
6. Main gate valve
7. Straight run

**1. Collector** - As mentioned earlier. Collector is a 4 meter long Iron volume, having an internal diameter of twice larger than the internal diameter of the pipes (D). The reason for selection of such diameter lies in dimensional requirements for the reducer discussed in next Item (i.e. item 2)

**2. Reducer**- Is of semi cone shape, made of rolled Iron sheets.According to ISO standards, a reducer, causing a smooth reduction in cross section from (2D) to (D) over a length of (1.25 to 3D) would match the requirements of these standards. The length was selected to be equal to (3D).

**3. Straight run** - As stated in ISO standards, straight length required before an orifice plate and after a reducer of the above characteristics is equal to (11D). More than such length of pipe was allowed before the orifice plate. However, circularity and cylindricality of the pipe, were also in accordance with ISO standards.

**4. Orifice plate** - Altogether 4 orifice plates of different internal diameter size (i.e. 0.25D , 0.35D , 0.50D and 0.65D) were designed and manufactured.Photo (3.16) shows three of these orifice plates. These plates are of sharp edge type, with pressure tapping spacings D and D/2 from the center of the orifice plate upstream and down stream respectively; in complete agreement with ISO standards.

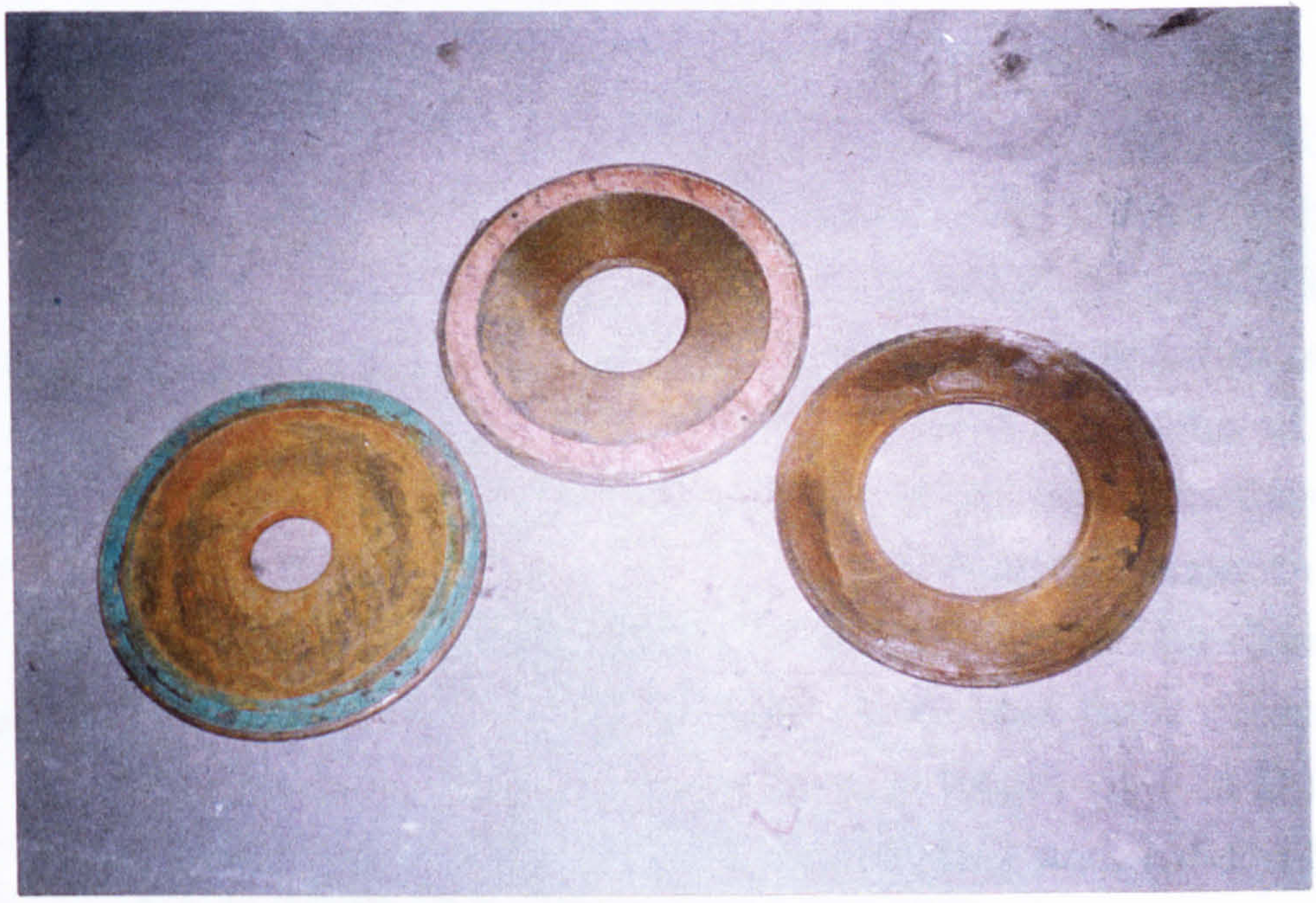


**Photo ( 3.16 )**

According to the same standards, a straight length of at least (7D) was allowed on down stream side after the orifice plate.However the following would present a summary of the installation requirements and also some details on design of orifice plates:

### 4.1. Installation requirements of an orifice

The following would present a summary of installation requirements for orifice plates according to ISO standards. Basically,the measuring process mentioned in these standards apply only to fluids flowing through pipe lines of circular cross-section.The pipe bore should be circular over the entire minimum length of straight pipe required.Circularity of cross-section is acceptable merely by visual inspection.This may be obtained by the circularity of the outside of the pipe . However , this circularity is not sufficient ,especially in the immediate vicinity of the orifice plate.On the upstream face of the orifice plate the length of the pipe adjacent to the plate should be at least equal to (2 D) and cylindrical.The pipe is said to be cylindrical when no diameter in any plane differs by more than 0.3%, from the value of ( D ) obtained as a mean value of all measurements of the diameter.The value for the pipe internal diameter ( D ) is the arithmetic mean of measurements at four diameters at least distributed in each of at least three cross-section themselves distributed over a length of ( .5 D ) ,two of which being at distances ( 0 D ) and ( .5 D ) from the upstream tapping.

it should be noted that within the above mentioned ( 2 D ) length,there should be no fittings or any other means of disturbance..

On the down stream face of the orifice at least along a length of ( 2 D ) from the upstream face the diameter of the pipe should not differ from the mean diameter of the upstream straight length by more than ± 3%.

The inside surface of the measuring pipe shall be clean,free from pitting and deposits and not encrusted for at least a length of ( 10 D ) upstream and ( 4 D ) downstream of the orifice plate.

Also there are minimum upstream and downstream straight length required for installation between various fittings and the orifice plate.Although some remarks have been made earlier on the subject but all the details are presented in appendix (IV).The values mentioned are only minimum values and application of longer lengths are recommended especially for research purposes.Table (3.1) represents some details on required straight length for orifice plates.

| $\beta$ | On upstream (inlet) side of the primary device | | | | | | | On downstream (outlet) side |
|---|---|---|---|---|---|---|---|---|
| | Single 90° bend or tee (flow from one branch only) | Two or more 90° bends in the same plane | Two or more 90° bends in different planes | Reducer (2 $D$ to $D$ over a length of 1,5 $D$ to 3 $D$) | Expander (0,5 $D$ to $D$ over a length of 1 $D$ to 2 $D$) | Globe valve fully open | Gate valve fully open | All fittings included in this table |
| $\leq$ 0,20 | 10 (6) | 14 (7) | 34 (17) | 5 | 16 (8) | 18 (9) | 12 (6) | 4 (2) |
| 0,25 | 10 (6) | 14 (7) | 34 (17) | 5 | 16 (8) | 18 (9) | 12 (6) | 4 (2) |
| 0,30 | 10 (6) | 16 (8) | 34 (17) | 5 | 16 (8) | 18 (9) | 12 (6) | 5 (2,5) |
| 0,35 | 12 (6) | 16 (8) | 36 (18) | 5 | 16 (8) | 18 (9) | 12 (6) | 5 (2,5) |
| 0,40 | 14 (7) | 18 (9) | 36 (18) | 5 | 16 (8) | 20 (10) | 12 (6) | 6 (3) |
| 0,45 | 14 (7) | 18 (9) | 38 (19) | 5 | 17 (9) | 20 (10) | 12 (6) | 6 (3) |
| 0,50 | 14 (7) | 20 (10) | 40 (20) | 6 (5) | 18 (9) | 22 (11) | 12 (6) | 6 (3) |
| 0,55 | 16 (8) | 22 (11) | 44 (22) | 8 (5) | 20 (10) | 24 (12) | 14 (7) | 6 (3) |
| 0,60 | 18 (9) | 26 (13) | 48 (24) | 9 (5) | 22 (11) | 26 (13) | 14 (7) | 7 (3,5) |
| 0,65 | 22 (11) | 32 (16) | 54 (27) | 11 (6) | 25 (13) | 28 (14) | 16 (8) | 7 (3,5) |
| 0,70 | 28 (14) | 36 (18) | 62 (31) | 14 (7) | 30 (15) | 32 (16) | 20 (10) | 7 (3,5) |
| 0,75 | 36 (18) | 42 (21) | 70 (35) | 22 (11) | 38 (19) | 36 (18) | 24 (12) | 8 (4) |
| 0,80 | 46 (23) | 50 (25) | 80 (40) | 30 (15) | 54 (27) | 44 (22) | 30 (15) | 8 (4) |

| | Fittings | Minimum upstream (inlet) straight length required |
|---|---|---|
| For all $\beta$ values | Abrupt symmetrical reduction having a diameter ratio > 0.5 | 30 (15) |
| | Thermometer pocket or well of diameter $\leq$ 0,03 $D$<br>Thermometer pocket or well of diameter between 0,03 $D$ and 0,13 $D$ | 5 (3)<br>20 (10) |

**Table 3.1**

Location of the orifice plate within the pipe line is also of importance.It should be centered in the pipe and the distance " $e_x$ " between the center line of the orifice and center lines of the upstream and downstream sides shall be less than or equal to:

$$e_x = ( .0005 \ D ) / ( .1 + 2.3 \ \beta )^4$$

( $\beta = d / D$ where d is the orifice diameter)

In addition it should be noted that the plate shall be perpendicular to center line of the pipe;to within $\pm 1°$.

**4.2. Orifice design**

Generally,various types of standard orifice plates,which basically is defined as a " restricted openings through which fluid flows " are similar and therefore need one discription.Each type of standard orifice plate is characterized by the arrangements of pressure tappings.Figure (3.3) represents an axial plane cross-section of an sharp edge orifice plate.The part of the plate inside the pipe shall be circular and concentric with the pipe center line.The faces of this plate shall always be flat and parallel.

In designing an orifice plate it should be noticed that the strength of the differential pressure or other stresses may cause plastic buckling and elastic deformation in the plate. These would have adverse effects on the accuracy of the device and therefore it is essential to take all the necessary measures while designing the plate.The upstream face of the plate ,**A**, shall be flat and completely perpendicular to plate's center line. The down stream

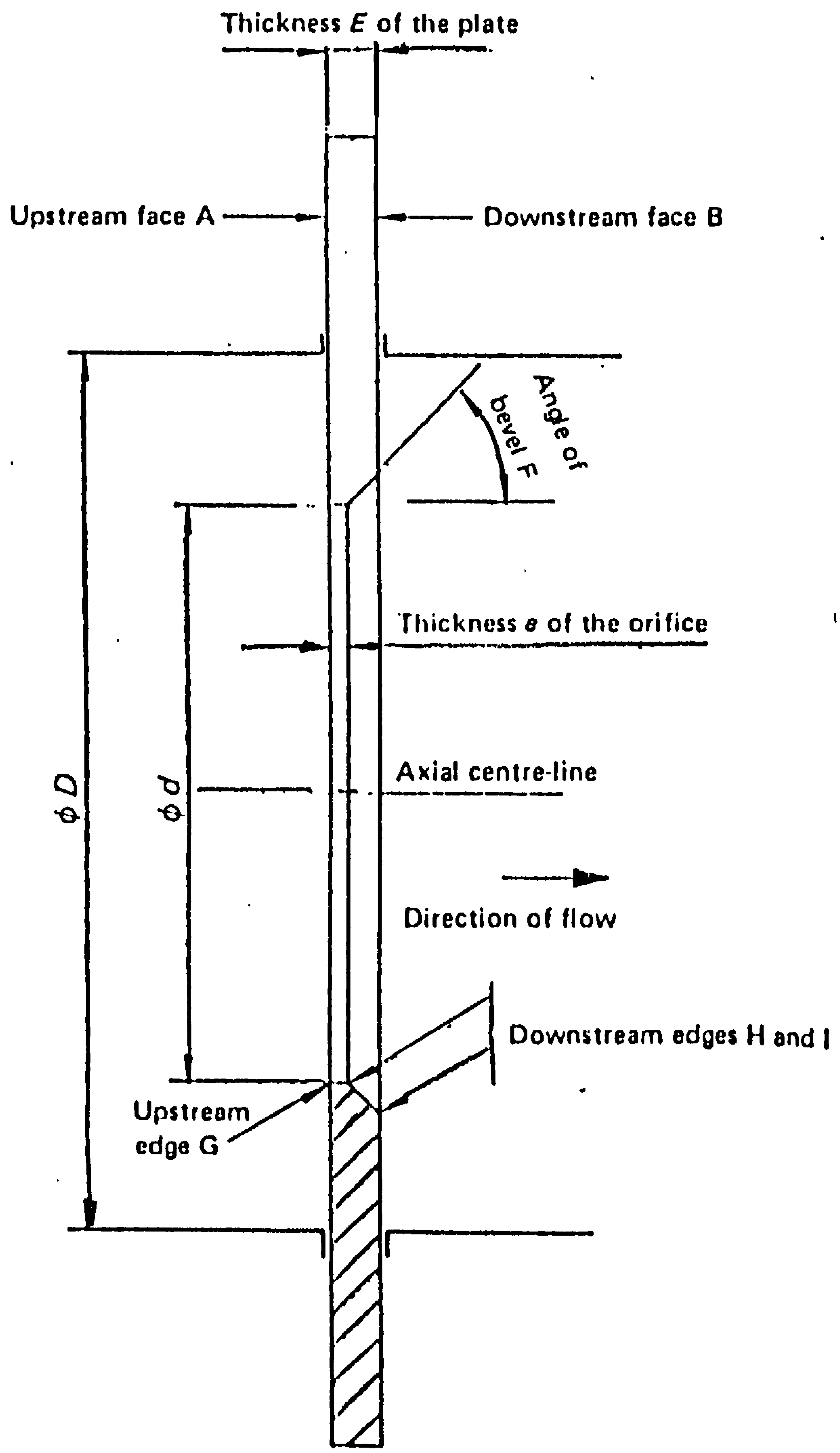



**Figure 3.3** Standard orifice plate

face,however,should be parallel to the other face.The most important dimensions in design of an orifice are ,E (thickness of the plate), e ( thickness of the orifice),and F (the bevel angle). The ranges determined by ISO standards for these dimensions are ( .005 to .02) for e , (e to .05 D ) for E, and (30 to 45 ) for F; respectively.However,values of (.02 D) for thickness "e" ,(.04 D) for thickness "E" ,and 45° for angle of bevel were selected.

Since the orifice plates used were of the type with D and D/2 tappings ,then the spacing of the pressure tappings were selected according to ISO standards.From figure ( 3.4 )it may be observed that the spacing $l_1$ of the upsteam pressure tapping is nominally equal to D,but it can vary between .9 D and 1.1 D ,without introducing any errors to the readings.

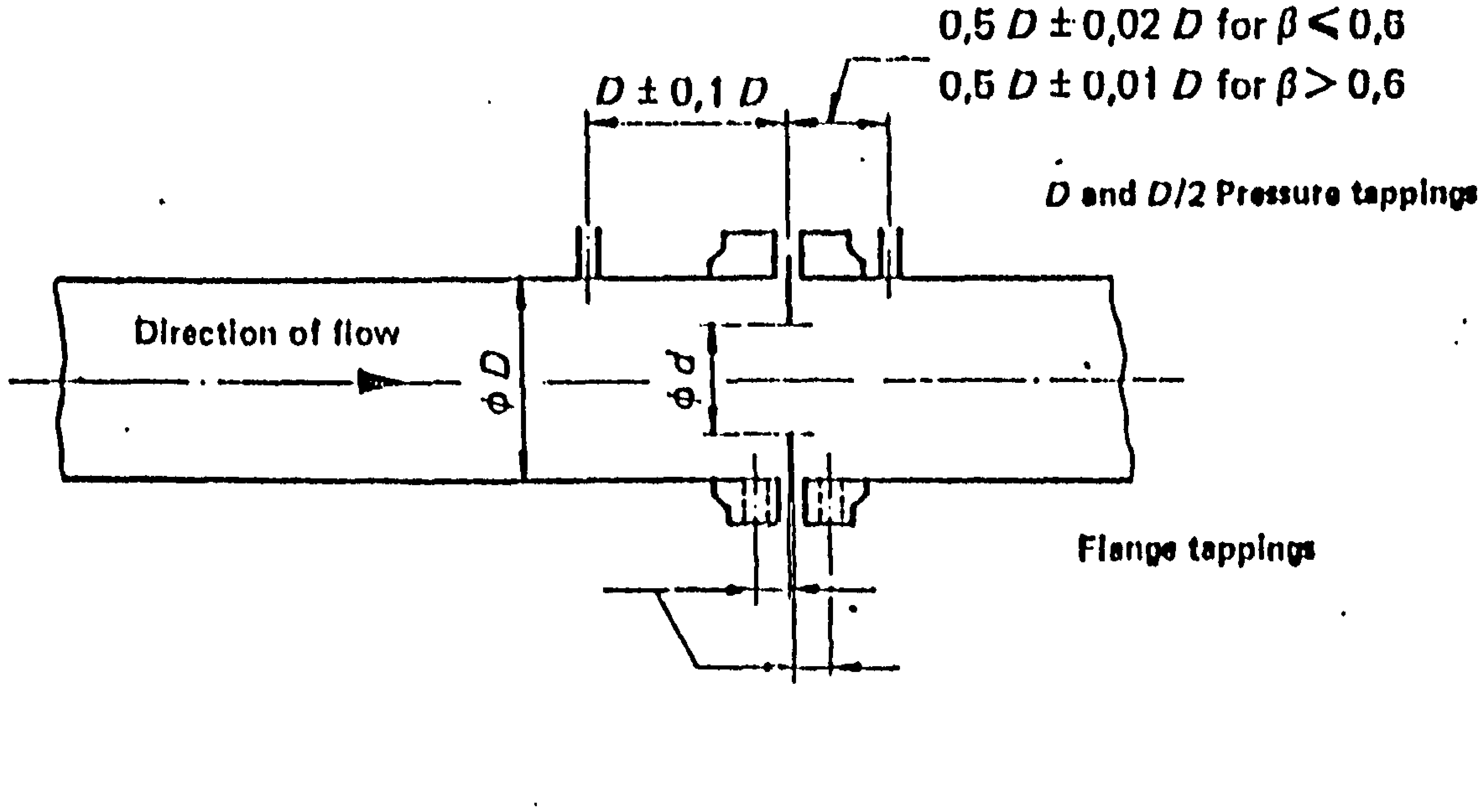



**Figure 3.4** - Spacing of pressure tappings for orifice plates with D and D/2 pressure tappings.

The spacing $l_2$ of the downstream pressure tapping which is nominally equal to .5 D but it can very between 0 .48D and 0 .52 D when $\beta \leq 0.6$ and between 0.49 D and 0.51 D when $\beta > 0.6$ . The spacing $l_2$ was selected and made in such a way so that it would suite all values of $\beta$.

Appendix (IV), presents all the standard details of the orifice plates their installation requirements and their design .

**5. 90° bends** - Two 90 ° bends with a 2.5 ( m ) straight length in between would bring the flow back to its rout towards the main reservoir.

**6. Main gate valve** - This gate valve which can control the flow rate (i.e. the head) to the turbine, is installed at the beginning of the return brunch of the pipings towards the main tank.

**7 Return straight length** - After the gate valve a straight length of over 30 D was allowed before the entrance to the turbine.

# CHAPTER

# ( IV )

# AGNEW TURBINE

# Agnew turbine

## History

Intrigued by the oil crisis during early 1970's , Mr. Agnew thought of benefiting other alternatives and rather unconventional sources of energy. He selected micro hydro potentials and started his work on designing a suitable turbine for such potentials.

At first, it was intended to make a turbine similar to that of Kaplan; but without guide vanes. Such a choice was made due to the fact that , kaplan turbines usually have high running speeds. This would help in reduction of losses due to conversion of turbine speed to a suitable speed for the generator; since most of small standard alternators, run at 3000, 1500, or 1000 r.p.m.

However, with a help of a small company the first prototype was manufactured; which was not a successful turbine. Figure (4.1), shows this turbine. Later through H.I.D.B. (Highlands and Islands development board) Mr.Agnew was joined by Mckellar Engineering Ltd. Their work then lead to the present turbine, which is a 45° inclined, adjustable blade axial propeller turbine.

Figure (4.2) , represents schematic view of the Agnew turbine. At present, 10 Agnew turbines are installed and working at different locations in Scotland.Also,an overall view of the turbine at the test rig is shown by the photo (4.1).

## Selection of turbine size

Agnew turbine has, basically been designed in different sizes. These turbines are actually named after their tip to tip blade diameter, such as 200mm , 300mm and 500mm. For the purpose of this work a

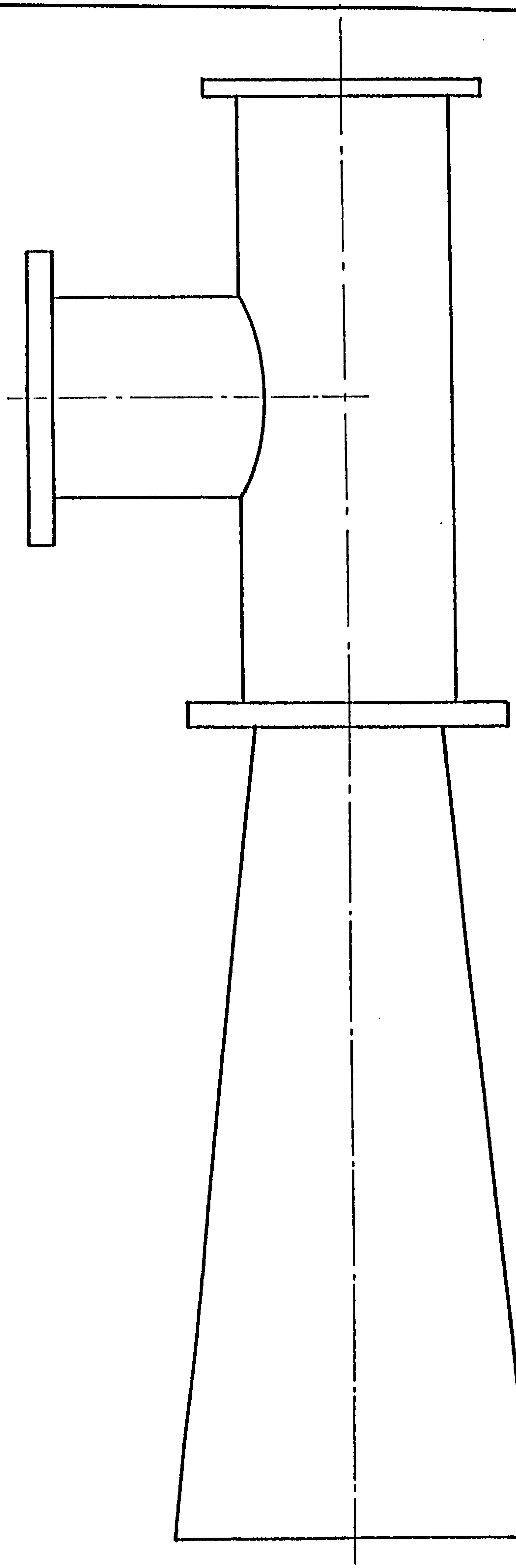


Figure ( 4.1 )

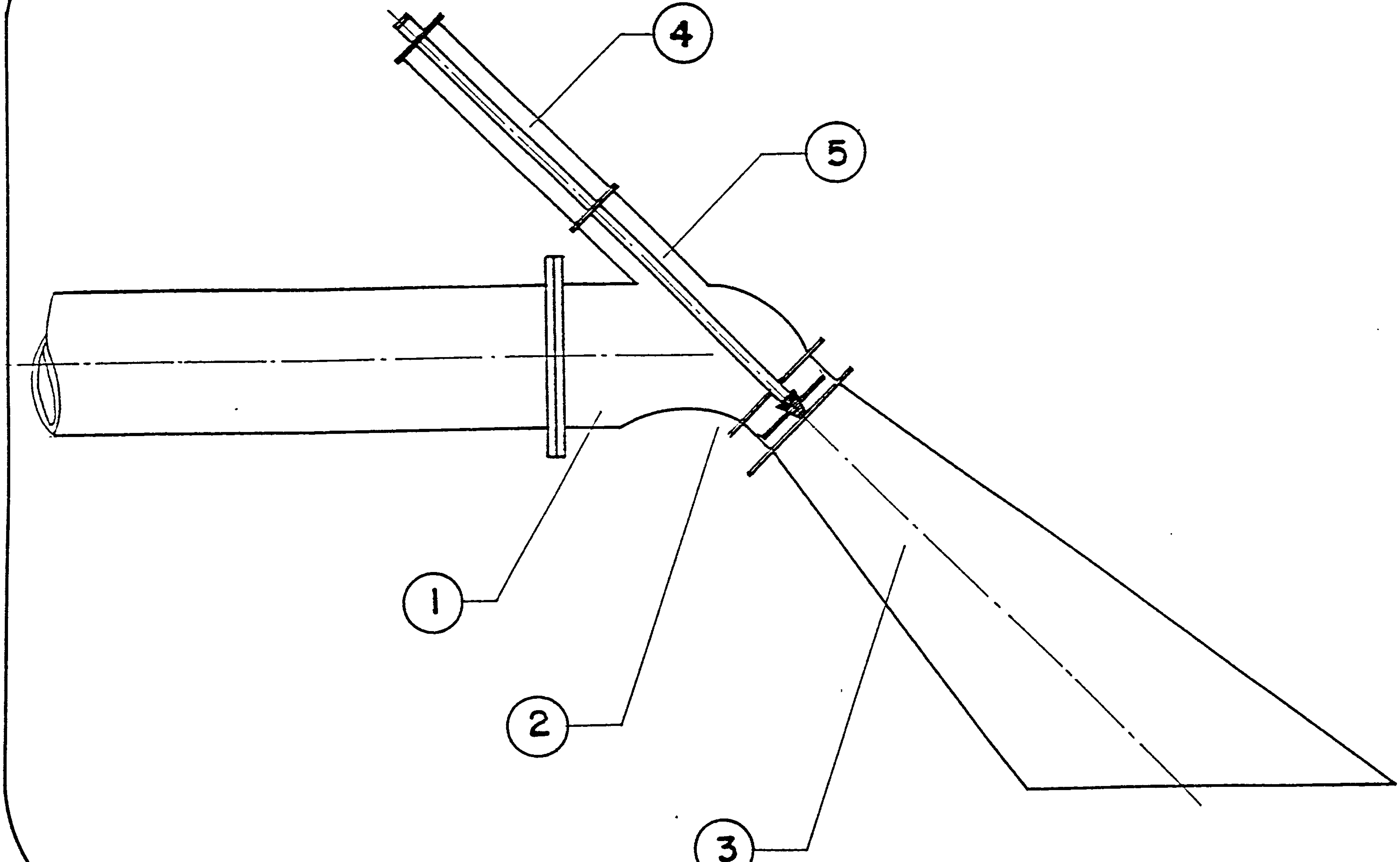



Figure ( 4.2 )

**Photo (4.1)**

150 mm design was differentiated directly from the 300mm design. This would serve two different purposes. First, a smaller turbine's requirements in terms of head and flow rate, would be more limited and closer to conditions present in Iran. Second, this turbine may also be used as a (1:2) modle for 300 mm design.

## Details of the turbine

The complete turbine and parts assembly are constituted of several sub. assemblies according to figure (4.2).

### i. Main turbine casing

This part consists of a short straight run as the entrance of the turbine, *1*. This run would lead to a 45° bend of reducing cross-section, These two parts are flanged together *2*.Photo (4.2 ) represents a view of the casing.

**Photo (4.2)**

**ii. <u>Draft tube assembly</u>**

It consists of a cone cut horizontally at the larger end and a flange at other end for connection to the main turbine casing, *3*.

**iii. <u>Main turbine shaft</u>**

This is a hollow shaft through which the operating rod runs. It rests on three bearings, two of which are tapered and other is of ordinary ball type. It also passes through a seal to avoid water entering the bearings housing. At one end it holds the runner and blades assembly and at another the toothed wheel, for transmission of power to the gear box; via a toothed belt.

iv. <u>Housings (Bearings and seal assembly)</u>

All three bearings are placed inside the bearings housing,***4***. The seal is placed inside a separate housing, ***5***. Both housings are well aligned , since their misalignment could cause excessive pressure on the seal resulting its rapid wear and deformation.Photo (4.3)

**Photo (4.3)**

**v. <u>Runner and blades assembly</u>**

Four blades are set around the runner which it self houses the blade angle controlling mechanism inside it. This mechanism is activated by the operating rod.Detailed drawings of the turbine sub assemblies are presented in appendix (V).Photo (4.4) and photo (4.5) show the runner.

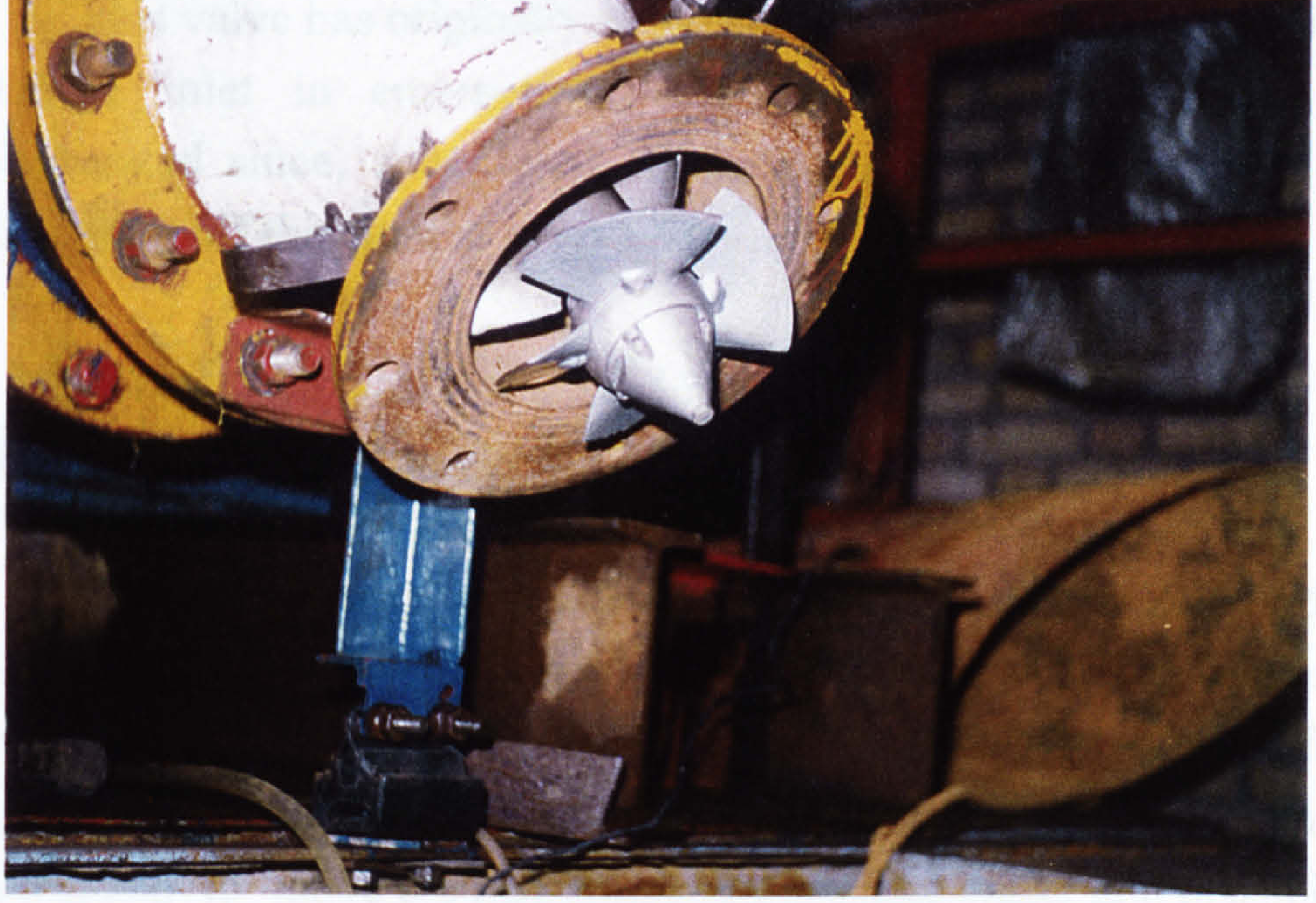


**Photo (4.4)**

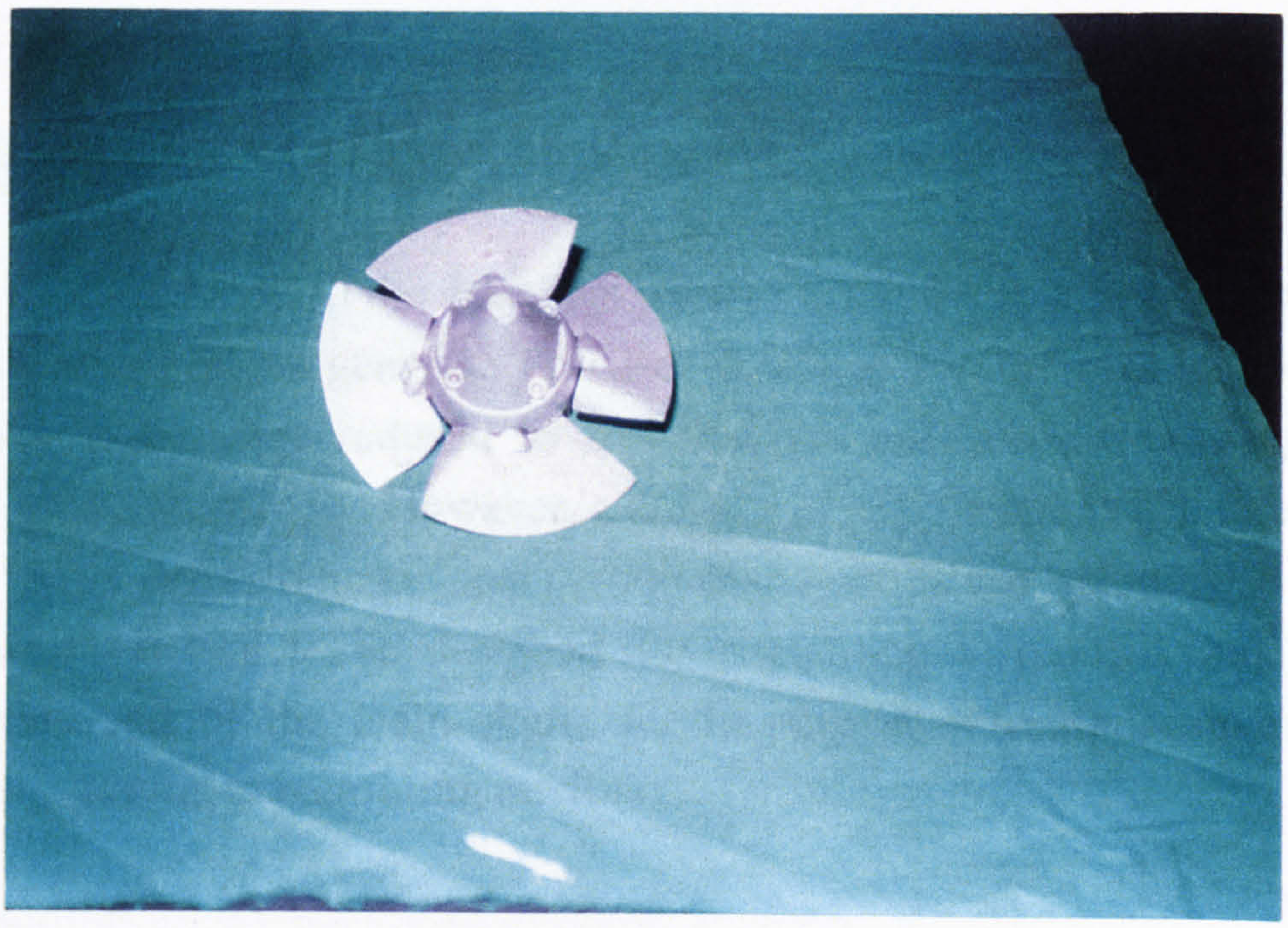


**Photo (4.5)**

**vi. Butterfly valve assembly**

This valve has originally been designed for the sudden closure of the turbine inlet in emergencies. However , for our purpose , it was eleminated since, shut down of the turbine could easly be achieved by switching off the pumps.

## Redesigning and manufacturing

Actual detailed study of the turbine and its parts and all drawings was accomplished in Glasgow. All queries were discussed with Mr.P.Agnew. It was then found that, for the purpose of manufacturing the turbine some final touches had been made to the design at the workshop to make the manufacturing process, easier and possible. Also, a number of drawings were missing. At the time, all the missing detailed drawings were redrawn according to their matching parts and in accordance to the assembly drawing of the turbine.

## Generation of 150 mm turbine (1:2 model)

Since all the drawings were of the 300mm turbine, therefore, it was essential to generate detailed drawings for the (1:2 ) model. All dimensions were reduced to half and part and assembly drawings were plotted accordingly. However, a number of modifications had to be made to some parts, to make installation easier on the test stand. Also , some new parts had to be designed for assembling the toothed wheel on the outer end of the main shaft, for the purpose of transmission of power from turbine to measuring devices.

## Assembling, testing and trouble shootings

All parts of the turbine except the main turbine casing and the draft tube were manufactured at I.R.O.S.T's workshop based at Asr-E-Enghelab complex, where actually the laboratory is situated. The casing and the draft tube were manufactured by an outside workshop. Assembly process took place at the laboratory, after which the turbine was craned up to the stand and installed. The turbine was then tested with different limited flow rates under no load conditions for many times. Since everything seemed under control then it was connected to the measuring devices. The turbine was then tested under different loading conditions with higher flow rates and speeds. Under such conditions a number of problems started arising, which seriously effected the turbine's performance and damaged some parts. The causes of these problems may fall into two categories:

**1**. Vibrations, mainly due to unbalanced rotors of the hydraulic dynamometer.
**2**. Weakening of some parts due to halving dimensions.

problems caused by vibrations and the way they were encountered may be highlighted as follows: Collision of running blades against the casing which caused deformation of blade edges, and in two cases, their breakage, from the welding point of the blade and the tail. To recover the blade every time it was neccessary to empty the tank and refill it. To strengthen the welding point, special electrodes were used at another outside workshop; after which no similar case was observed.

Also, Large fluctuations in torque and speed readings were other indications of presence of vibrations. Naturally,these vibrations could cause uncertainties in collected data.To reduce vibrations and their effects on the system,dampers were applied at certain points.

Since unbalanced rotors of the hydraulic dynamometer were a major source of propagation of vibrations , later, another dynamometer of Pronney type was designed, manufactured and installed; instead of the original hydraulic type.

One of the major problems of the second category which were chiefly caused by reduction of dimensions was rapid wear of the seal, which resulted in, water traveling upwards into the bearings housing and damaging bearings. This required changing of the seal after a few hours working of the turbine under load. This problem was caused due to the fact that, because of halving the dimensions, the bearing housing did not penetrate deep enough into the seal housing . Therefore, the joint was not strong enough against the pull of the toothed belt, connecting the main shaft to the gearbox and the dynamometer. This belt pull could cause eccentricity between the main shaft's center line and the turbine's casing center, which in turn could, cause excessive vibrations and danger of collision of running blades against the casing.Photo (4.6) shows the connection of the turbine and gear box by the toothed belt.

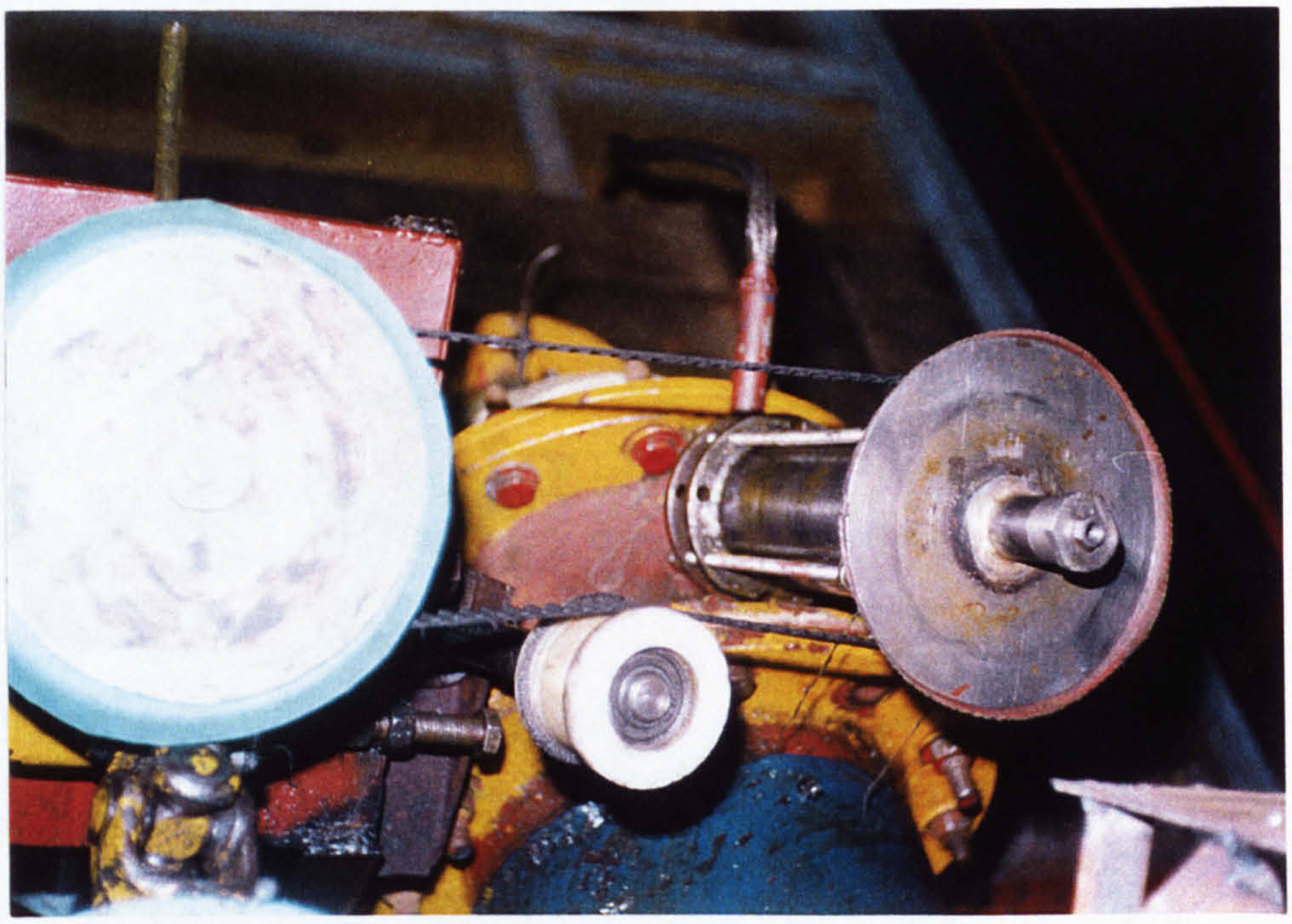


**Photo (4.6)**

To strengthen the part against the belt pull and its effects the seal housing was redesigned and manufactured in order to allow the bearing housing to penetrate well inside it.After this modification ,the seal worked for a year and is still working. Photo( 4.3) also shows this section .

Deformation of the parts of the blade angle controlling mechanism was another problem resulted from the reduction of dimensions. The constituting small parts of the mechanism simply could not take the loads exerted upon them and therefore they either were deformed or broken at times. To Circumvent these problems the blade angle controlling parts were redesigned and strengthened in a way to tolerate the shocks and the bending moments exerted on them by the tail of the blade. Modifications were made to these parts according to available space inside the runner. Drawings of these parts are also presented in appendix (V).

## Improvements on Agnew turbine

As manufacturing of the base modle turbine ended it was assembled and primary running tests,and measuring devices tests were performed.Later the turbine was tested under different loading conditions and all readings were taken.It was then appropriate for performing suggested improvements on the test modle to check its behavior and compare it with the base modle.

## Areas of improvements

As mentioned before early studies on the turbine indicated three main research areas which could possibly affect the performance of the turbine. These areas are namely,guide vanes , draft tube and the runner blades.

## Selection of prior improvement area

During early running tests and actual base modle tests the general behavior of the turbine was closely observed.These observations indicated reasonable performance of the running blades and the draft tube.Therefore it was decided to begin with introduction of guide blades to the turbine,and parallel with testing the turbine,observe and study the performance of both the running blades and the draft tube.In this way a better view of their weak points,design shortcomings,and performances could have been obtained.

## Guide blades

To design the guide blades it was essential to define clearly the purpose of their application.This would therefore, govern the ultimate shape of the blades. On the other hand,their installation, and control mechanism were also of paramount importance.The following presents the major steps involved in design,manufacture and installation of the guide blades mechanism.

### 1.Number of blades

In order to avoid any synchronous effects it is suggested to select the number of guide blades,so that,their number is either one less, or one more than the number of runner blades.

i.e. $Z_G = Z_R - 1$

$Z_G$ = No.of guide blades

$Z_R$ = No.of runner blades

Therefore, 3 blades were decided to be installed,as the flow guiding mechanism. Appendix (V) presents the detailed drawings of the mechanism.

**2.<u>Shape of the blades</u>**

Blades are of aerofoil cross section.Their original shape is a rectangle of ($10 \times 5.5$ cm²) . Due to limitation of space, these dimensions could not have been selected any larger.

Blades were also curved so that, flow could whirl before_entering the runner .Photo(4.7), shows the guide blades.

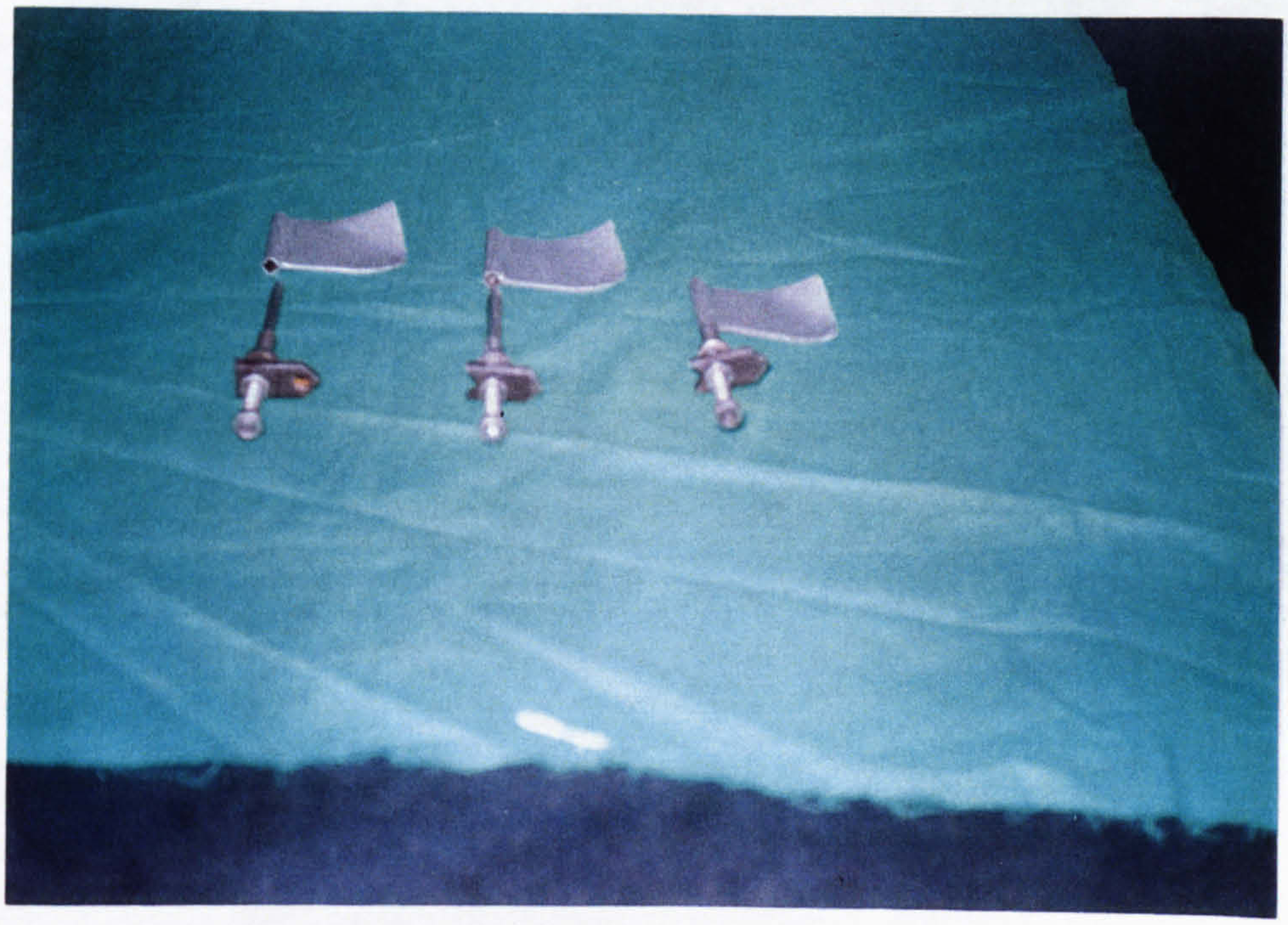


**Photo ( 4.7 )**

### 3.Position of the blades

Naturally, guide blades should be installed at an immediate position before the runner blades.With original turbine design this would mean,inside the main casing at the bend,which in turn could cause too much complication in design,manufacturing and installing the blades.The straight,cylindrical section at the end of the main casing and just before the entrance of the draft tube seemed to be an appropriate space to install the guide blades mechanism. This is where the runner blades operate in the original design Therefore it had to be shifted downwards to make the neccessary space for the guide blades.Also appropriate room for the runner to operate had to be considered.This would in turn require extension of the main shaft. Photo (4.8 ) and photo ( 4.4 ) represent the guide blades at their position.

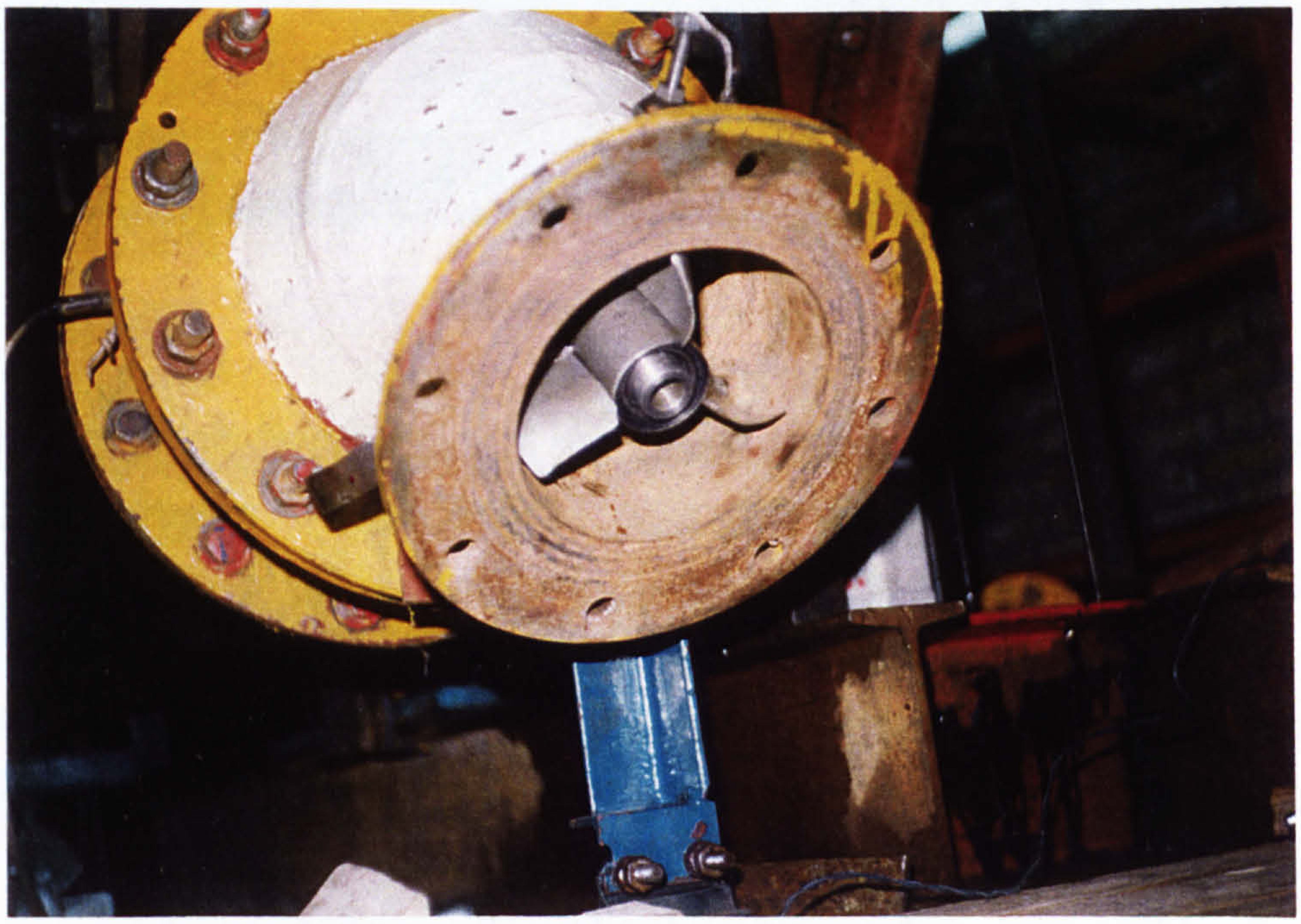


**Photo ( 4.8 )**

**4.Extension of the main shaft**

This extension is of 13 (cm) length and consists of Three main parts ( Photo 4.9 ).

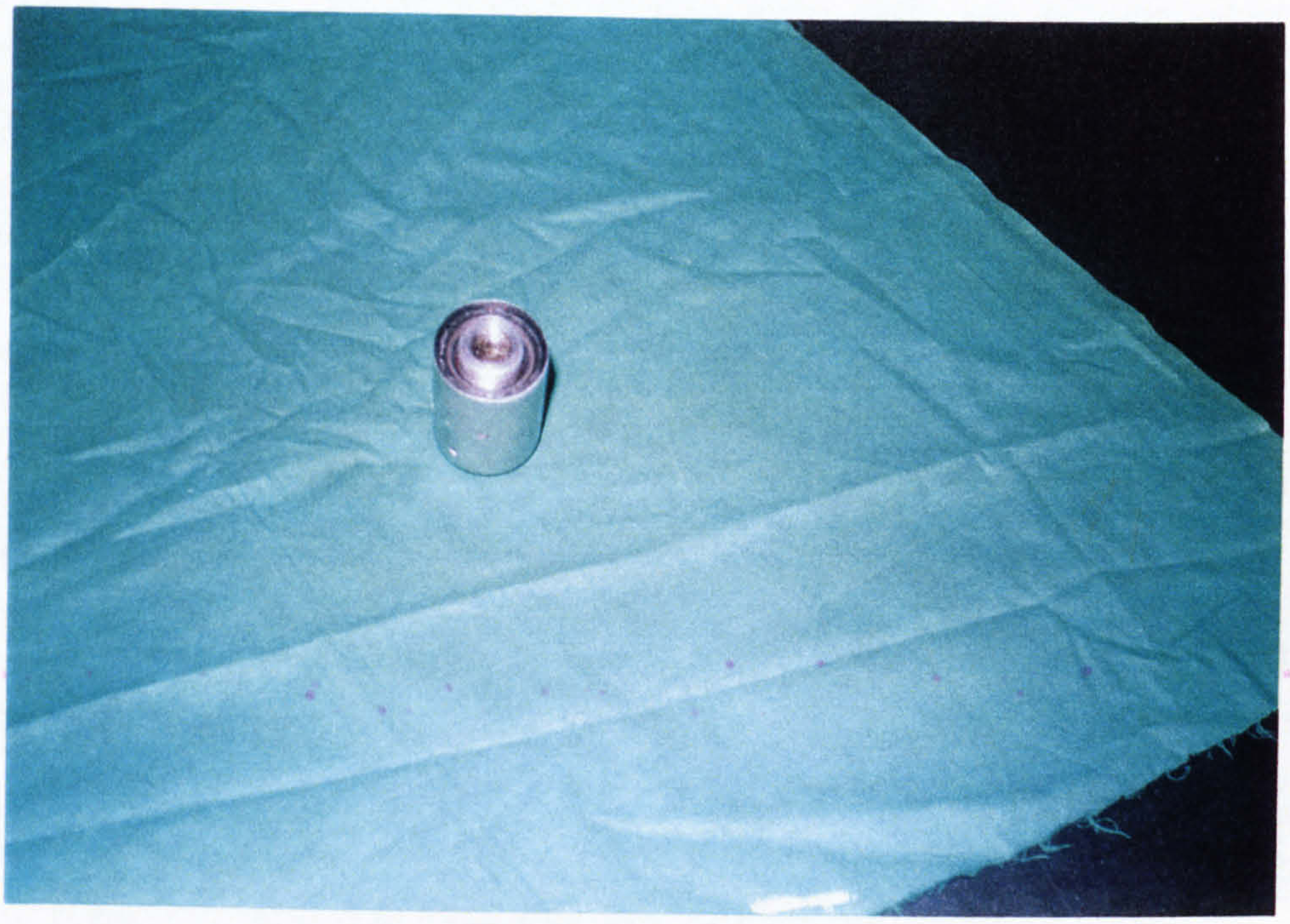


**Photo ( 4.9 )**

One of these parts is the internal shaft.This part is connected to the turbine`s main shaft and actually rotates with it.Another important part is the stationary casing.This part actually is an stationary cover for the rotating part which also operates as a holder of guide blades holding rods.Three holes of $120^\circ$ are made around this casing , inside which the roots of the holding rods rest. A needle bearing and two seals on each side of it comprise the third main part and separate the internal shaft and the casing.The bearing makes it possible for the internal part to rotate with the turbine`s shaft inside the stationary casing . Photo (4.10), represents the assembly of this part and the guide blades.

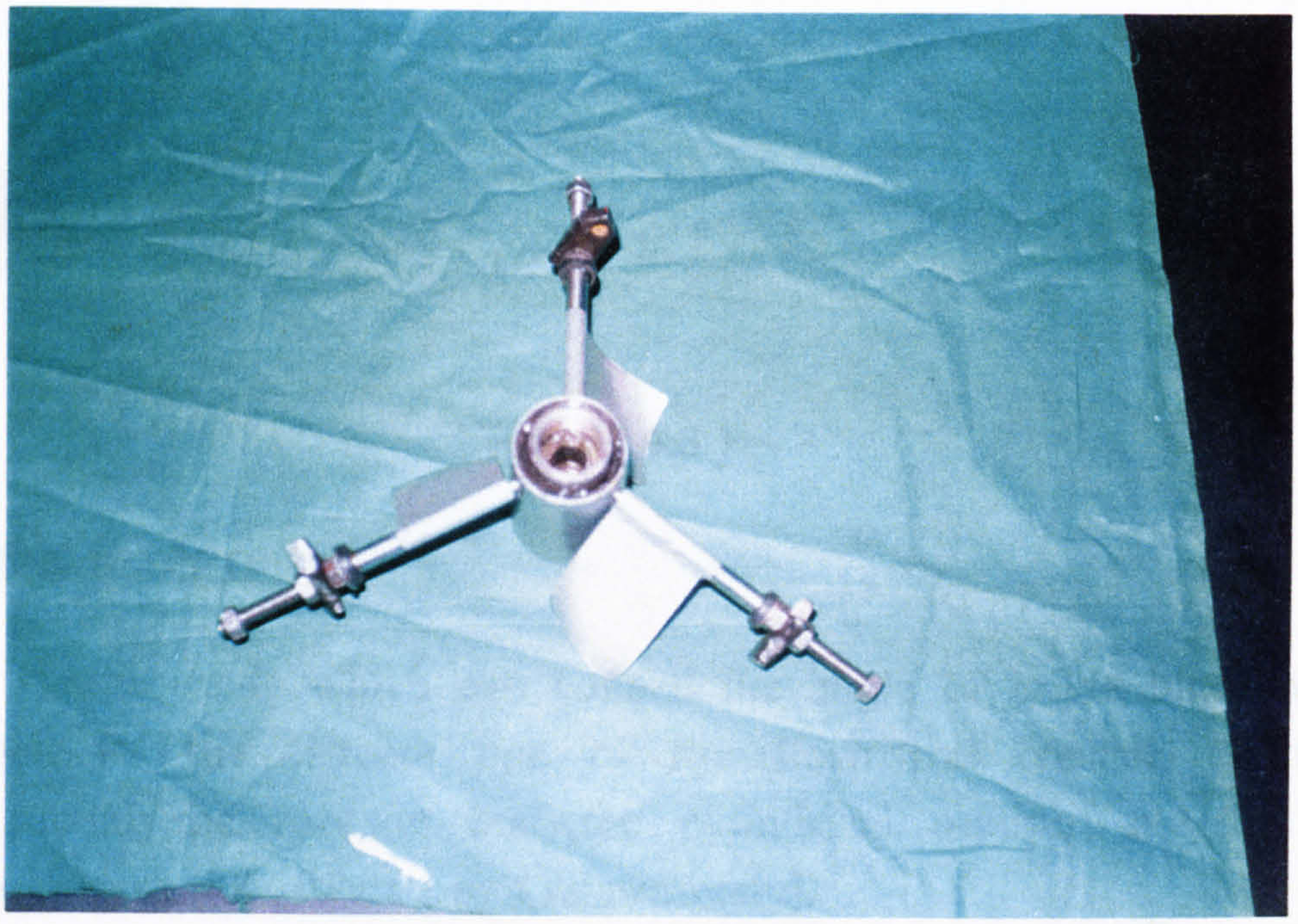


**Photo ( 4.10 )**

**5.Blades**

A hollow cylindrical rod is welded to the straight,shorter side of the blade.The through hole is of square cross section and allows the blade holding rod to pass through. (photo 4.11). Also figure (4.3) represents a sketch of one of the guide blades.

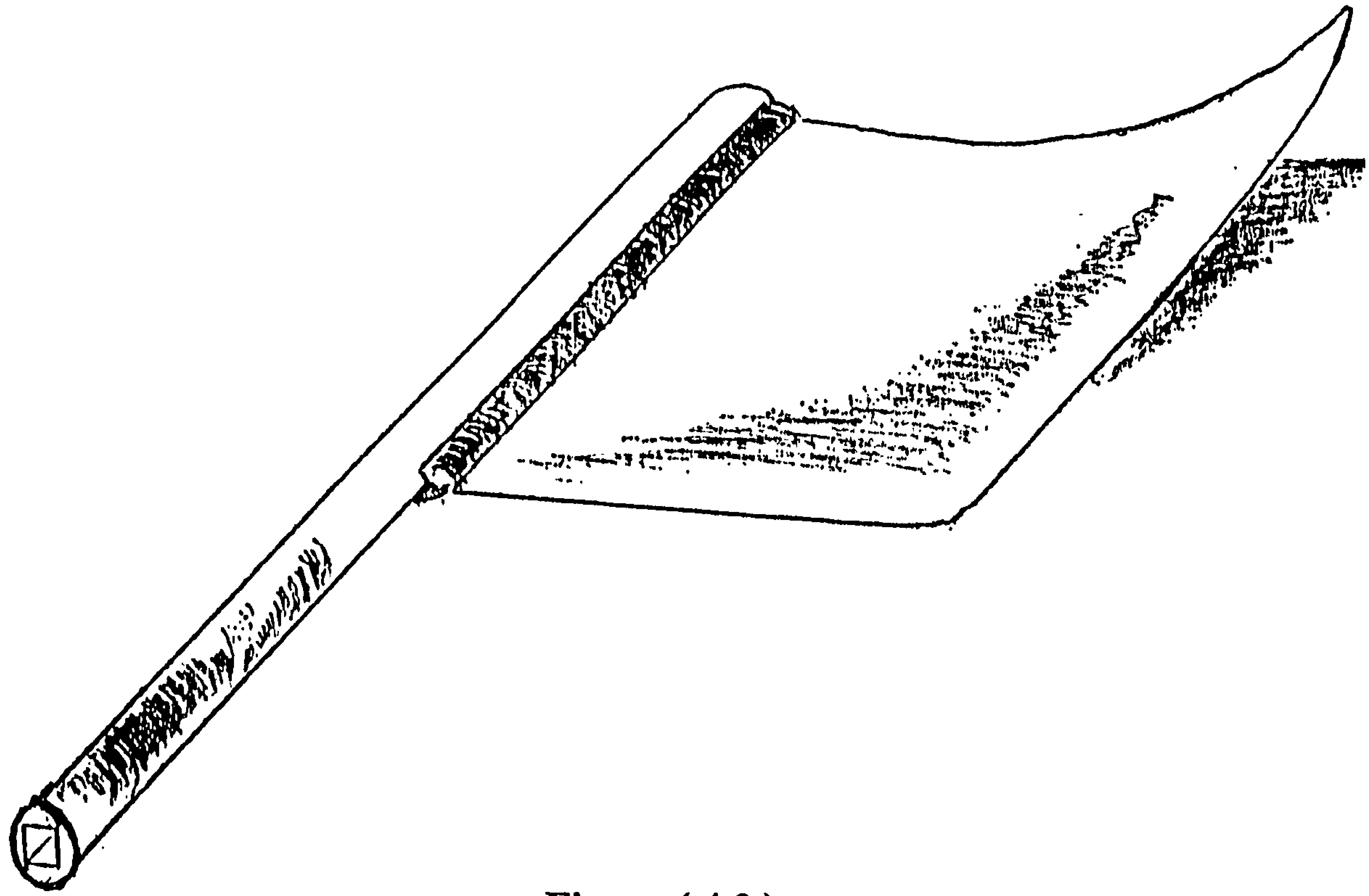


Figure ( 4.3 )

### 6.Guide Blade rods

These rods which can control the angle of guide blades,with respect to the stream line of flow.Each rod consists of four parts,namely,the root,the body,the cylindrical section and the end screw,(Figure 4.4 , items 1 to 4 respectively).

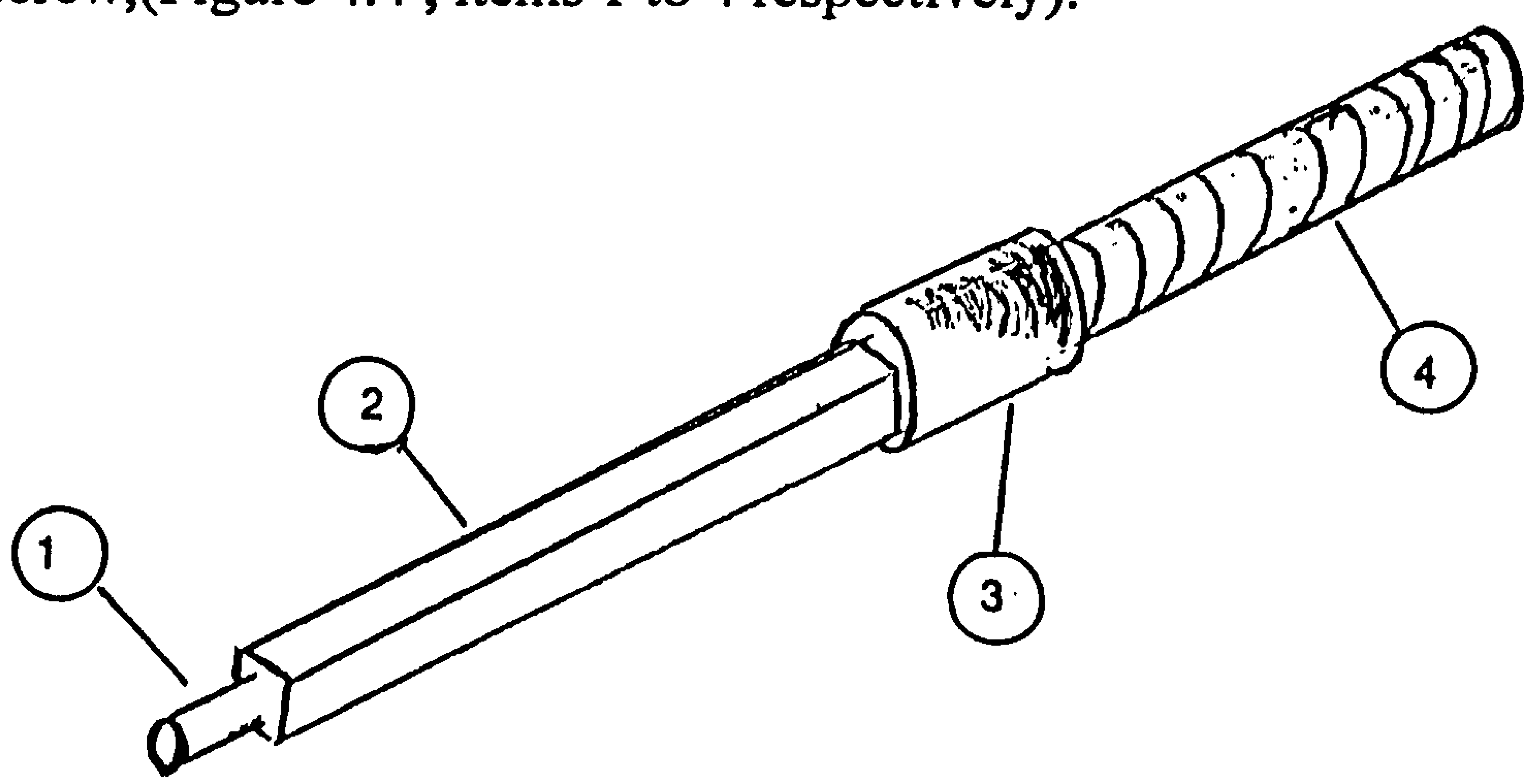



Figure ( 4.4 )

*1.*The root is of cylindrical cross-section (5mm diameter and 7mm long) , and rests inside the casing of the extension of the main shaft. *2.*The body is of square cross section and 5.5 cm long ,which passes through the hollow shaft welded to the blade. The square cross-section helps avoid rotation of the guide blade around the operating rod due to water pressure. *3.*The cylindrical section passes through the bushings welded on the turbine`s casing. The close tolerance between this section and the bearing serves two purposes. First, is to avoid water passing through and ejecting into atmosphere. Second, is to secure the holding rod against side movements. *4.*The end screw holds the rubber seal, the pointer, and a holding nut. The holding nut actually secures the pointer so that its position would not change once it is set at the beginning of the test. (Photo 4.12 )

## 7. Bushes and stoppers

Three bushes of internal diameter the same as the cylindrical section of the operating rod are welded to the casing . These bushes guide and support the rods against side movements,therefore, the rods can only rotate around their central axis . Also , the radial movements of the rods are controlled by " ˥ " shape part . The free end of the long side is welded to the turbine's casing and the short side acts as the stopping part to the outwards displacements of the rod (Photo 4.13 ).

## 8. Pointers

At the end of each blade operating rod a pointer is screwed,so that the angle of blade with respect to the direction of flow may be shown . Rotating these pointers result in rotation of guide blades . Degree scalings are provided on the turbine 's casing , with the sharp end of the pointer aiming at them , so that the rotation of the pointers can be

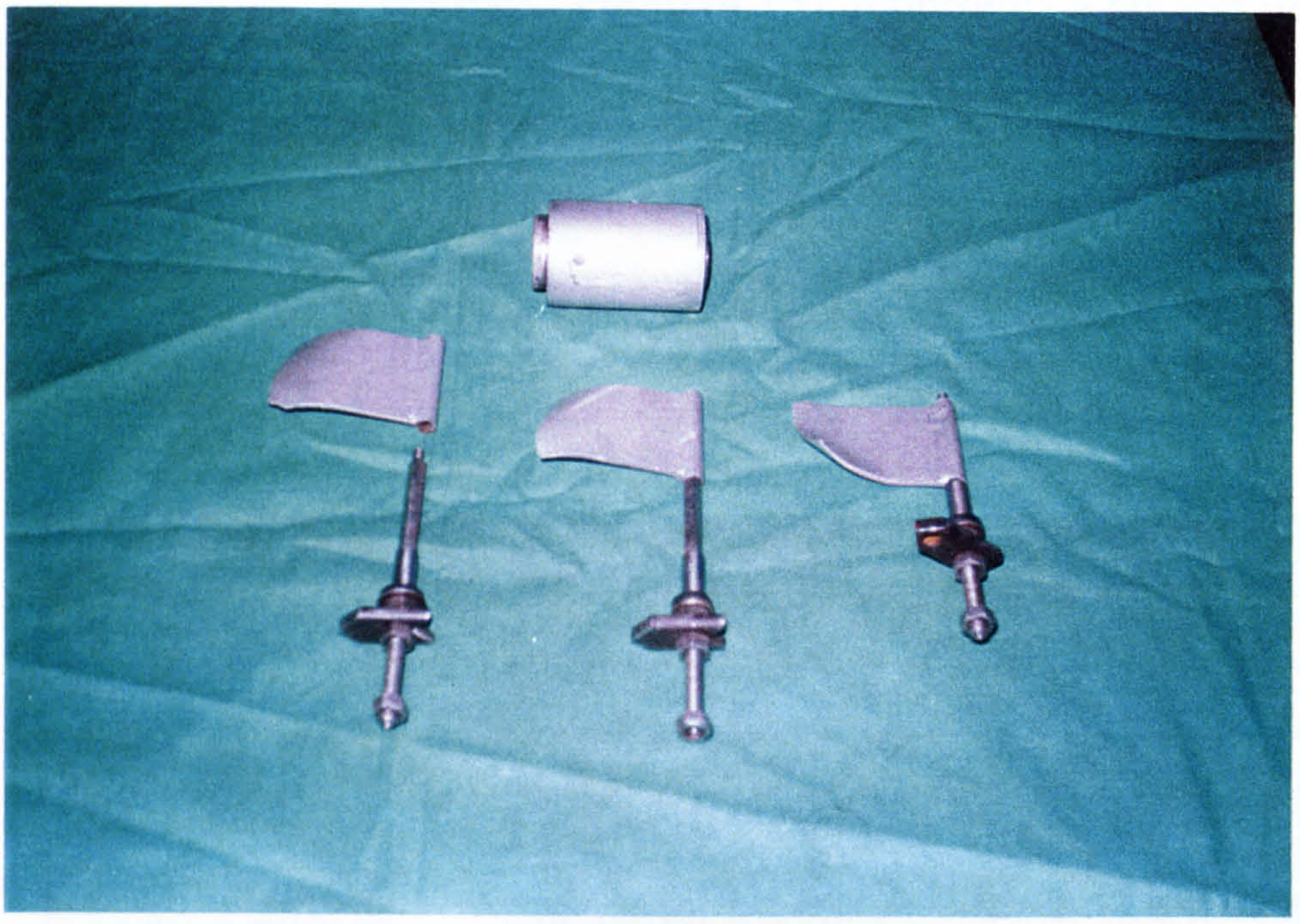


Photo ( 4.11 )

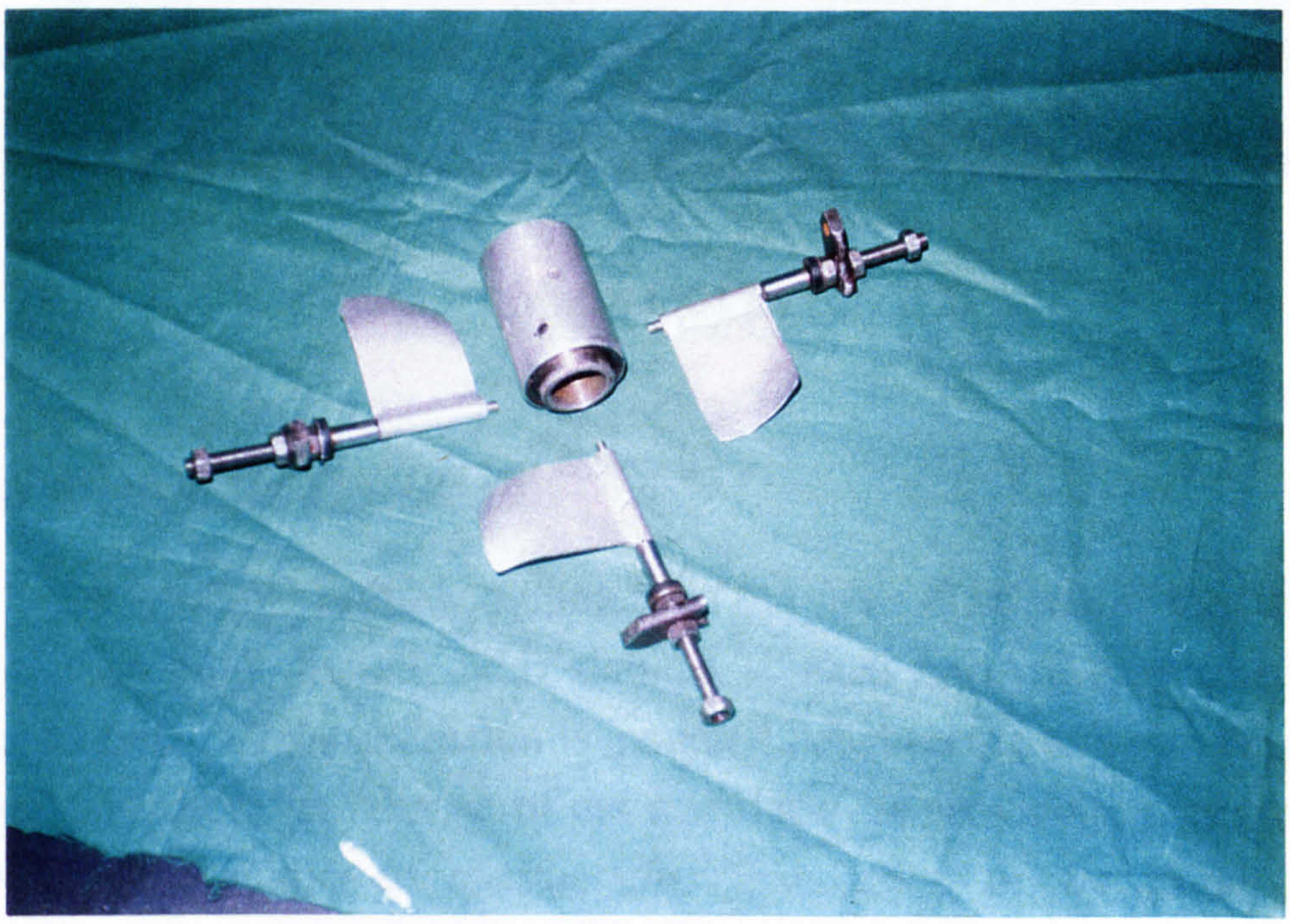


Photo ( 4.12 )

measured in terms of degrees . Photo (4.13), represents a pointer on the main casing of the turbine.

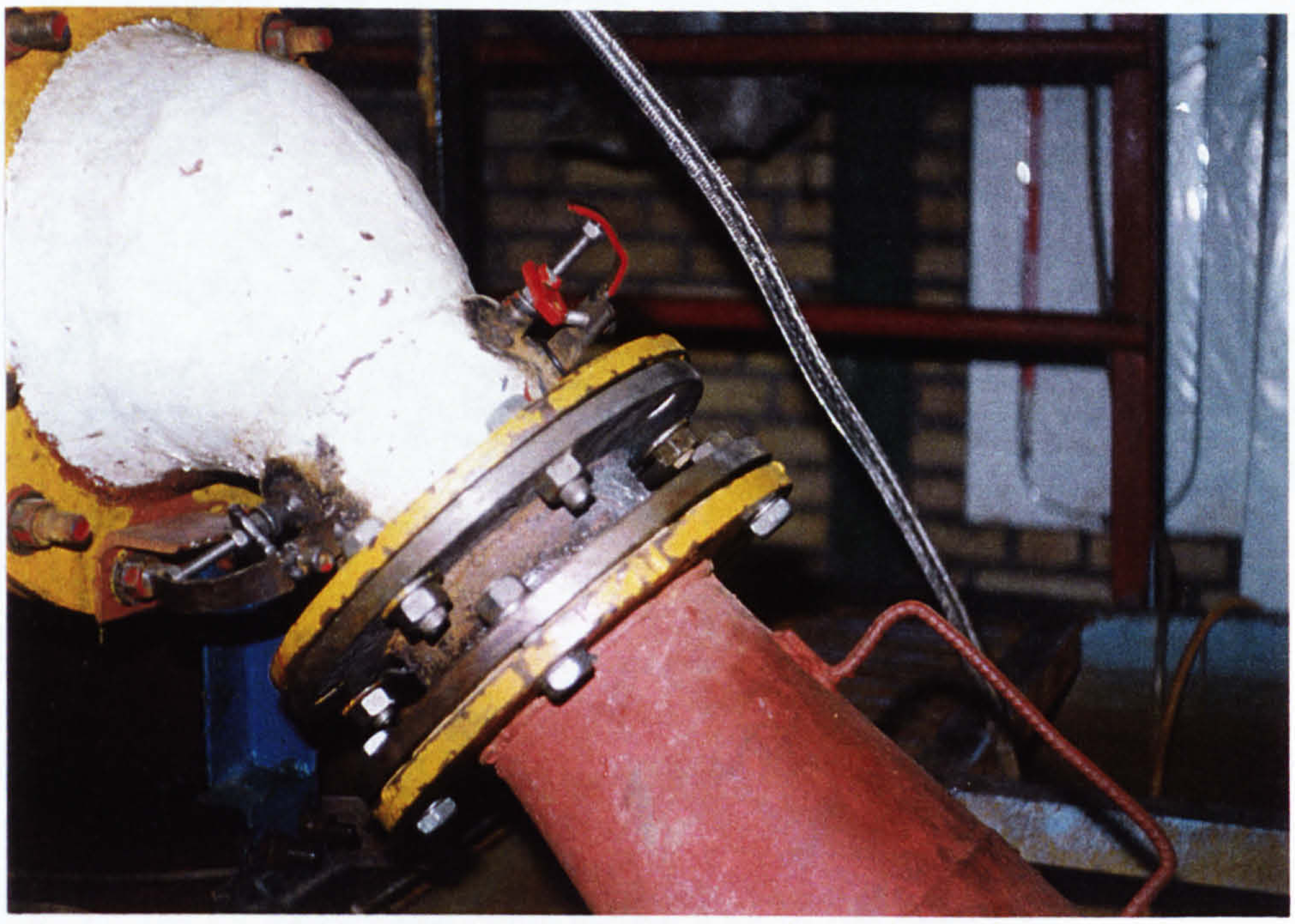


photo ( 4.13 )

## 9. Extension of the turbine exit

A straight hollow cylinder of the same diameter as the turbine exit and 16 (cm) length with flanges at each end was designed and manufactured . This extension is installed between the exit flange of the turbine and entrance flange of the draft tube , to provide the neccessary space for the runner to operate after installation of the guide blade mechanism .Photo (4.14),represents this part.

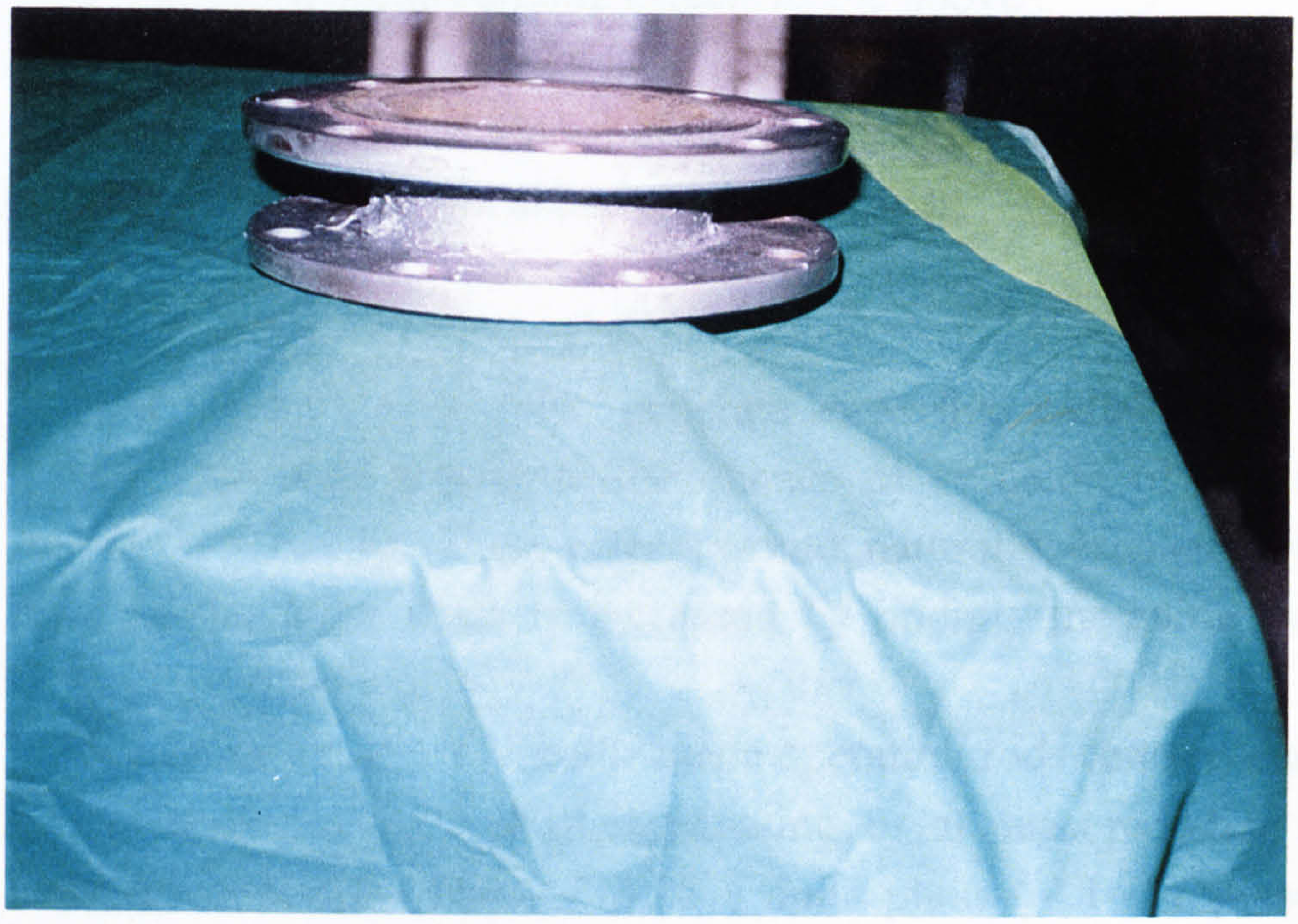


**Photo ( 4.14 )**

## The secondary purpose of the guide mechanism

Basically , the main shaft of the turbine is supported by three bearings inside the bearings housing . Its long free end which also holds the runner is left inside the turbine casing without any supports . There are different factors affecting the normal operation of the main shaft,which are as follows:

**1.** The weight of the runner,which can cause some bending moment in the shaft. This would in turn cause eccentricity between the shaft 's center line and the center line of the casing of the turbine.

2. Load exerted by the bulk of water flowing over the blades . This would cause the same bending effects as the above .

3. Vibrations due to turbulence generated by flow of water through the runner, and minor unbalancing of the runner .

These factors not only generate excessive radial loads on bearings,they can also propagate the chance of collision of running blades against the casing . These effects would ,naturally, escalate as the free length of the main shaft is increased by adding the guide blade holding extension to it.

Introduction of 3 guide blade operating rods can serve as a second support to the main shaft,eleminating or at least reducing the above mentioned effects .Photo ( 4.15 ) and photo ( 4.16 ) represent different views of the assembly of the runner ,connecting rod ,the main shaft's extension and the guide blades mechanism,showing how the mechanism supports the main shaft.

## OTHER MODIFICATIONS

One simple,yet important modification made to the turbine is a vent hole for the purpose of draining out the excess air packets collected inside the turbine. As testing periods were extended,rapid decreases or rather fluctuations in speed ( i.e. power ) was observed frequently. At first it was thought, that the breakage of blades or change of their angle was responsible for such an effect, and naturally, the draft tube had to be dismantled every time to check the blades,and their angle. Since no sign of breakage or angle change was observed then search began for other possible sources of causing such decreases . Eventually , after checking all mechanical factors and obtaining no results, thoughts were diverted towards the flow. The flow was closely

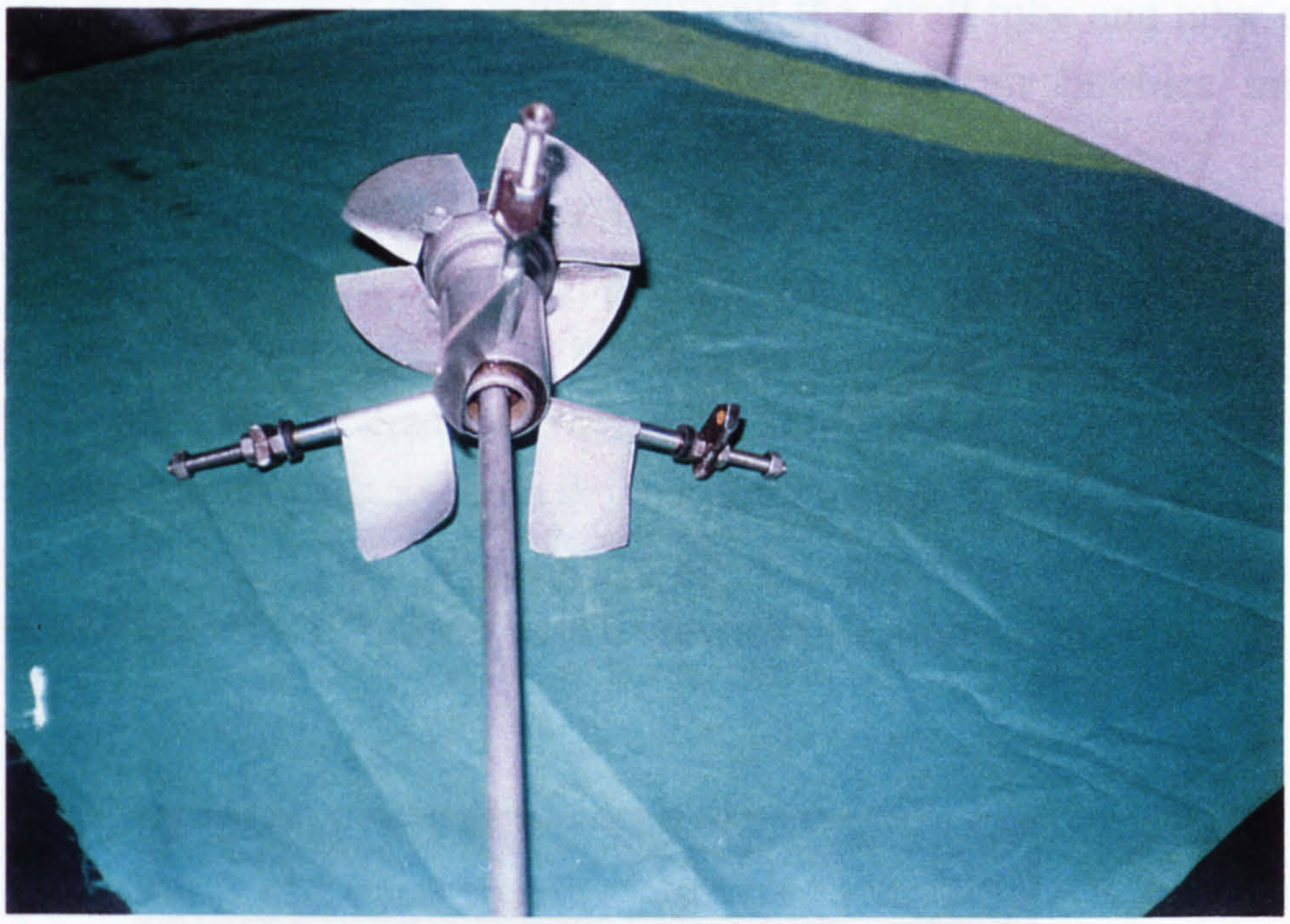


**Photo ( 4.15 )**

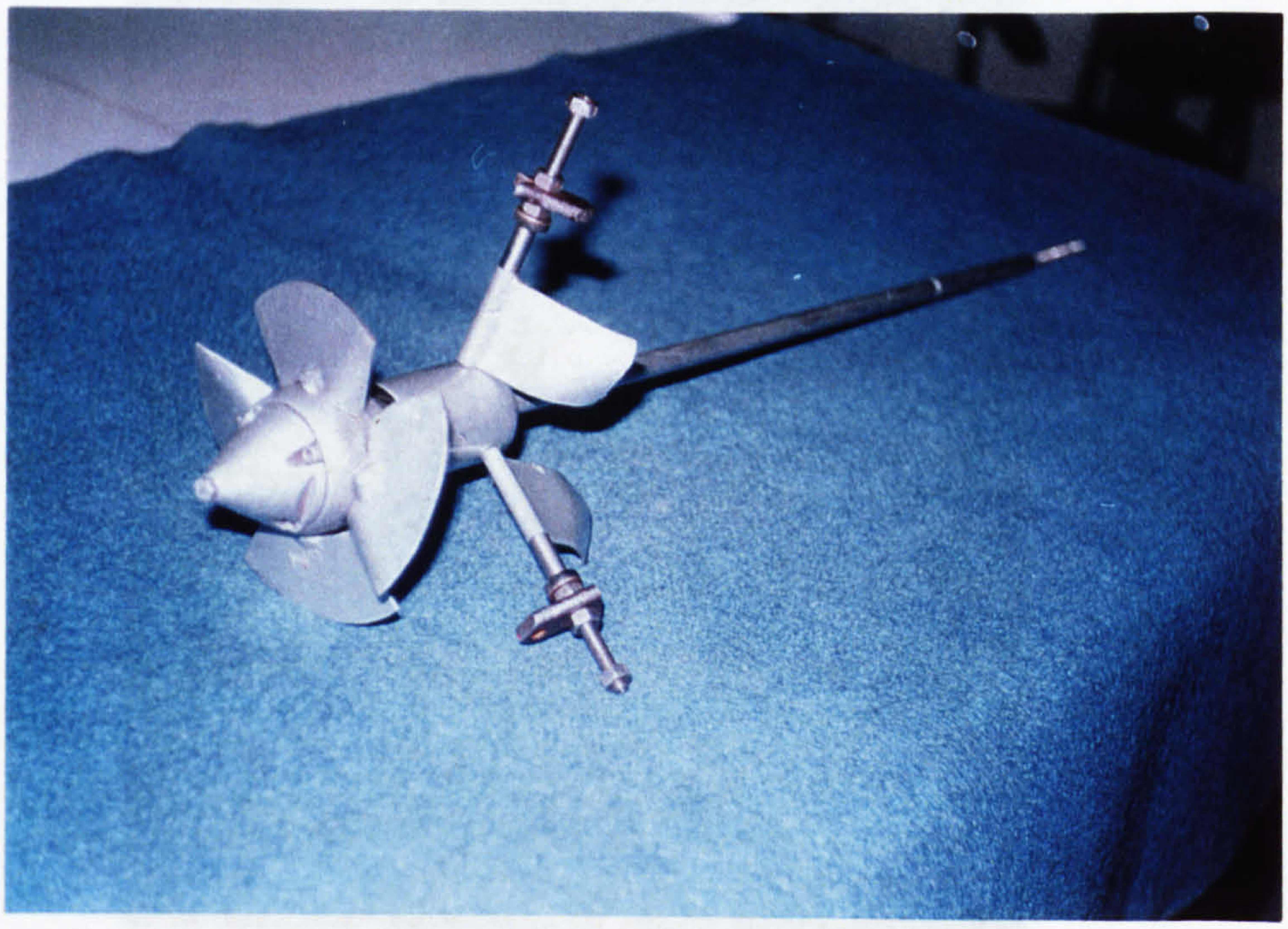


**Photo ( 4.16 )**

observed at the exit of the turbine without the draft tube and later after installing the draft tube. It was then noticed that air bubbles mixed with the flow,were discharged into the reservoir.

It was then concluded that the possible cause of loss of speed could be as the result of air packets mainly caused by cavitation inside the turbine`s casing and it`s interference with flow. Therefore, a vent hole of 10(mm) diameter was made at highest point of the casing. As a result, the air packets were discharged through this hole, and via a flexible pipe into the reservoir, and the problem was completely eliminated.Figure (4.17) represents a view of this modification.

**Photo (4.17)**

# CHAPTER

# ( V )

# TESTING THE TURBINE

## Introduction

In the following chapter various tests which are required to be carried out on the Agnew turbine are presented.These tests are meant to produce information regarding the performance of the unit under different conditions.Such conditions include varying of,load,speed and head.Also these tests are performed to find the best efficiency of the turbine under different conditions.

Also some details on measurement of discharge,torque ,speed and head will be given.

## Procedure of testing the turbine

Real operating conditions for a water turbine is such that, the turbine operates with natural flow (Q) under some head (H).Provision of such conditions in a laboratory is possible by application of a suitable pumping unit. Here, the pumps (1-6) draw water from a reservoir and deliver it under some specified pressure to the turbine. The rate of flow and the head may be varied by starting a pump additional to what is operating at the time, or by opening or shutting off a valve.

How ever, the following steps are to be taken before taking any readings:

*1* . pumps are started and come in to the circuit one by one, by opening their related gate valves.

*2.* The turbine is allowed to run under no load condition for about 10 minutes, on one pump.Then other pumps are started as per requirement of the test, and the turbine is run for another 10 minutes.

*3.* In case of applying hydraulic dynamometer, water inlet to the dynamometer is opened under no load conditions ( i.e. out let fully open )

*4.* The head operating the turbine is kept constant as far as possible.This may be achieved by regulating gate valves.

## Data to be measured

The following data are to be taken during the course of the experiment:

1. Pressure difference caused by the orifice plate to calculate the flow rate (11). Readings are taken from the electronic panel in terms of (mbar).
2. Turbine gauge pressure, which is obtained from the pressure gauge connected to the turbine.These values are read off the electronic panel in terms of (mbar).
3. Torque generated by the turbine,read directly from the electronic panel in terms of (Nm).
4. Turbine speed is measured via an electronic sensor connected to the dynamometer.The reading is also shown on the electronic panel, in terms of rpm.
5. Suction head, measured by the vacuum sensor connected to the draft tube. These values are also taken from the electronic panel.

## Data required for calculations

1. Internal diameter of the pipe at a place where the pressure gauge is connected. Also, the internal diameter of the pipe where the vacuum gauge is connected is to be measured . These diameters are required to calculate the kinetic head of the flowing water in order to determine the effective head working on the turbine.

2. Internal diameter of the pipe (of the piping system ) and the internal diameter of the orifice plate. These two measurements are required for the calculation of the flow rate (11).

## Calculated data at constant head

a.. Turbine effective head,which is the sum of the pressure head of the turbine ( $H_{gauge}$ ),the vacuum pressure head ( $H_{vacuum}$ )and the kinetic head ( $U_i^2 - U_o^2$ )/2g .

i.e. $$H = H_{gauge} + H_{vacuum} + ( U_i^2 - U_o^2 )/2g$$

where $U_i$ = Inlet velocity of water at the pipe cross-section where the pressure gauge is fitted in (m/s).

$U_o$ = exit velocity of water at the pipe cross-section where the vacuum gauge is fitted in(m/s).

These two values may be calculated by using the value of flow rate (Q ).The following would present the relations used to calculate the velocity values.

Since $Q = A \cdot U$

Where $A$ = Cross sectional area of the pipe = $\pi \cdot D^2/4$

Therefore having the values of flow rate (Q) and cross sectional diameter at any point ,the value of the velocity (U) may be calculated by:

$$U = 4 \cdot Q / (\pi \cdot D^2)$$

b. Discharge Q - the discharge is calculated by means of an orifice plate and it is expressed in ($m^3/s$).The following expresses the method applied to calculate the flow rate (11). Figure (5.1)shows flow through a sharp -edged orifice plate.

Here it is seen the streamlines continue to converge a short distance downstream of the plane of the orifice ;hence ,the minimum -flow area is actually smaller than the orifice area.To relate the minimum flow-area ,often called the contracted area of

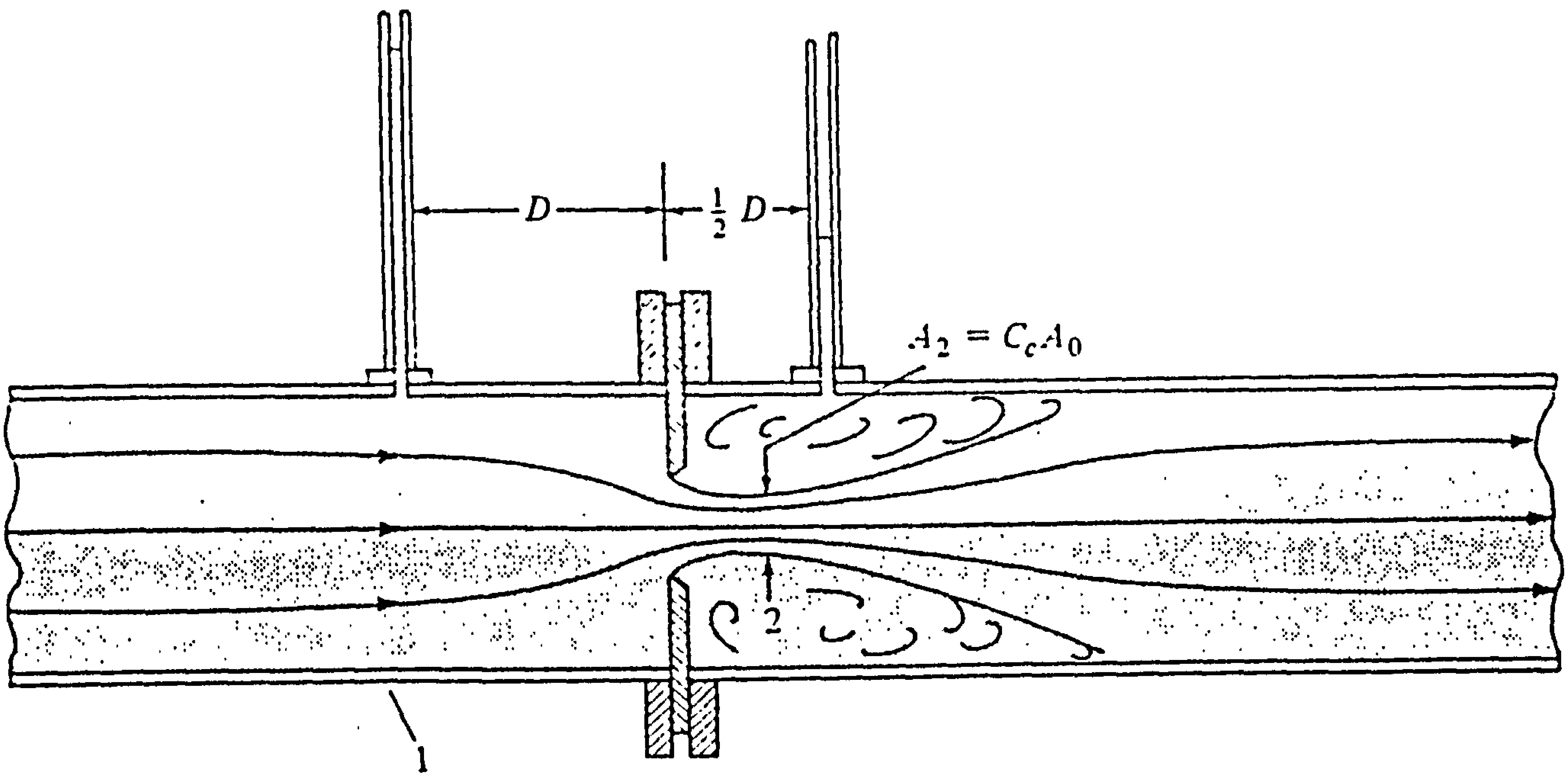



**Figure ( 5.1 )**

the jet or, *vena contracta* ,to the area of the orifice ($A_o$),a contraction coefficient is used which is defined as follows:

$$A_j = C_c . A$$

$$C_c = A_j / A_o$$

Then for a circular orifice ,

$$C_c = \frac{(\pi / 4)\, d_j^2}{(\pi / 4)\, d^2} = ( d_j^2 / d^2 )$$

Since $d_j$ and $d_2$ are identical then ,$C_c = (d_2 / d)$ .At low values of Reynolds number,$C_c$ is function of the Reynolds number;however,at high values of the Reynolds number ,$C_c$ is only a function of the geometry of the orifice .For $d / D$ ratios less than 0.3 , $C_c$ has a value of approximately 0.62 ;however, if $d / D$ is increased to 0.8, $C_c$ increases to a value of 0.72.The derivation of discharge equation for the orifice is started by writing the Bernoulli equation between section 1 and section 2 in figure (5.1).The equation would then be :

$$\frac{P_1}{\rho g} + \frac{U_1^2}{2g} + Z_1 = \frac{P_2}{\rho g} + \frac{U_2^2}{2g} + Z_2$$

Here, $U_1$ may be eliminated by means of the continuity equation

$U_1 . A_1 = U_2 . A_2$ ; and solving for $U_2$ gives,

$$U_2 = \left\{ \frac{2g\,[( P_1/\rho g + Z_1 ) - ( P_2/\rho g + Z_2)]}{1 - ( A_2 / A_1)^2} \right\}^{½} \qquad (1)$$

However, $A_2 = C_c A_0$ and $h = P / \rho g + z$ ; so that equation (1) reduces to

$$U_2 = \frac{2g ( h_1 - h_2 )}{\sqrt{1 - C_c^2 A_0^2 / A_1^2}} \qquad (2)$$

Here,the prime objective is to obtain an expression for discharge in terms of the $h_1$ , $h_2$ and geometric characteristics of the orifice.The discharge is given by $U_2 A_2$ ;hence when we multiply both sides of equation (2) by $A_2 = C_c A_0$ we obtain the desired result:

$$Q = \frac{C_c A_0 \sqrt{[ 2g (h1 - h2 )]}}{\sqrt{1 - C_c^2 A_0^2 / A_1^2}} \qquad (3)$$

Equation ( 3 ) is the discharge equation for the flow of an incompressible inviscid fluid through an orifice.However,this is valid at relatively high Reynolds numbers.For low and moderate values of Reynolds number,viscous effects are significant and an additional coefficient must be applied to the discharge equation to relate the ideal to the actual flow.This is called the *coefficient of velocity* Cv;thus, for viscous flow in an orifice the following discharge equation will hold:

$$Q = \frac{C_v C_c A_0 \sqrt{[2g(h1 - h2)]}}{\sqrt{1 - C_c^2 A_0^2 / A_1^2}} \qquad (4)$$

The product $C_vC_c$ is called the *discharge coefficient* $C_d$ ,and the combination $C_v C_c / (1 - C_c^2 A_0^2 / A^2)^{½}$ is called the flow coefficient K.Thus, $Q = K A_0 \sqrt{2g(h_1 - h_2)}$ ,where

$$K = \frac{C_d}{\sqrt{1 - C_c^2 A_0^2 / A_1^2}} \qquad (5)$$

If $\Delta h$ is defined as $h_1 - h_2$ the final form of the discharge equation for an orifice would then reduce to

$$Q = KA_0 \sqrt{2g\,\Delta h} \qquad (6)$$

Experimentally values of K as a function of d / D and Reynolds number based upon orifice size , $4Q / \pi d_v$, are given in figure ( 5.2 ) .If Q is given $Re_d$ is equal to $4Q / \pi d_v$ , K is obtained from figure ( 5.2 ) ,by using the vertical lines and the bottom scale and

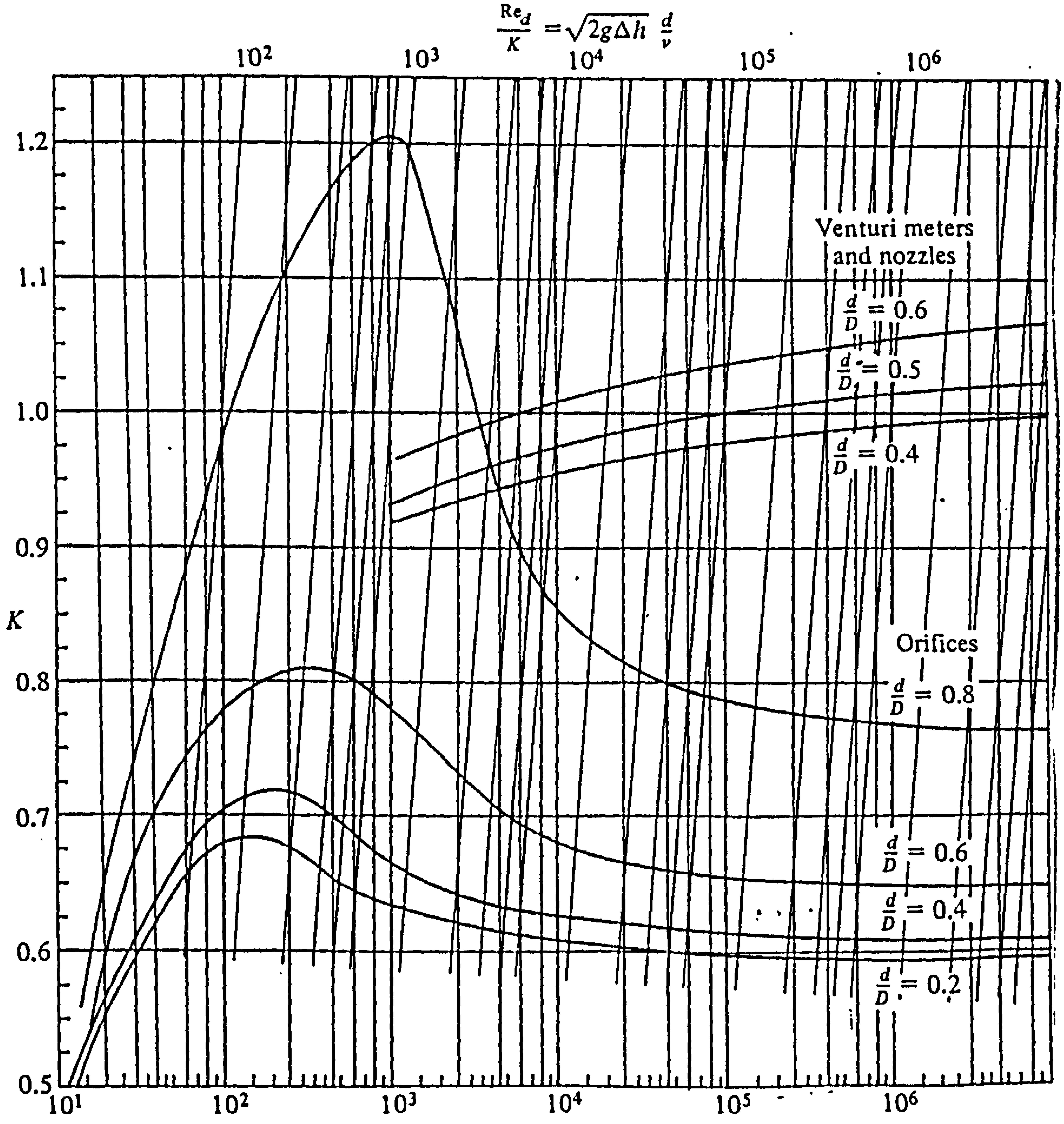

$\frac{Re_d}{K} = \sqrt{2g\Delta h}\ \frac{d}{\nu}$
$10^2$ $10^3$ $10^4$ $10^5$ $10^6$
1.2
1.1
1.0
0.9
K
0.8
0.7
0.6
0.5
$10^1$ $10^2$ $10^3$ $10^4$ $10^5$ $10^6$
Venturi meters
and nozzles
$\frac{d}{D} = 0.6$
$\frac{d}{D} = 0.5$
$\frac{d}{D} = 0.4$
Orifices
$\frac{d}{D} = 0.8$
$\frac{d}{D} = 0.6$
$\frac{d}{D} = 0.4$
$\frac{d}{D} = 0.2$

**Figure ( 5.2 )**

$\Delta h$ is then computed from equation ( 6 ). However the main problem here is to determine discharge Q when a certain value of $\Delta h$ is given. When Q is to be determined there is no direct way to obtain K by entering figure (5.2) with Re , because Re is a function of the flow rate which is still unknown. Hence, another scale which does not involve Q is constructed on the graph of figure ( 5.2 ). The variables for this scale are obtained in the following manner : because $Re_d = 4Q / \pi d\nu$ and $Q = k\ \pi d^2 / 4\sqrt{2g\,\Delta h}$ ,then $Re_d$ may be written in terms of $\Delta h$ as

$$Re_d = K \sqrt{2g\,\Delta h}\quad d / \nu$$

or

$$( Re_d / K ) = \sqrt{2g\,\Delta h}\quad d / \nu$$

Thus the slanted lines and the top scale in figure (5.2 ) are used when $\Delta h$ is known and the flow rate is to be determined .

c . Torque T - read directly from the panel in (Nm). This value is the product of the length of the arm of the dynamometer L and the force exerted by the arm on the load cell, F.

Therefore $T = F . L$ (N m)

To calibrate the torque meter a known weight was connected to the end of the arm of the dynamometer .The product of this weight and the length of the arm ( from the center of the dynamometer to the end of the arm where the weight was connected ) is the expected value to be shown on the panel.In case the value was not as expected then the electronic sensor had to be set to show the expected value.This procedure was repeated a few times with different weights to make sure of certainty of the readings.

**d.** Brake or turbine output power, ($P_o$), is determined as follows:

$P_o = \omega . T . \pi / 30$ (watts)

Turbine speed N (rev/min)

where $\omega$ = speed of the turbine in (rads/s)

**e.** Input power ( $P_i$ )

$$P_i = \rho \, g \, Q \, H \quad \text{(watts)}$$

**f.** Turbine overall efficiency ( $\eta_t$ ).

$$\eta_t = P_i / P_o$$

## Calculated values at unit head

Since bringing a pump into operation additional to what is operating at the time has the same effect as variations of inlet opening of the turbine, therefore the constant head does not have the same value for different number of pumps in operation.Therefore as the usual practice suggests, it is appropriate to convert all the readings to unit head values, which would then produce the constant head for all of them. In this way all the values obtained may be easily compared with one another. For this purpose the following relations were used:

$$Q_1 = Q/(H)^{1/2} \quad , \quad N_1 = N/(H)^{1/2} \quad , \quad T_1 = T/H$$

$$Pi_1 = Pi/(H)^{3/2} \quad , \quad Po_1 = Po/(H)^{3/2}$$

## Testing the base modle turbine

As mentioned earlier it was decided to test the turbine as it was designed, and then introduce improvements and compare the behaviour of the improved turbine with that of the original design.Therefore, tests began according to the procedure mentioned at the beginning of this chapter.

## Trouble shootings

At the beginning of tests everything was operating as normal, but as time passed by, during preparatory long runs and actual early tests, a number of problems arose, which were circumvented appropriately. The following presents the most important ones of these problems:

1. Noise and vibrations due to unbalancing of the hydraulic dynamometer's rotors. ( Discussed in chapters 3 and 4 ). These vibrations caused high levels of fluctuations in all readings, specially at speeds over 800 rpm. However, at first, these were reduced to acceptable levels by applying dampers and shock absorber to the dynamometer and the turbine respectively.Later a mechanical Pronney dynamometer was designed and manufactured and used instead.

2. Breakage and distortion of runner blades, due to collision of blades against the turbine's casing. Therefore , all related parts and joints were then redesigned and improved. ( Discussed in chapter 4). As a result the problem was completely eliminated.

3. Rapid loss of speed - as the testing periods were extended,rapid decreases in speed ( i.e. power ) was observed frequently. At first it was thought, that the breakage of blades or change of their angle was responsible for such an effect, and naturally, the draft tube had to be dismantled every time to check the blades,and their angle. Since no sign of breakage or angle change was observed then search

began for other possible sources of causing such decreases. Eventually, after checking all mechanical factors and obtaining no results, thoughts were diverted towards the flow.The flow was closely observed at the exit of the turbine without draft tube and later after installing the draft tube. It was then noticed that air bubbles mixed with the flow,were discharged into the reservoir.

It was then concluded that the possible cause of loss of speed could be as the result of air packets mainly due to cavitation inside the turbine's casing and it's interference with flow.

Therefore, a vent hole of 10 (mm) diameter was made at highest point of the casing. As a result, the air packets were discharged through this hole, and via a flexible pipe into the reservoir, and the problem was completely eliminated.Photo (4.17) ( chapter 4 ) represents a view of this modification.

# CHAPTER
# ( VI )
# RESULTS

## INTRODUCTION

To appraise the performance of the turbine it is now essential to process the data collected and analyse them through plotting appropriate curves. In previous chapters all the details of the required data were presented and in the following chapter their application will be presented in the form of standard plots.

The turbine is designed in such a way that its runner blades can rotate around their axis of the blade tails between zero to thirty degrees. The turbine was tested for different settings of blades such as zero, ten, fifteen, twenty and thirty degrees. For the purpose of comparison only results of two settings of fifteen and thirty degrees were selected.

Therefore the following conditions for the original turbine are presented

A. Angle of runner blades is 15 degrees, which is described as "semi-open" blades.

B. Angle of runner blades is 30 degrees, which is described as "open" blades.

After installation of guide vanes the same tests were performed on the turbine, for different settings of runner blades (i.e. open and semi-open) in conjunction with various settings of guide vanes. Guide vanes can rotate around their axis by ten degrees. Zero degree guide vane would mean that the blades are along the line of flow and ten degrees guide vanes mean that vanes are rotated by ten degrees clockwise in the direction as the subsequent rotation of the impeller. In this condition the flow is diverted by ten degrees by the guide vanes. Therefore the ultimate settings after modification are as follows:

C. Runner open blades with zero degrees angle of the guide vanes
D. Runner open blades with ten degrees angle of the guide vanes
E. Runner semi-open blades with zero degrees angle of the guide vanes
F. Runner semi-open blades with ten degrees angle of the guide vanes

For all six (two original and four modified turbine) above situations the results are presented for three conditions:

1. Constant head
2. Constant speed
3. Constant efficiency (which are extracted from 1 & 2)

The results presented in this chapter for the above six situations are unit flow, torque, power and efficiency against unit speed. The operating point for each condition is also presented.

To clarify the best conditions, the comparison of the results is shown.

Due to the important role of the draft tube in overall performance of the turbine ,its behavior was under close scrutiny all through the testing process . For this purpose the turbine was tested with the draft tube as a complete cone and without it. Later the exit cross section of the draft tube was cut at an angle of 45 ° to the center line of the cone so that its exit cross section stood horizontal and parallel to the free surface of water.

The outcomes of testing the turbine without the draft tube were, splashing high speed water,deafening noise, relatively poor performance of the turbine and considerable vibrations,as expected.

When the draft tube was tested as a complete cone although the performance of the turbine was recovered and the noise and vibrations diminished yet high levels of turbulence were observed at the exit of the tube just below the water free level.This turbulence caused considerable fluctuations in all readings.The main cause of the turbulence was the fact that , the tube was of complete cone shape and subtended 45° to the line of horizon.Therefore the water exiting the tube did not discharge into a constant pressure medium.Here,at the upper parts of the exit cross section of the tube,water was discharged into relatively lower depth of the pool and at the lower ends of the cross section, water was discharged into deeper levels of the pool.Therefore ,the outcoming flow had the tendency to discharge into parts with lower pressure and this pushed the flow into higher parts of the exit of the draft tube,causing turbulence at the exit.

At the end the exit of the draft tube was cut horizontally ,so that water was discharged into the pool at a uniform pressure level across the exit cross section. Therefore the turbulence at the exit completely diminished.

As indicated before , six pumps have been installed at the test stand.For obtaining various head and flow conditions different number of pumps came into operation in each test.As the prime objective of these tests was to ascertain the behavior of the turbine for heads below ten meters therefore higher heads had to be ignored.Although all tests were taken for one to six pumps ,but experience showed that the heads obtained for more than four pumps in operation, exeeded ten meters.For this purpose only the data for upto four pumps are presented here and the rest are available at the laboratory's library.

## RESULTS

The following presents the results of the tests in the form of appropriate plots in order of blades settings.These results include the main characteristics ( constant head ),operating characteristics ( constant speed ) and constant efficiency plots,also comparison of these results are presented where appropriate.

In the following, the figures are numbered in two and three parts numbers.The two parts numbers are not coded but the three parts numbers are coded in such a manner that the last digit on the right refers to the related blades setting as indicated at the beginning of each heading for blades settings with the same number.

### *1. OPEN BLADES*

Figures ( 6.1.1 ) to ( 6.4.1 ) show the main characteristics or constant head curves ,including unit flow,unit torque,unit power and efficiency against unit speed, for one to four pumps in operation respectively. Figures ( 6.5.1 ) to ( 6.8.1 ) show a comparison of the above characteristics.Figures ( 6.9.1 ) to ( 6.14.1 ) show the operating characteristics or constant speed curves.Figure ( 6.9.1 ) shows the efficiency v power curves for speeds of 500 r.p.m and 1000 r.p.m.Figures

( 6.10.1 ) to ( 6.12.1 ) represent the efficiency and power plots against the flow rate,indicating the operating point of the turbine, for 500, 1000 and 1300 r.p.m. respectively .Figure ( 6.13.1 ) shows curves of efficiency v percentage of full load for constant speeds of 500,and 1000 r.p.m.On figure ( 6.14.1 ) the constant efficiency curves ,of power v speed are shown.

## *2. SEMI OPEN BLADES*

Figures ( 6.1.2 ) to ( 6.8.2 ) show the plots of main characteristics.On each figures ( 6.1.2 ) to ( 6.4.2 ) four plots of unit flow,unit torque ,unit power and efficiency against unit speed are shown for one to four pumps respectively.Figures ( 6.5.2 ) and ( 6.6.2 ) represent comparative graphs of unit flow and unit torque against unit speed for one to four pumps.Also Comparative curves of unit power and efficiency against unit speed are shown on figures ( 6.7.2 ) and (6.8.2 ).Operating characteristics plots are represented by figures ( 6.9.2 ) to ( 6.13.2 ). Curves of efficiency v power for constant speeds of 500,1000 and 1300 r.p.m. are shown on figure ( 6.9.2 ).Three plots of efficiency and power against flow rate are represented by figures ( 6.10.2 ) to ( 6.12.2 ) for constant speeds of 500,1000 and 1300 r.p.m. respectively.Also curves of efficiency v percentage of full load for the same constant speeds are shown on figure ( 6.13.2 ).The constant efficiency curves are represented by figure ( 6.14.2 ) on a base of power v speed.

## *3.OPEN BLADES WITH 0° GUIDE VANES*

Plots of main characteristics are shown by figures ( 6.1.3 ) to ( 6.4.3 ) ,for one to four pumps respectively.Comparative plots of these main characteristics for unit flow, unit torque, unit power and efficiency against unit speed for one to four pumps are presented in figures ( 6.5.3 ) to ( 6.8.3 ),respectively.Plots of the operating characteristics are shown by figures ( 6.9.3 ) to ( 6.13.3 ).Curves of efficiency v power for constant speeds of 500,1000 and 1300 r.p.m. are shown by figure ( 6.9.3 ).Plots of efficiency and power against flow rate for the same constant speeds are represented in figures ( 6.10.3 ) to ( 6.12.3 ). Figure ( 6.13.3 ) shows the efficiency v percentage of full load curves for constant speeds of 500,1000 and 1300 r.p.m. on the same plot.Figure ( 6.14.3 ) shows the constant efficiency curves on the base of power against speed.

### *4.OPEN BLADES WITH 10° GUIDE VANES*

Figures ( 6.1.4 ) to ( 6.8.4 ) show the plots of main characteristics.On each of figures ( 6.1.4 ) to ( 6.4.4 ) four plots of unit flow,unit torque ,unit power and efficiency against unit speed are shown for one to four pumps respectively.Figures ( 6.5.4 ) and ( 6.6.4 ) represent comparative graphs of unit flow and unit torque against unit speed for one to four pumps.Also Comparative curves of unit power and efficiency against unit speed are shown on figures ( 6.7.4 ) and ( 6.8.4 ).Operating characteristics plots are represented by figures ( 6.9.4) to ( 6.13.4 ). Curves of efficiency v power for constant speeds of 500,1000 and 1300 r.p.m. are shown on figure ( 6.9.4 ).Three plots of efficiency and power against flow rate are represented by figures ( 6.10.4 ) to ( 6.12.4 ) for constant speeds of 500,1000 and 1300 r.p.m. respectively.Also curves of efficiency v percentage of full load for the same constant speeds are shown on figure ( 6.13.4 ).The constant efficiency curves are represented by figure ( 6.14.4 ) on the base of power v speed.

## 5.SEMI OPEN BLADES WITH 0° GUIDE VANES

Figures ( 6.1.5 ) to ( 6.4.5 ) show the main characteristics or constant head curves ,including unit flow,unit torque,unit power and efficiency against unit speed, for one to four pumps in operation respectively. Figures ( 6.5.5 ) to ( 6.8.5 ) show a comparison of the above characteristics.Figures ( 6.9.5 ) to ( 6.14.5 ) show the operating characteristics or constant speed curves.Figure ( 6.9.5 ) shows the efficiency v power curves for speeds of 500 r.p.m and 1000 r.p.m.Figures ( 6.10.5 ) to ( 6.12.5 ) represent the efficiency and power plots against the flow rate,indicating the operating point of the turbine, for 500, 1000 and 1300 r.p.m. respectively .Figure ( 6.13.5 ) shows curves of efficiency v percentage of full load for constant speeds of 500,and 1000 r.p.m.On figure ( 6.14.5 ) the constant efficiency curves ,of power v speed are shown.

## 6.SEMI OPEN BLADES WITH 10° GUIDE VANES

The constant head curves of unit flow,unit torque,unit power and unit efficiency against unit speed are represented by figures ( 6.1.6 ) to ( 6.4.6 ) ,for one to four pumps respectively.On each figure all four main characteristics plots are shown .Comparison curves of these characteristics are shown by figures ( 6.5.6 ) to ( 6.8.6 ) for one to four pumps on single plots. Plots of the operating characteristics are shown by figures ( 6.9.6 ) to ( 6.13.6 ).Curves of efficiency v power for constant speeds of 500,1000 and 1300 r.p.m. are shown by figure ( 6.9.6 ).Plots of efficiency and power against flow rate for the same constant speeds are represented in figures ( 6.10.6 ) to ( 6.12.6 ). Figure ( 6.13.6 ) shows the efficiency v percentage of full load curves for constant speeds of 500,1000 and 1300 r.p.m. on the same plot.Figure ( 6.14.6 ) shows the constant efficiency curves on the base of power against speed

## *7.COMPARISON CURVES*

In order to compare the behavior of the turbine before and after modifications, comparison curves of different conditions were plotted.Figures ( 6.15 ) to ( 6.29 ) represent comparisons of main characteristics curves ( unit speed ) .Six plots of different blades settings are shown on each figure,for one to four pumps,separately.

Figures ( 6.15 ) to ( 6.18 ) show the comparison curves of unit flow ,unit torque, unit power and efficiency against unit speed respectively,for one pump.The same comparisons of characteristics plots are represented by figures ( 6.19 ) to ( 6.22 ) for two pumps. Figures ( 6.23 ) to ( 6.26 ) and figures ( 6.27 ) to ( 6.30 ) represent the comparison characteristics for all different blades settings ,for three and four pumps respectively.

The study of the above plots could indicate the best performing blades settings before and after installation of the guide blades.To generate a vivid comparative picture of the performance of the turbine new sets of plots were produced for the best performing blades settings as mentioned above.Figures ( 6.31 ) to ( 6.34 ) represent these plots,for one to four pumps respectively.On each plot two curves of efficiency v unit speed for semi open blades ( before modification ) and semi open blades with zero degree guide blades ( after modification ) are shown.

Figures ( 6.35 ) to ( 6.37 ) show comparison curves of operating characteristics for constant speeds of 500, 1000 and 1300 r.p.m.

respectively.On each plot six curves of efficiency v power are shown for each of the six different blades settings.Also figures ( 6.38 ) to ( 6.40 ) represent the same comparison curves for the two best performing blades settings for the same constant speeds.

Curves of efficiency v percentage of full load for all six blades setting are compared on figures ( 6.41 ) to ( 6.43 ) for 500, 1000 and 1300 r.p.m respectively. Also figures ( 6.44 ) to ( 6.46 ) represent the same comparison curves for the two best performing blades settings for the same constant speeds.

## *8. THE DRAFT TUBE*

For the purpose of appraising the behavior of the draft tube curves of its efficiency v flow rate were plotted.Although twenty four plots,for six different blades settings and one to four pumps in operation were originally generated but a number of them were selected for representation purposes.Figure ( 6.47 ) for one pump,open blades and figure ( 6.48 ) for two pumps, semi open blades, reflect the behavior of the draft tube before installation of the guide blades for relatively low to moderate flow rates.Figures ( 6.49 ) and ( 6.50 ) represent the plots for semi open blades with zero and ten degrees guide blades ,for three pumps respectively.Figure ( 6.51 ) shows the same plot for open blades with ten degrees guide vanes for three pumps. Also the plot for open blades with ten degrees guide blades for four pumps, is represented by figure (6.52 ).

# DISCUSSIONS AND CONCLUSIONS

The following would present some discussions on the plots earlier presented.Here an attempt is made to appraise the general behavior of the turbine and compare it with usual behavior of axial flow turbines.Also its general behavior along with its performance characteristics would be compared for before and after installation of guide vanes.

## DISCUSSIONS

Constant head characteristics of the turbine for open blades are shown by figures ( 6.1.1 ) to ( 6.4.1 ).In comparison with the usual characteristics of axial flow turbines a complete agreement is observed between the obtained results for Agnew turbine and those of axial flow turbines.

Figures ( 6.5.1 ) to ( 6.8.1 ) show the comparative characteristics for open blades.On figure ( 6.5.1 ) a reduction of flow with respect to speed ( unit flow ) is observed as the number of pumps increase.This is mainly due to the head increase as a result of increase in number of pumps in operation.Figure ( 6. 6 .1 ) shows a different torque generating behavior for the turbine against variations of flow rate.This would be naturally reflected in power generating characteristic of the turbine shown by figure ( 6. 7.1 ) .In these two figures highest levels of torque and power , relative to speed variations ,have been obtained when two pumps were in operation. Finally from figure ( 6.8.1 ) it may be observed that the highest efficiency has been obtained during operation of two pumps.As the flow rate increases ( increase in number of pumps ,over two pumps ) values for efficiency tend to decrease,moderately.Although wider ranges of speed is covered for three and four pumps in operation ,but values for efficiency at these areas are not very high and rapidly fall as speed increases.

Figures (6.1.2 ) to ( 6.4.2 ) represent comparisons of main characteristic plots for semi open blades.General behavior of these plots are almost the same as those of open blades.The major difference may be noticed in figure ( 6.8.2 ),and that is the fact that ,the efficiency curve for two pumps in operation covers the widest range of unit speed amongst all curves as well as showing the highest efficiency obtained.

Comparative plots of characteristics for open blades with zero degree guide vanes are shown in figures ( 6.1. 3 ) to ( 6.4.3 ).In figure ( 6.5.3 ) the same trend as before is observed for unit flow curves. Highest values of torque with respect to unit speed have been obtained when two and three pumps were in operation ( figure (6.6.3 )).Values for unit power generated by two and three pumps are almost the same and so are the efficiencies gained by them, figures ( 6.7.3 ) and (6.8.3). However,as may be visualized from figure ( 6.8.3 ),values of efficiency for 2,3 and 4 pumps in operation are very close ,with three pumps,producing highest efficiencies.

Comparative characteristics of the turbine for open blades with ten degree guide vanes are shown by figures ( 6.5.4 ) to ( 6.8.4 ).Unit flow and unit torque curves show the same trend as for zero degree guide vanes.Highest values of unit torque are obtained for two and three pumps,although their values are very close.However,unit power curves show clear difference between two and three pumps,with two pumps generating highest amounts of unit power.This is where the unit power curves for three and four pumps are very close with the four pumps curve covering a wider unit speed range,figure ( 6.7.4 ).From figure ( 6.8.4 ) it may be observed that the highest levels of efficiency have been attained when two pumps were in operation ,where efficiency curves for three and four pumps are very close to one another.

Figures ( 6.5.5 ) to ( 6.8.5 ) show comparative characteristics of the turbine for semi open runner blades with zero degree guide vanes. In figures ( 6.5.5 ) and (6.6.5 ) the usual trend for unit flow and unit torque curves is observed.From figure ( 6.7.5 ) it may be deduced that the highest values for the unit power have been generated by two and three pumps,whereas,the levels of unit power fell rapidly as the number of pumps increased to four.Highest efficiencies have also been obtained when two and three pumps were in operation,according to figure ( 6.8.5 ).At the highest point the two curves seem to be very close ,whereas,at other points efficiency levels attained by two pumps are clearly higher than the rest of other pumps.As the number of pumps increase the efficiency of the turbine tend to fall.

Comparative characteristics curves of semi open blades with ten degrees guide vanes are shown by figures ( 6.5.6 ) to ( 6.8.6 ).Here,again the same trend as before is shown for unit flow and unit torque graphs,figures (6.5.6 ) and (6.6.6 ).Also as it may be observed from figures (6.6.7 ) and (6.8.6 ),highest levels of unit power and efficiency are obtained when two and three pumps were in operation,and their fall can clearly be observed as the fourth pump comes into operation,exactly the same as semi open blades with zero degree guide vanes.

As mentioned before figures (6.15 ) to ( 6.30 ) represent comparisons of characteristics curves for all six different blades settings, on one plot for 1, 2 , 3 and 4 pumps in operation, separately.

From figures ( 6.15 ) to ( 6.18 ) it may be observed that semi open blades with guide vanes have always had the lowest amount of unit flow rate in comparison with other blades settings.Figures ( 6.19 ) to ( 6.22 ) show that the highest amount of unit torque has always been obtained by open blades with zero degree guide vanes.This would explain the high amount of unit power generated by the same blade setting shown on figures (6.23 ) to ( 6.26 ).From figures ( 6.27 ) to ( 6.30) it is clearly observed that the turbine having the guide vanes has always reached the highest levels of efficiency for any of four combinations of blades settings.Figures ( 6.31 ) to (6.34 ) select the best performance conditions before and after installation of guide vanes ( i.e. semi open blades, and semi open blades with zero degree guide vanes).

It has already been mentioned that operating characteristics are basically used for the purpose of appraising the power generating capability of the turbine. The following continues the discussion and lead it to operating characteristics ( constant speed ) of the turbine.

The following figures show plots of efficiency against power for different constant speeds.Selected constant speeds are 500,1000 and 1300 r.p.m.On figure ( 6.9.1 ) only two curves of 500 r.p.m. and 1000 r.p.m. can be observed.This is due to the fact that there were not enough data points for 1300 r.pm. to generate the appropriate curve, since the turbine with open blades for one and two pumps in operation could not attain the speed of 1300 r.p.m.On this plot the values of efficiency gained by the turbine for any amount of power for higher speed of 1000 r.p.m. have always been more than those of 500 r.p.m. speed.In case of semi open blades ,figure ( 6.9.2 ), the highest speed of 1300 r.p.m. has always had the highest values of efficiency for all values of power.Curves of 500 r.p.m and 1000 r.p.m are very close at lower amounts of power and for higher values of power the 1000 r.p.m. curve shows higher values of efficiency, as well as much wider range of power generated. On figure ( 6.9.3 ),the plot for open blades with zero degree guide vanes, both curves of higher speeds clearly show much higher values of efficiency than those of 500 r.p.m. curve.These two curves are also very close to one another.For open blades with ten degree guide vanes,conditions of curves are almost the same and the only difference is in wider gap between the two higher speed curves.Here the 1300 r.p.m curve shows the highest values of efficiency against power , figure ( 6.9.4 ). Plots for both semi open blades with zero and ten degree guide vanes show almost the same behavior all through, for all three curves. Here again the higher speed curves show highest values of efficiency at all times.As it may be observed from both plots, the 1000 r.p.m. curve show higher values of efficiency almost up to half way through the values of power.There after,the values of efficiency for 1300 r.p.m. curve clearly exceed those of 1000 r.p.m. and also cover a wider range of power generated ,figures ( 6.9.5 ) and ( 6.9.6 ).

Figures ( 6.35 ) to ( 6.37 ) show comparisons of (efficiency v power ) for all different blades settings on one plot,separately for 500, 1000, and 1300 r.p.m speeds. In all three cases ,curves are very close for lower values of power but they divert as the values of power increase. However, the curve for semi open blades with zero degree guide vanes shows higher values of efficiency against higher values of power for all three cases.

Selected curves of efficiency v power for best performance conditions before and after installation of guide blades for above mentioned constant speeds are represented by figures ( 6.38 ) to ( 6.40). In all three cases ,curves of semi open blades with zero degree guide blades clearly exceed those of semi open blades both in terms of efficiencies gained and the power ranges.

The second set of constant speed plots show the plots of efficiency v flow rate and power v flow rate both on one plot for various blades settings and for different constant speeds.Blades settings and constant speeds are the same as indicated above.On each plot an intersection point between the graph of power and the curve of efficiency is observed,which may be regarded as the "*operating point*" of the turbine for that specific condition.

Study of these operating points show that in all cases highest operating points belong to the turbine with the guide blades.Comparison of operating points for best performance blades settings before and after installation of guide blades show that for the semi open blades, the turbine has gained an efficiency of 49% with a generated power of 3150 (w) for a flow rate of 65 (l/s). These values for the semi open blades with zero degree guide blades are 78% , 4600 (w) ,and 56.5 (l/s) respectively.Values of the latter also represent the highest operating point amongst all.

Next set of plots of constant speed plots represent curves of efficiency v percentage of load for each blades setting separately,figures ( 6.13.1 ) to (6.13.6 ). On figure (6.13.1 ) two curves for 500 and 1000 r.p.m. for open blades are observed.Here the highest values of efficiency have been obtained for 1000 r.p.m speed.On figure ( 6.13.2 ) plots for semi open blades are shown.Here again the values of efficiency for speed of 1000 r.p.m are the highest,whereas those of 1300 r.p.m. are the lowest for up to 50% of the load and after that they exceed the values of efficiency for 500 r.p.m. On figure ( 6.13.3 ) it can clearly be observed that the values of efficiency for open blades with zero degree guide blades for 1300 r.p.m are the highest and then come those of 1000 r.p.m. and the lowest ones are of 500 r.p.m. .Open blades with ten degrees guide vanes show a different behavior ,here, highest values of efficiency are gained for the speed of 1000 r.p.m and then come those of 1300 r.p.m. and then 500 r.p.m as the lowest ,figure ( 6.13.4 ).Plots for semi open blades with zero and ten degree guide vanes show almost the same behavior.In both cases loads of more than 70 % gain higher values of efficiency for 1300 r.pm. and below that 1000 r.p.m. speed obtains higher values of efficiency.Therefore these two curves cross each other at around 70 % of the load.Also values of efficiency for zero degree guide vanes are clearly higher than those of ten degree guide vanes, figures ( 6.13.5 ) and ( 6.13.6 ).

Figures ( 6.41 ) to ( 6.43 ) provide a comparative illustration of the above curves for different constant speeds. From figure ( 6.41 ) it can be observed that for low speeds ( i.e. 500 r.p.m. ) the turbine with open blades and zero degree guide vanes attain the highest values of efficiency at all times,apart from last ten percent of the full load (i.e. from 90 % to 100 % ) where the efficiency values for semi open blades with zero degree guide vanes exceed all others.Figures ( 6.42 ) and ( 6.43 ) show clear dominance of the curves of semi open blades with zero and ten degrees guide blades .In almost the last 10 % of the full load they fall slightly below the values of open blades with zero degree guide blades.Figures ( 6.44 ) to ( 6.46 ) show the efficiency v percentage of full load curves for selected best performance blades settings before and after modification, for different constant speeds. Figure ( 6.44 ) shows how both blades settings behave very closely for loads below 50 % and over that the values of semi open blades with zero degree guide blades exceed those of semi open blades, for 500 r.p.m speed. For 1000 r.p.m speed, a clear wide gap is visualized between the two curves with semi open blades with zero degree guide blades having higher values. This gap reduces as the load percentage increases, figure ( 6.45 ). For 1300 r.p.m speed again a gap is observed between the two curves .Here the gap is much wider than that of 1000 r.p.m. speed and again it tends to become narrower towards higher load percentages,figure ( 6.46 ). These three plots draw a clear picture of the performance of the turbine under its best performing conditions before and after introduction of guide blades.

Figures ( 6.14.1 ) to ( 6.14.6 ) represent the constant efficiency curves for various blades settings.Basically these curves help detect the highest efficiency zones.Also with the help of these curves it would be possible to determine the best operating range of the turbine , for specified ranges of power and speed. A study of the above plots show that in the best operating range with 81 % efficiency, the generated power would vary between 1000 (w) to 4600 (w) for a speed range of 800 r.p.m to 1700 r.p.m., respectively. These values are related to the semi open blades with zero degree guide vanes turbine,illustrated by figure ( 6.14.5 ) . The plot for best working blades setting before modification ( i.e. semi open blades ) shows 61 % efficiency for a power generating range of 700 (w) to 3800 (w) for a speed range of 600 r.p.m. to 1850 r.p.m., figure ( 6.14.2 ).

## EFFICIENCY OF THE DRAFT TUBE

In order to obtain an apraisal of the performance of the draft tube, curves of its efficiency against the flow rate were plotted. A number of these plots have been selected and are represented by figures ( 6.47 ) to ( 6.52 ), for different blades settings and varios number of pumps in operation. Although lower values of efficiency are observed when one pump is in operation, yet, still the values obtained for the efficiency of the draft tube are acceptable, which are mostly over 75% and around 80% to 85%, figure ( 6.47 ). In all other cases when more than one pump are in operation the efficiency is in the range of 80% to 90% which is quite acceptable,figures ( 6.48 ) to (6.52 ).

## QUICK GUIDE TO THE PLOTS

The first six set of plots are related to six different blades settings, representing, the main characteristics for one to four pumps, camparison curves of the main characteristics of one to four pumps on one plot, operating characteristics including, efficiency v power, efficiency and power against the flow rate on one plot, efficiency v percentage of full load, and the constant efficiency curves. These plots are represented as follows:

*1.* **Open blades:** figures (6.1.1 ) to ( 6.14.1 ), pages 6.20 to 6.33 .

*2.* **Semi open blades:** figures (6.1.2 ) to ( 6.14.2 ), pages 6.34 to 6.47 .

*3.* **Open blades + 0° guide vanes:** figures (6.1.3 ) to ( 6.14.3 ) pages 6.48 to 6.61 .

*4.* **Semi open blades +10° guide vanes:** figures (6.1.4 ) to ( 6.14.4) pages 6.62 to 6.75 .

*5.* **Semi open blades +0° guide vanes:** figures (6.1.5 ) to ( 6.14.5) pages 6.76 to 6.89 .

*6.* **Semi open blades +10° guide vanes:** figures (6.1.6 ) to ( 6.14.6) pages 6.90 to 6.103 .

Figure (6.15 ) to (6.30 ) illustrate comparison graphs and curves of main characteristics for all blades settings simultanously for one to four pumps, pages 6.104 to 6.119. Comparison of the efficiency curves against unit flow for one to four pumps, pages 6.120 to 6.123,figures (6.31 ) to (6.34 ).

Figures (6.35 ) to (6.40 ) illustrate comparative constant speed plots of efficiency v power for different blades settings simultanously, pages 6.130 to 6.135.

Figures ( 6.47 ) to ( 6.52 ) represent plots of efficiency v flow rate for the draft tube, pages 6.136 to 6.141 .

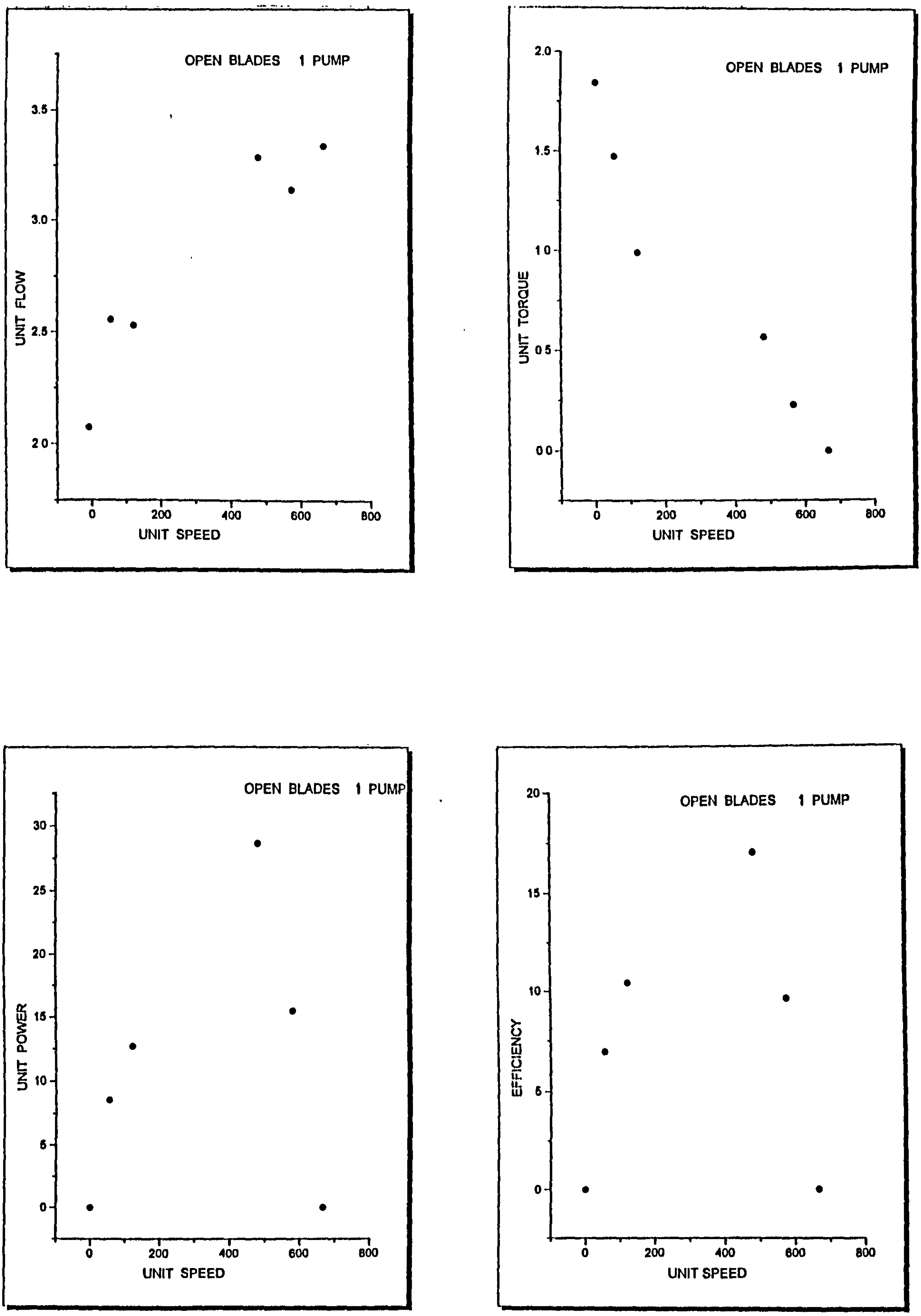

OPEN BLADES 1 PUMP
UNIT FLOW
UNIT SPEED
OPEN BLADES 1 PUMP
UNIT TORQUE
UNIT SPEED
OPEN BLADES 1 PUMP
UNIT POWER
UNIT SPEED
OPEN BLADES 1 PUMP
EFFICIENCY
UNIT SPEED

Figure ( 6.1.1 )

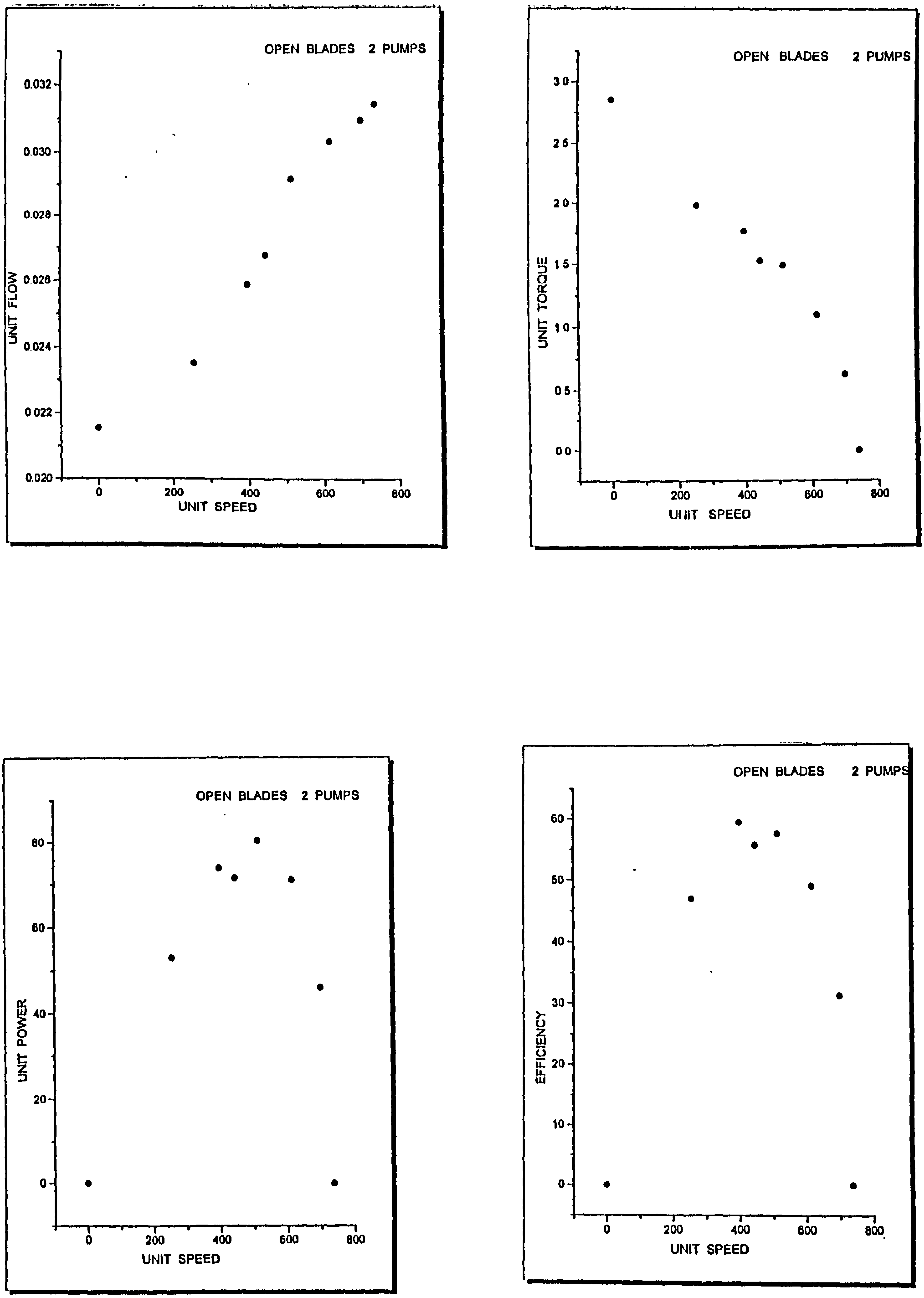

OPEN BLADES 2 PUMPS
UNIT FLOW
UNIT SPEED
OPEN BLADES 2 PUMPS
UNIT TORQUE
UNIT SPEED
OPEN BLADES 2 PUMPS
UNIT POWER
UNIT SPEED
OPEN BLADES 2 PUMPS
EFFICIENCY
UNIT SPEED

Figure ( 6.2.1 )

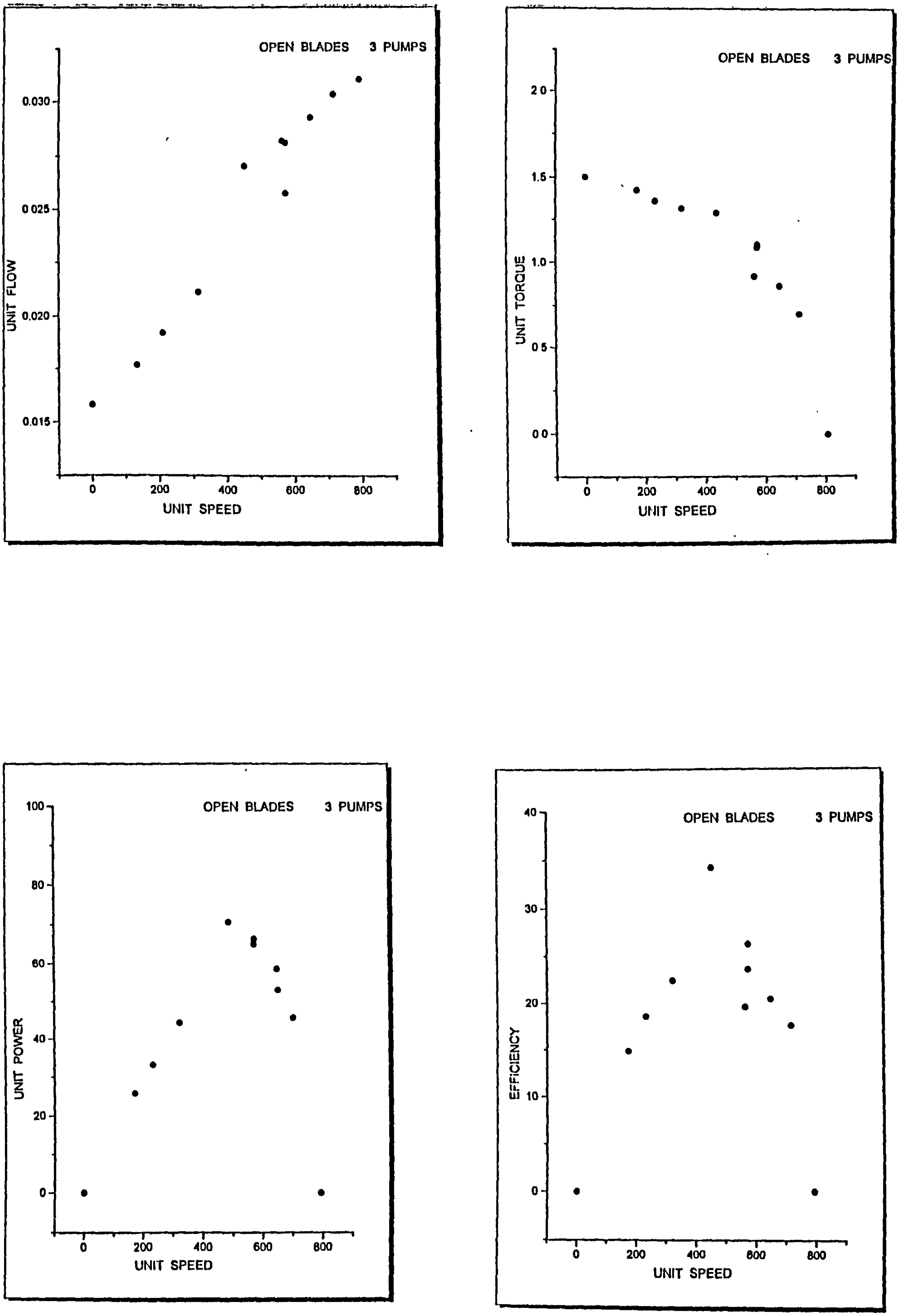
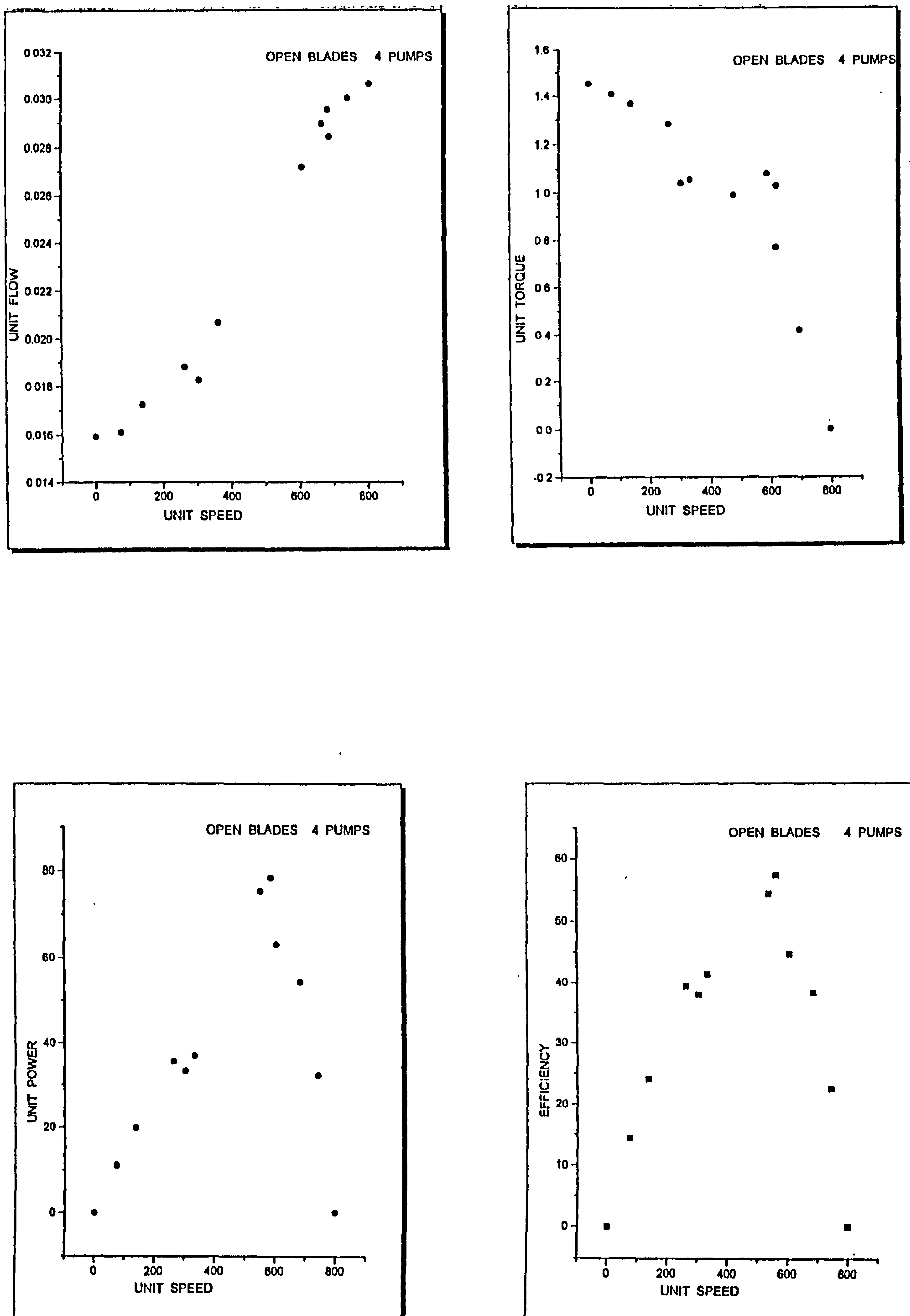

OPEN BLADES 3 PUMPS
UNIT FLOW
0.015
0.020
0 025
0.030
0
200
400
600
800
UNIT SPEED
OPEN BLADES 3 PUMPS
UNIT TORQUE
0 0
0 5
1.0
1.5
2 0
0
200
400
600
800
UNIT SPEED
OPEN BLADES 3 PUMPS
UNIT POWER
0
20
40
60
80
100
0
200
400
600
800
UNIT SPEED
OPEN BLADES 3 PUMPS
EFFICIENCY
0
10
20
30
40
0
200
400
600
800
UNIT SPEED

Figure ( 6.3.1 )



OPEN BLADES 4 PUMPS
UNIT FLOW
0.032
0.030
0.028
0.026
0.024
0.022
0.020
0.018
0.016
0.014
0
200
400
600
800
UNIT SPEED
OPEN BLADES 4 PUMPS
UNIT TORQUE
1.6
1.4
1.2
1.0
0.8
0.6
0.4
0.2
0.0
-0.2
UNIT SPEED
OPEN BLADES 4 PUMPS
UNIT POWER
80
60
40
20
0
UNIT SPEED
OPEN BLADES 4 PUMPS
EFFICIENCY
60
50
40
30
20
10
0
UNIT SPEED


Figure ( 6.4.1 )

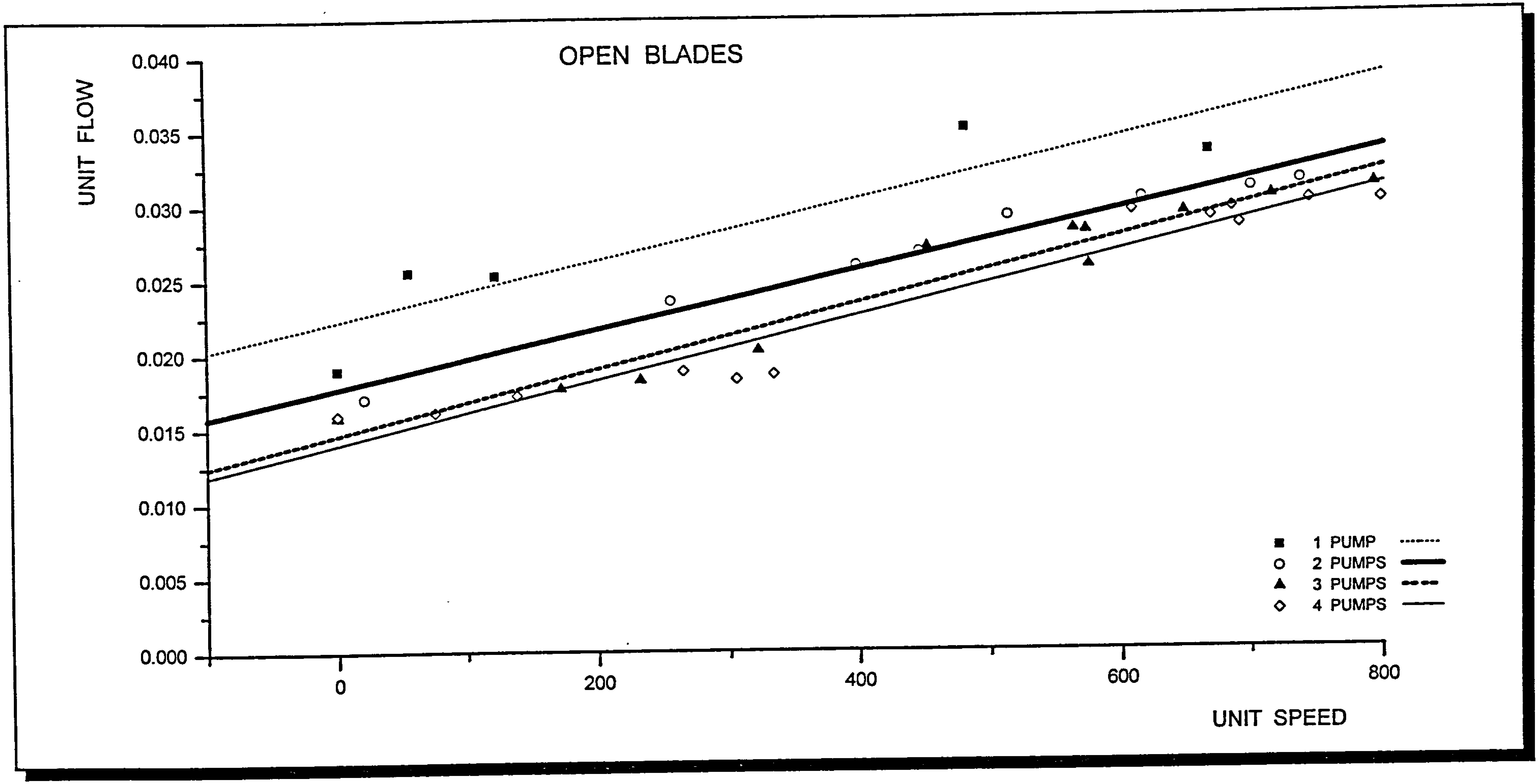

OPEN BLADES
UNIT FLOW
0.040
0.035
0.030
0.025
0.020
0.015
0.010
0.005
0.000
0
200
400
600
800
UNIT SPEED
1 PUMP
2 PUMPS
3 PUMPS
4 PUMPS

Figure ( 6.5.1 )

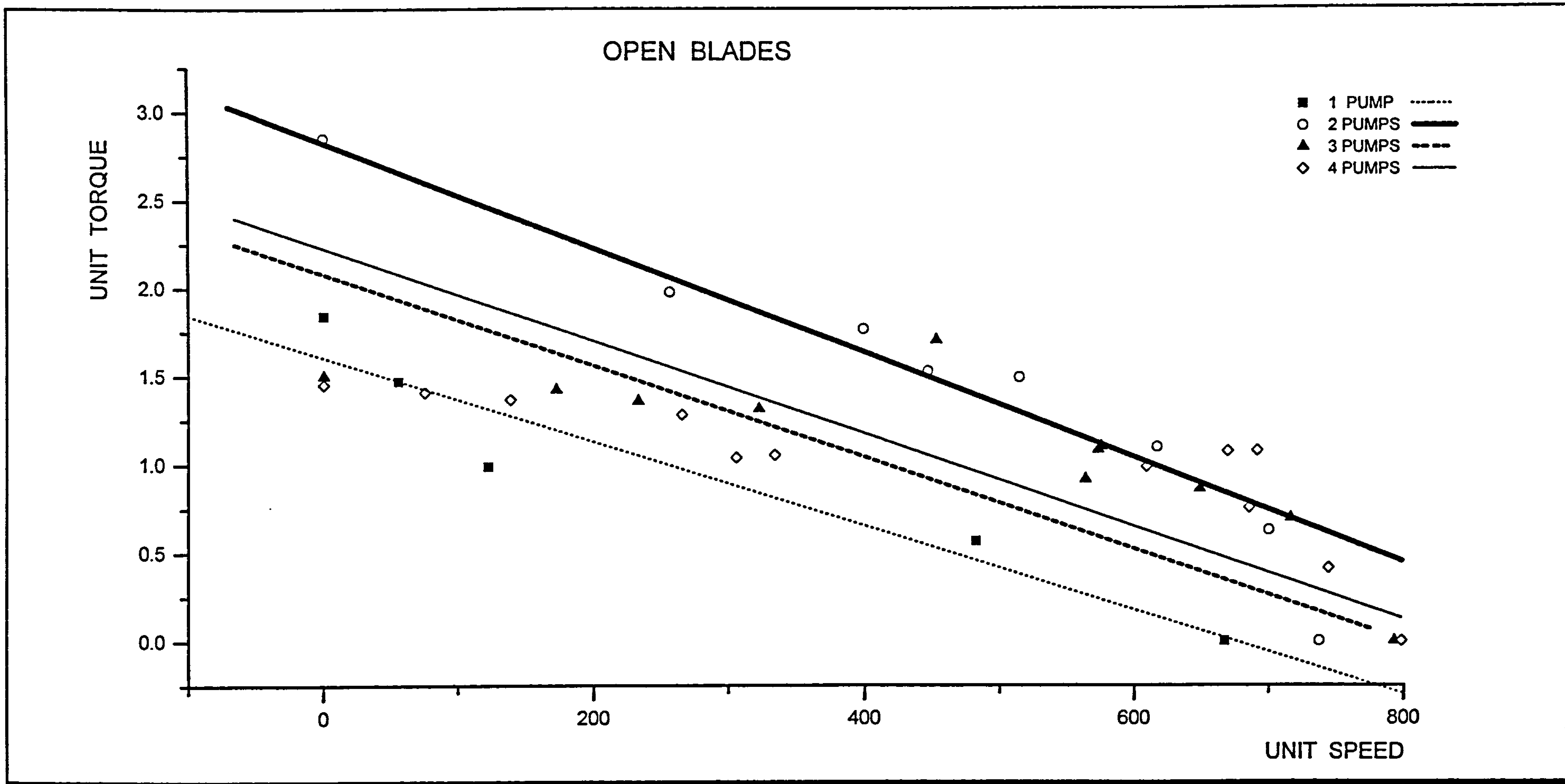

OPEN BLADES
UNIT TORQUE
3.0
2.5
2.0
1.5
1.0
0.5
0.0
0
200
400
600
800
UNIT SPEED
1 PUMP
2 PUMPS
3 PUMPS
4 PUMPS

Figure ( 6.6.1 )

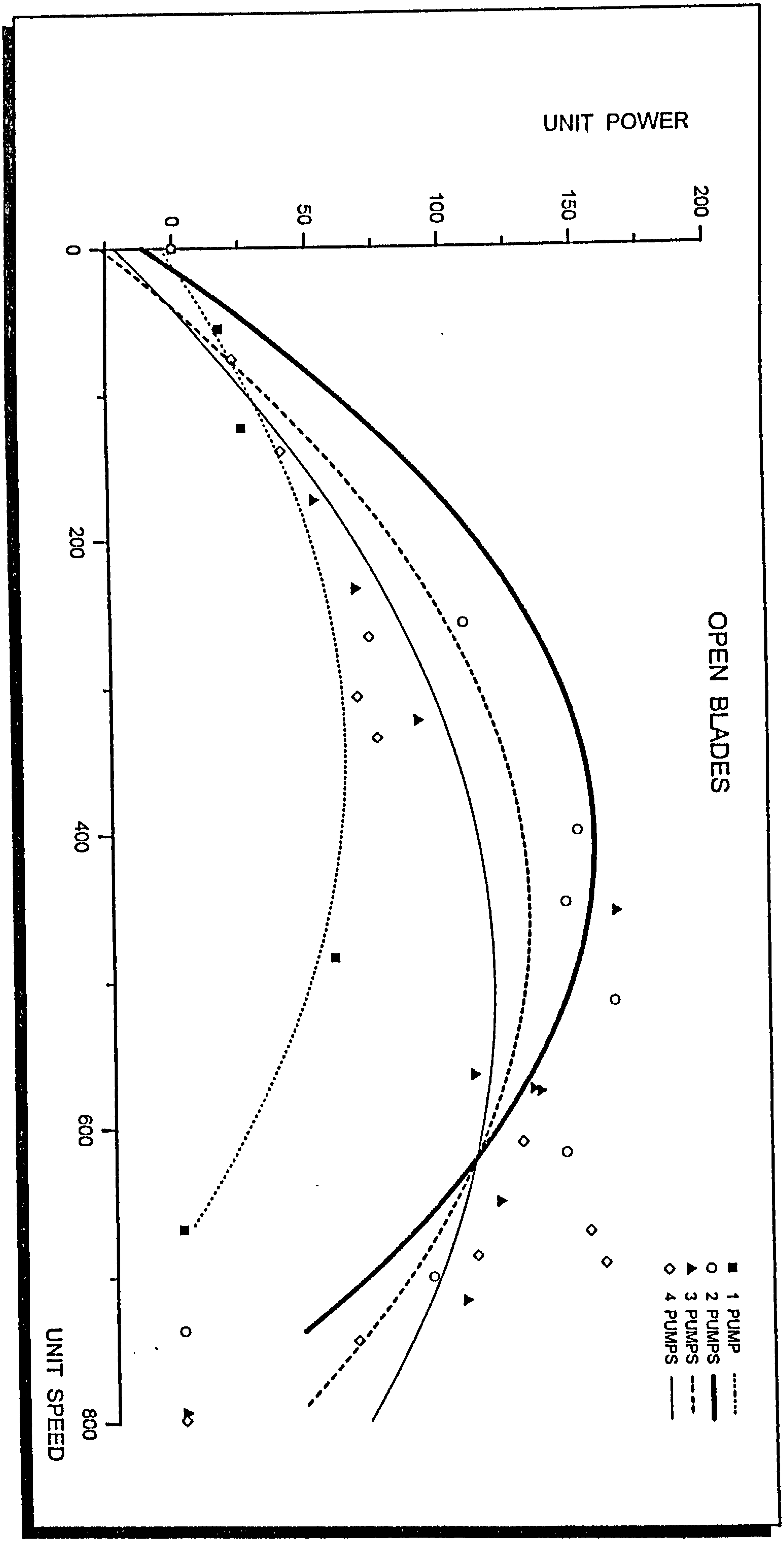
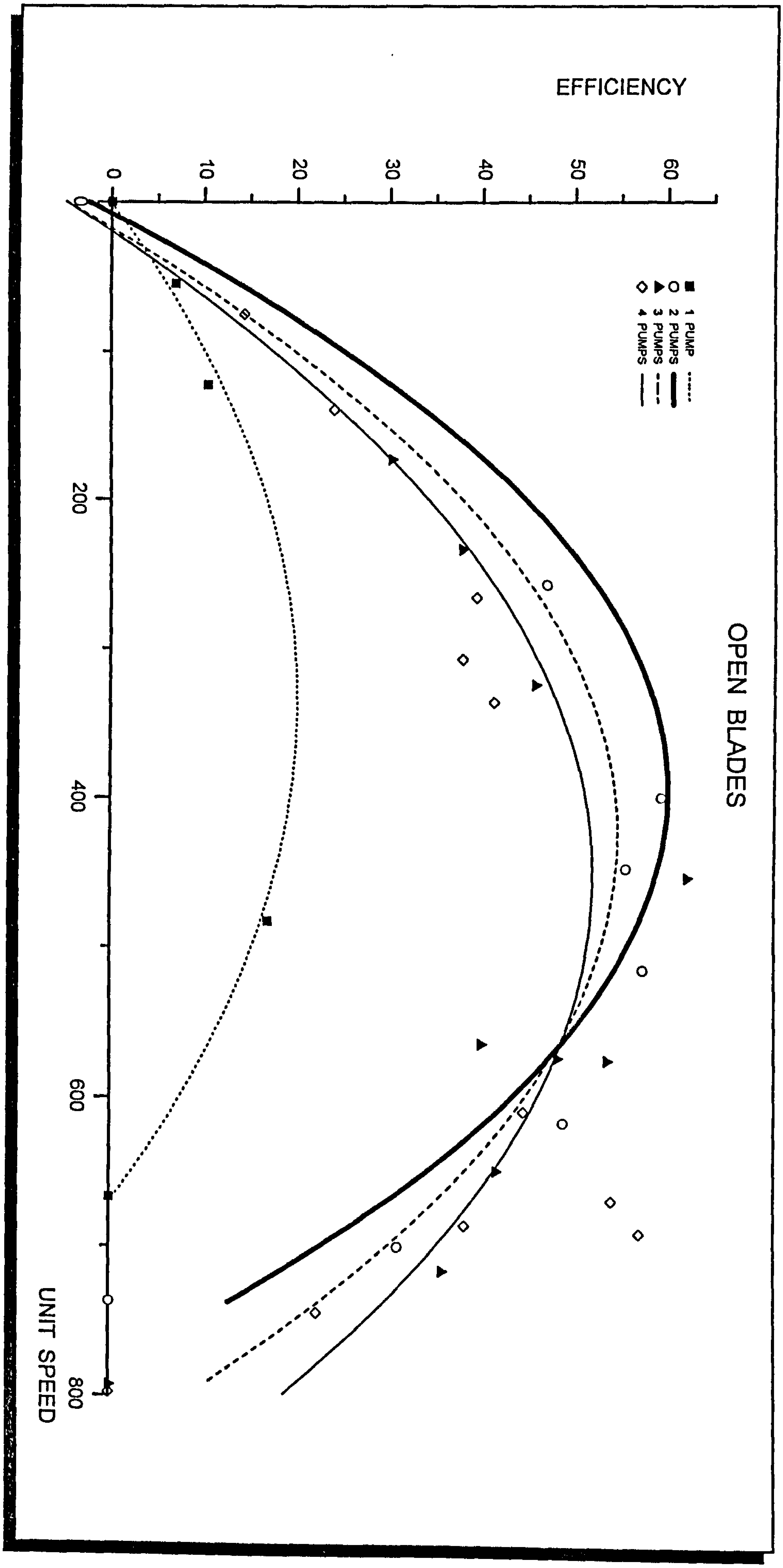

OPEN BLADES
UNIT POWER
UNIT SPEED
0
50
100
150
200
0
200
400
600
800
1 PUMP
2 PUMPS
3 PUMPS
4 PUMPS

Figure ( 6.7.1 )



OPEN BLADES
EFFICIENCY
UNIT SPEED
0
10
20
30
40
50
60
200
400
600
800
1 PUMP
2 PUMPS
3 PUMPS
4 PUMPS


Figure ( 6.8.1 )

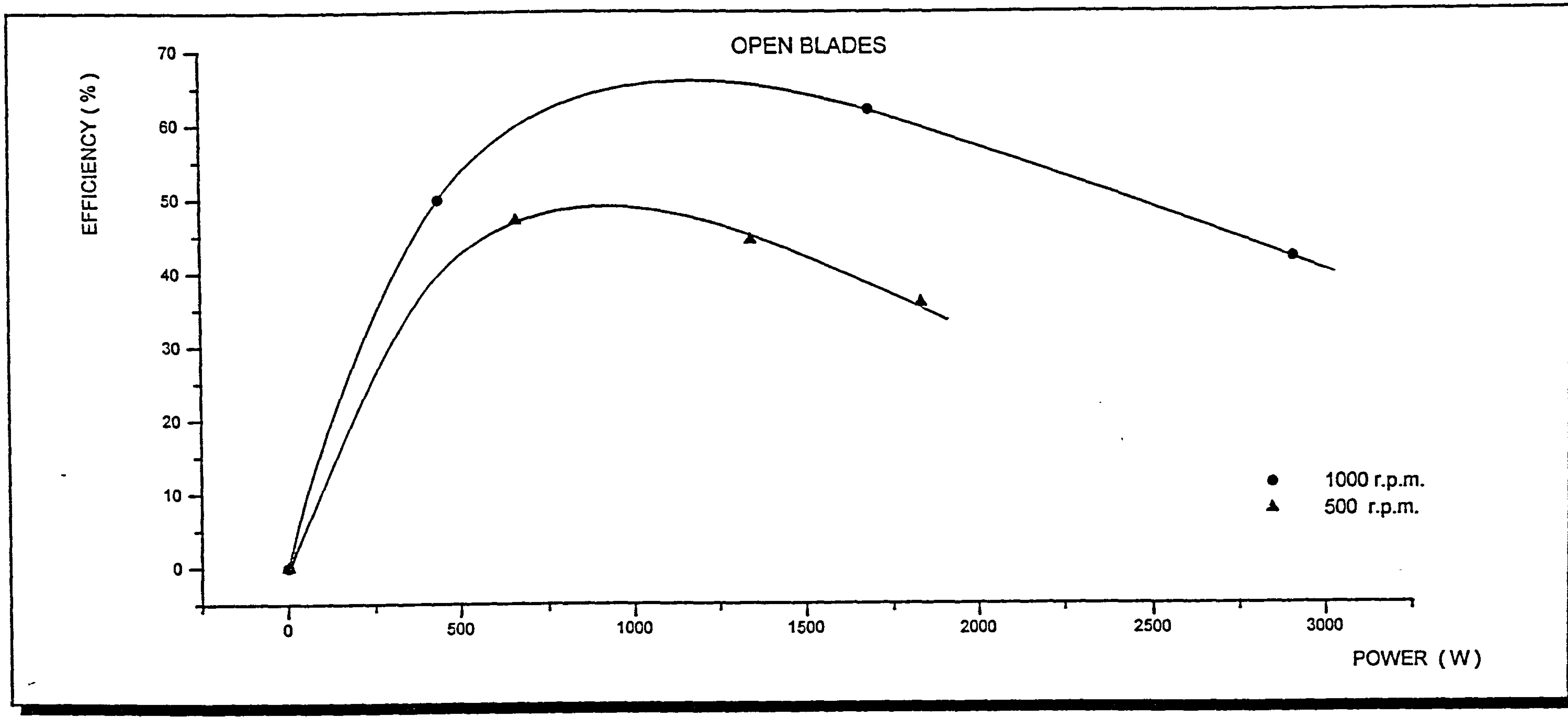

OPEN BLADES
EFFICIENCY ( % )
70
60
50
40
30
20
10
0
0
500
1000
1500
2000
2500
3000
POWER ( W )
1000 r.p.m.
500 r.p.m.

Figure ( 6.9.1 )

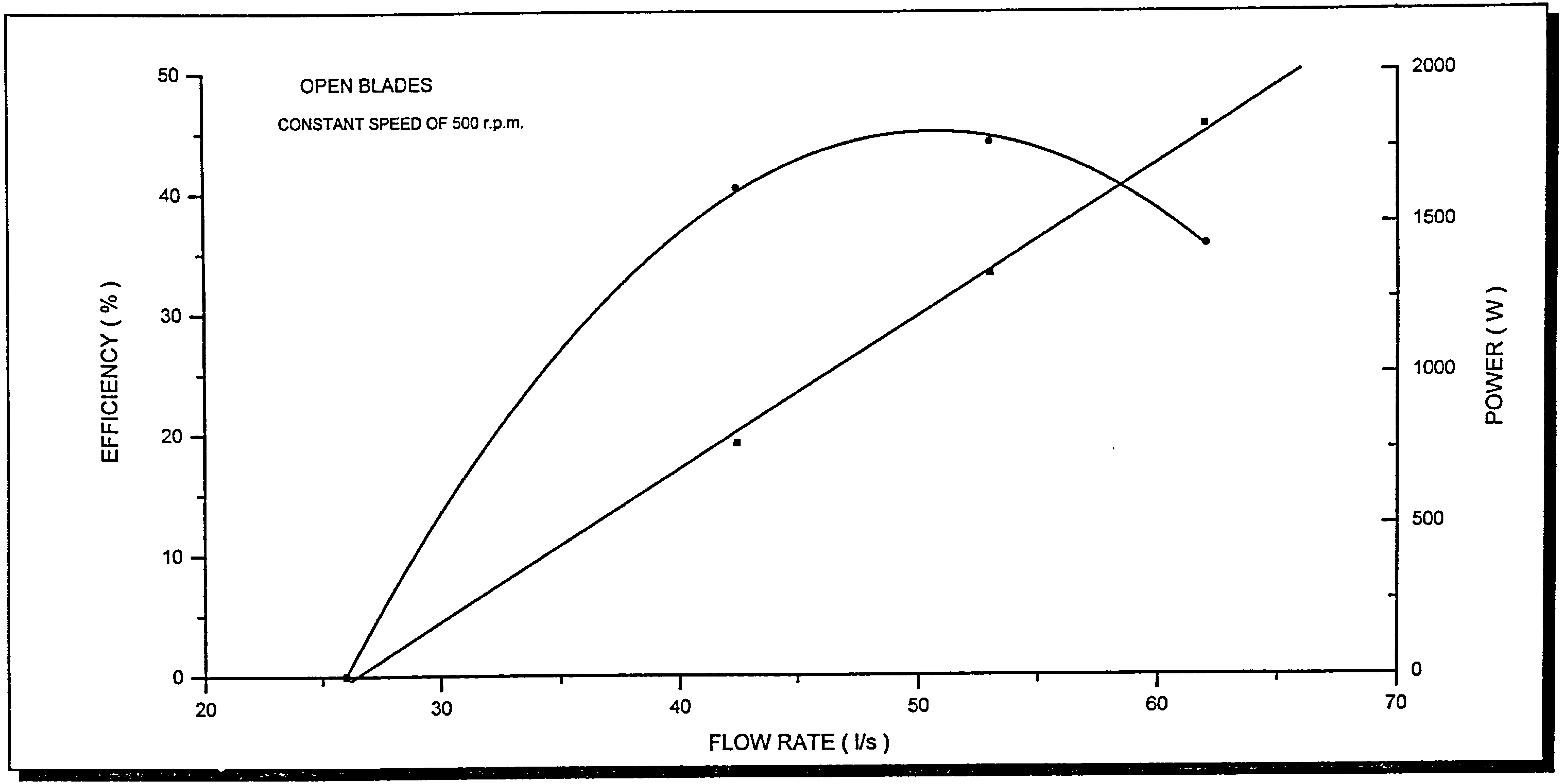

OPEN BLADES
CONSTANT SPEED OF 500 r.p.m.
EFFICIENCY ( % )
0
10
20
30
40
50
POWER ( W )
0
500
1000
1500
2000
FLOW RATE ( l/s )
20
30
40
50
60
70

Figure ( 6.10.1 )

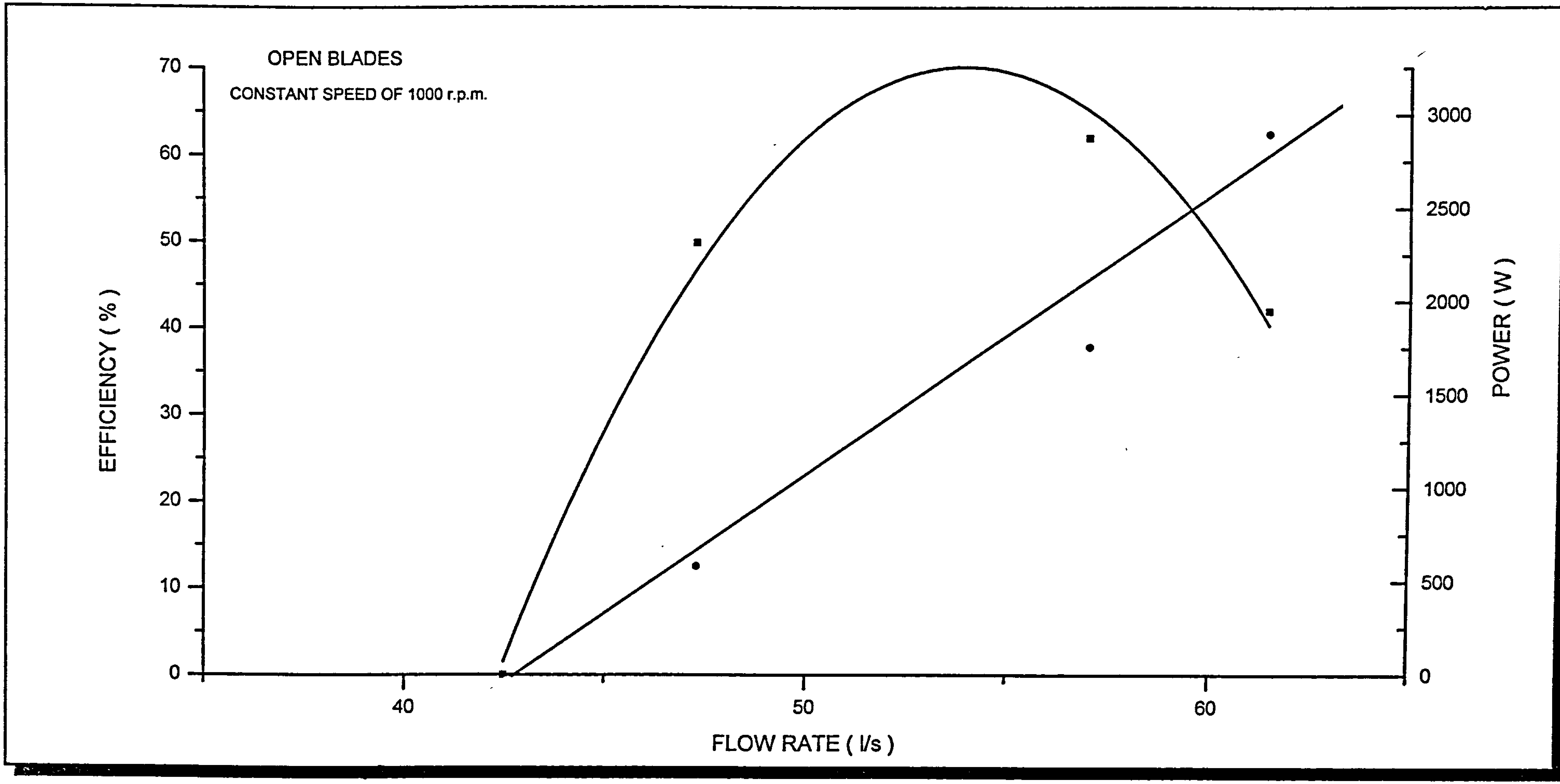
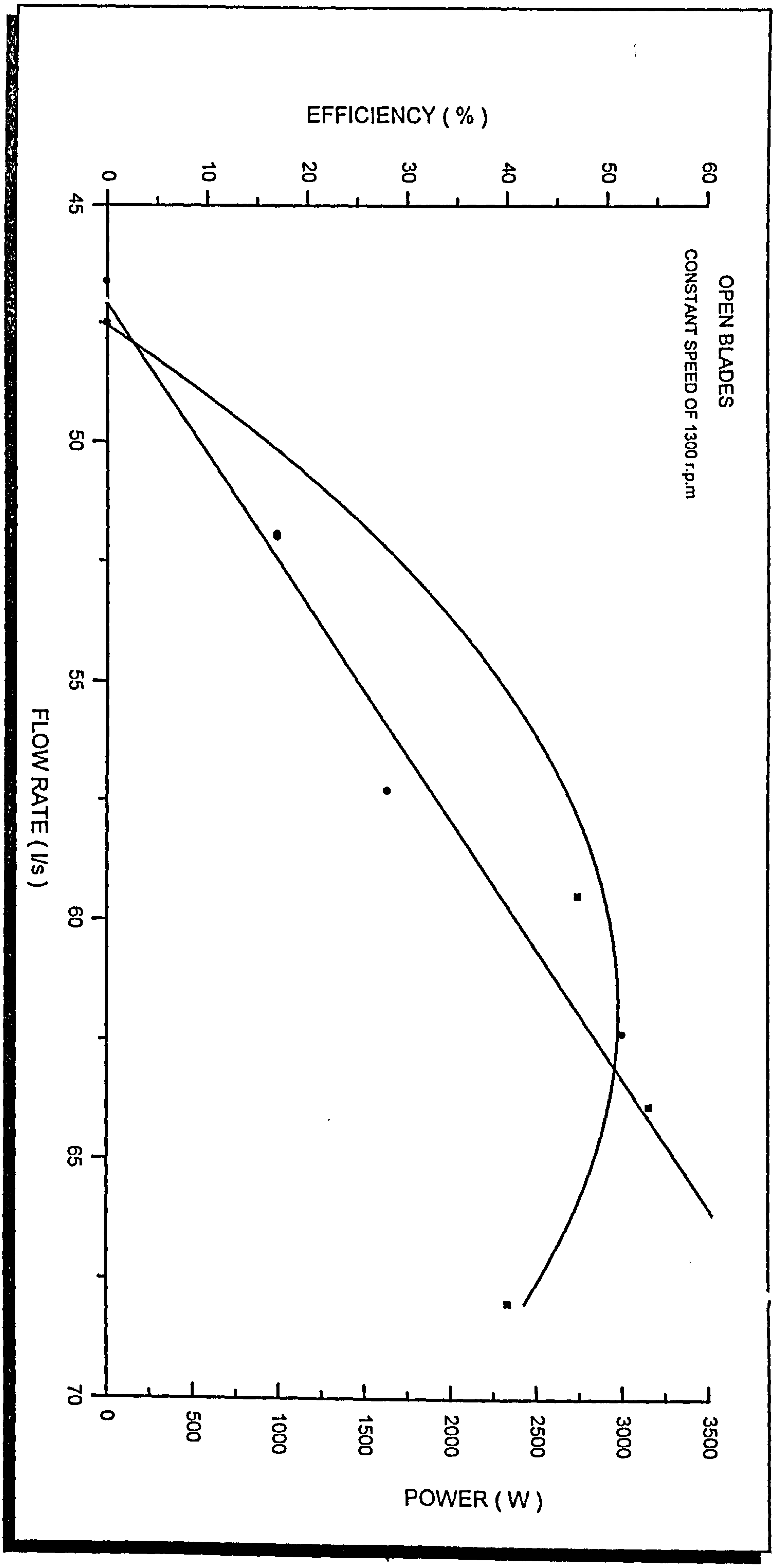

OPEN BLADES
CONSTANT SPEED OF 1000 r.p.m.
EFFICIENCY ( % )
0
10
20
30
40
50
60
70
FLOW RATE ( l/s )
40
50
60
POWER ( W )
0
500
1000
1500
2000
2500
3000

Figure ( 6.11.1 )



EFFICIENCY ( % )
OPEN BLADES
CONSTANT SPEED OF 1300 r.p.m
0
10
20
30
40
50
60
45
50
55
60
65
70
FLOW RATE ( l/s )
0
500
1000
1500
2000
2500
3000
3500
POWER ( W )


Figure ( 6.12.1 )

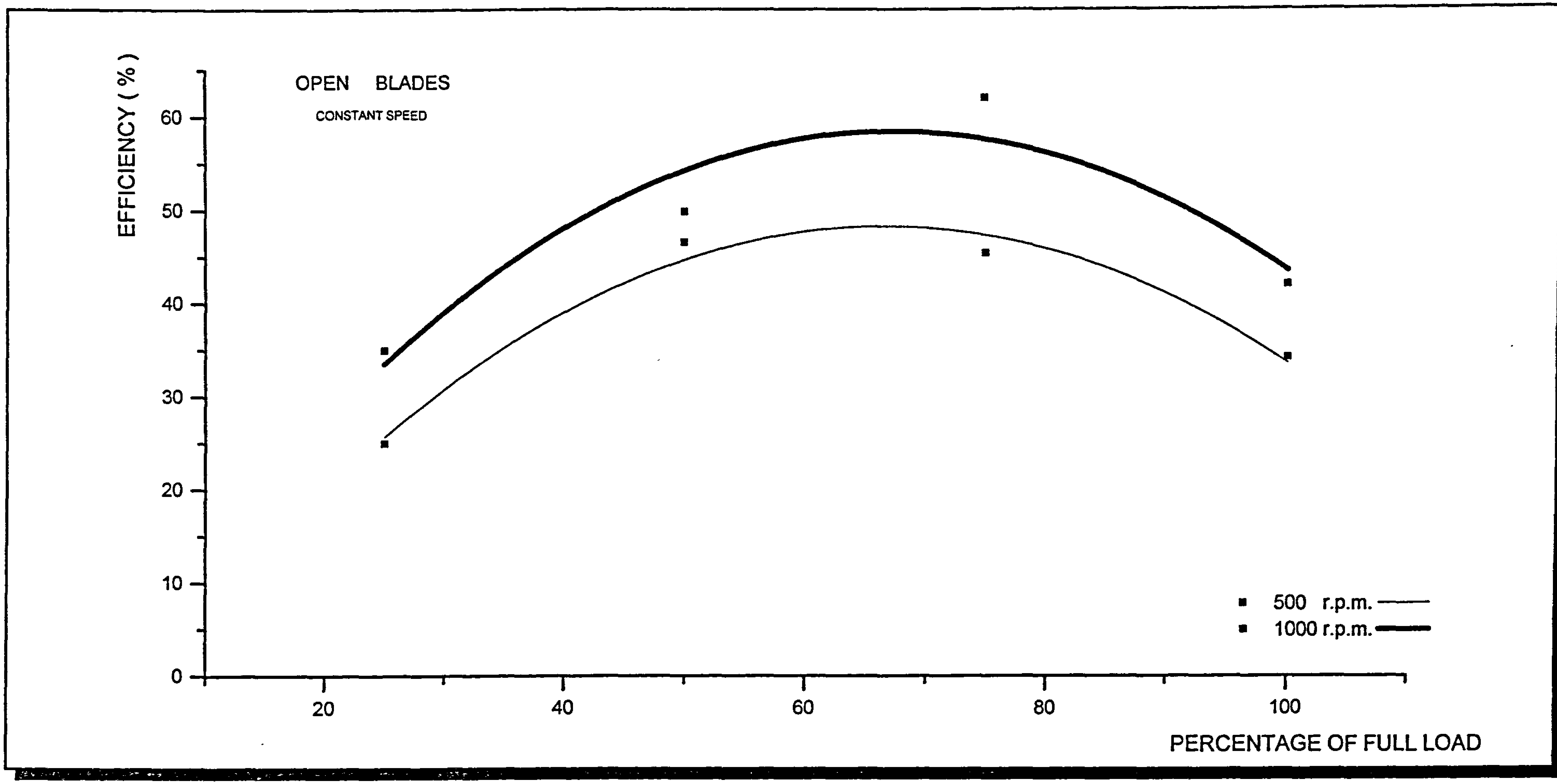

EFFICIENCY ( % )
OPEN BLADES
CONSTANT SPEED
60
50
40
30
20
10
0
20
40
60
80
100
500 r.p.m.
1000 r.p.m.
PERCENTAGE OF FULL LOAD

Figure ( 6.13.1 )

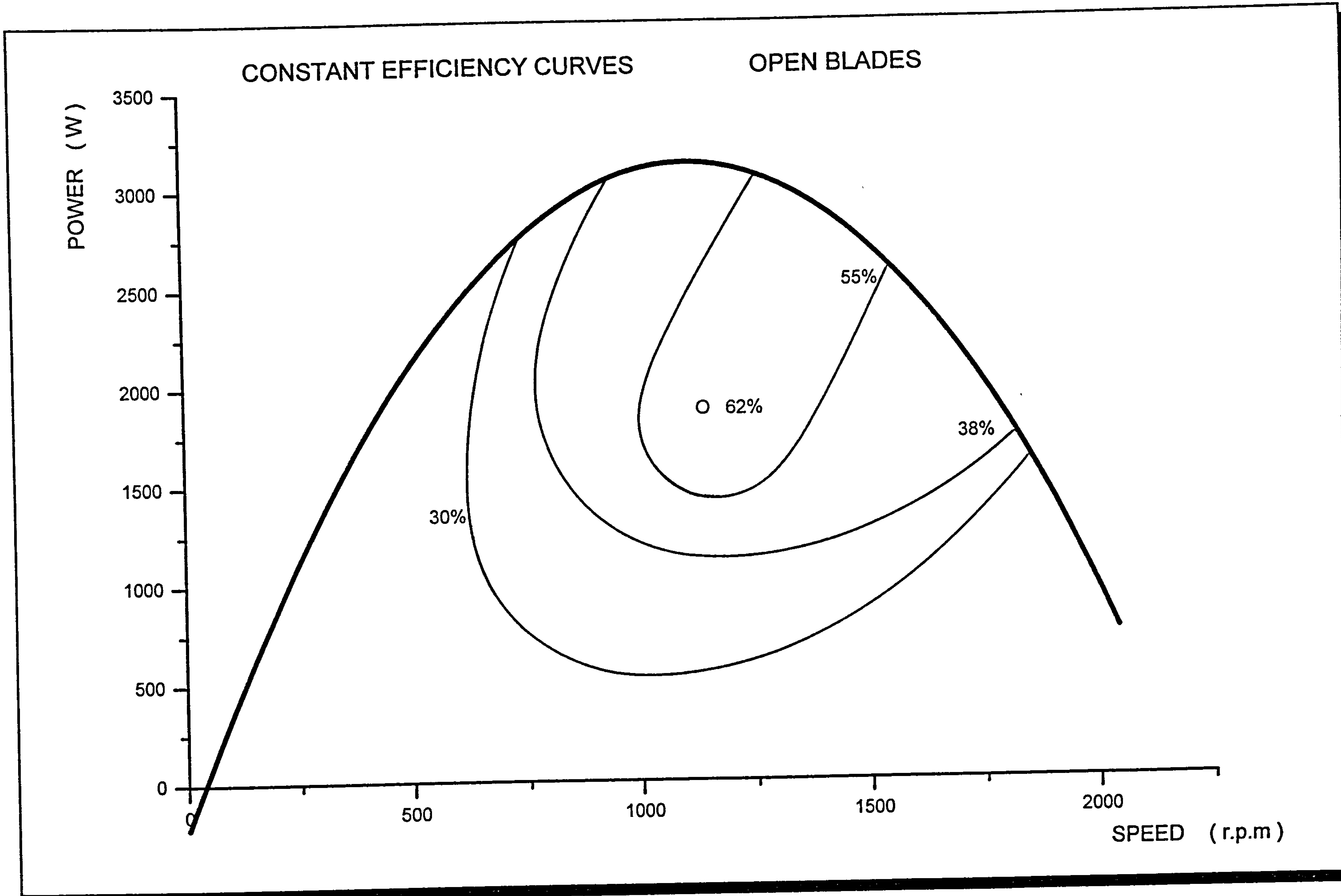

CONSTANT EFFICIENCY CURVES
OPEN BLADES
POWER (W)
3500
3000
2500
2000
1500
1000
500
0
0
500
1000
1500
2000
SPEED (r.p.m)
30%
38%
55%
62%

Figure ( 6.14.1 )

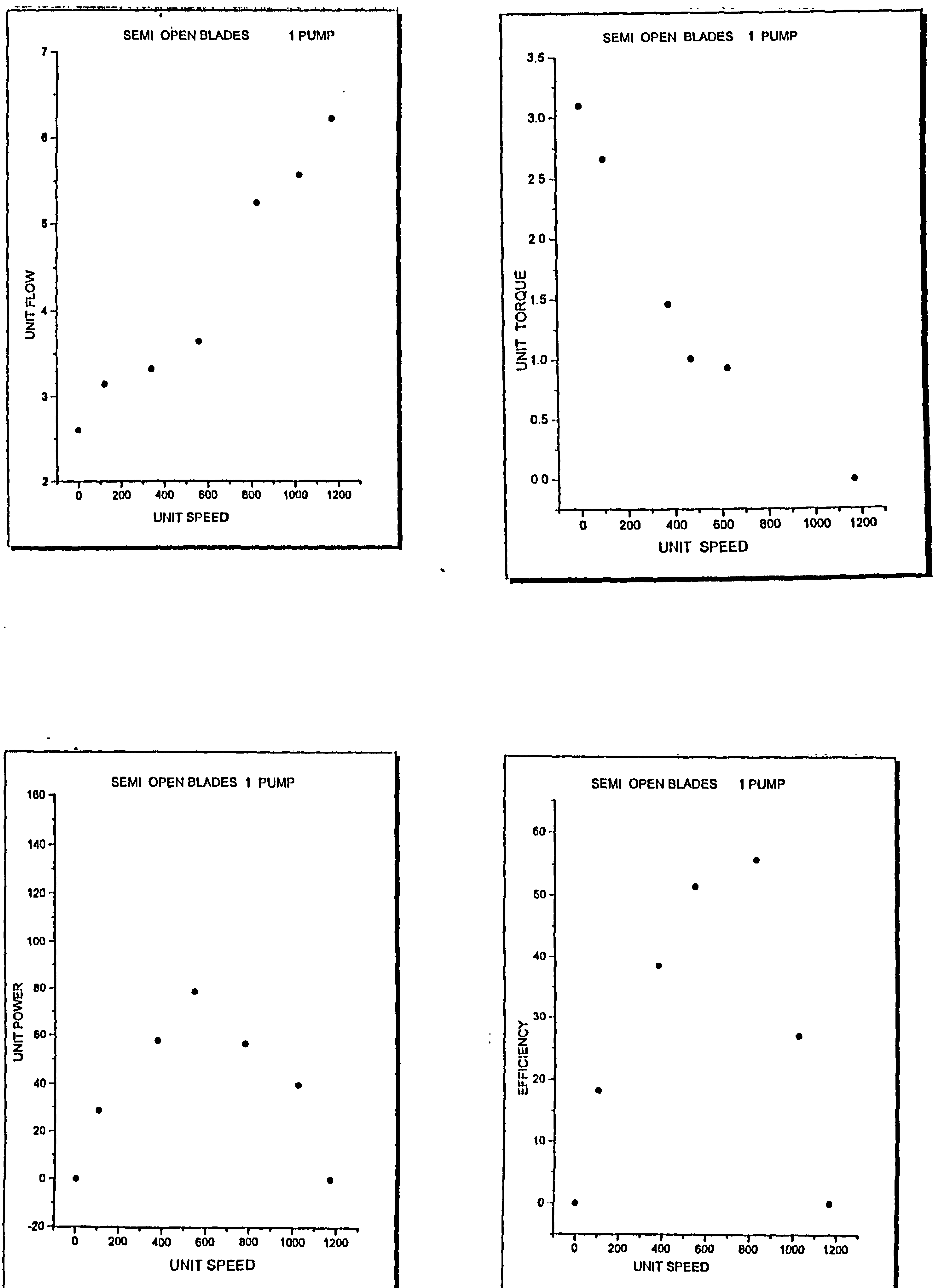

SEMI OPEN BLADES 1 PUMP
UNIT FLOW
UNIT SPEED
SEMI OPEN BLADES 1 PUMP
UNIT TORQUE
UNIT SPEED
SEMI OPEN BLADES 1 PUMP
UNIT POWER
UNIT SPEED
SEMI OPEN BLADES 1 PUMP
EFFICIENCY
UNIT SPEED

Figure ( 6.1.2 )

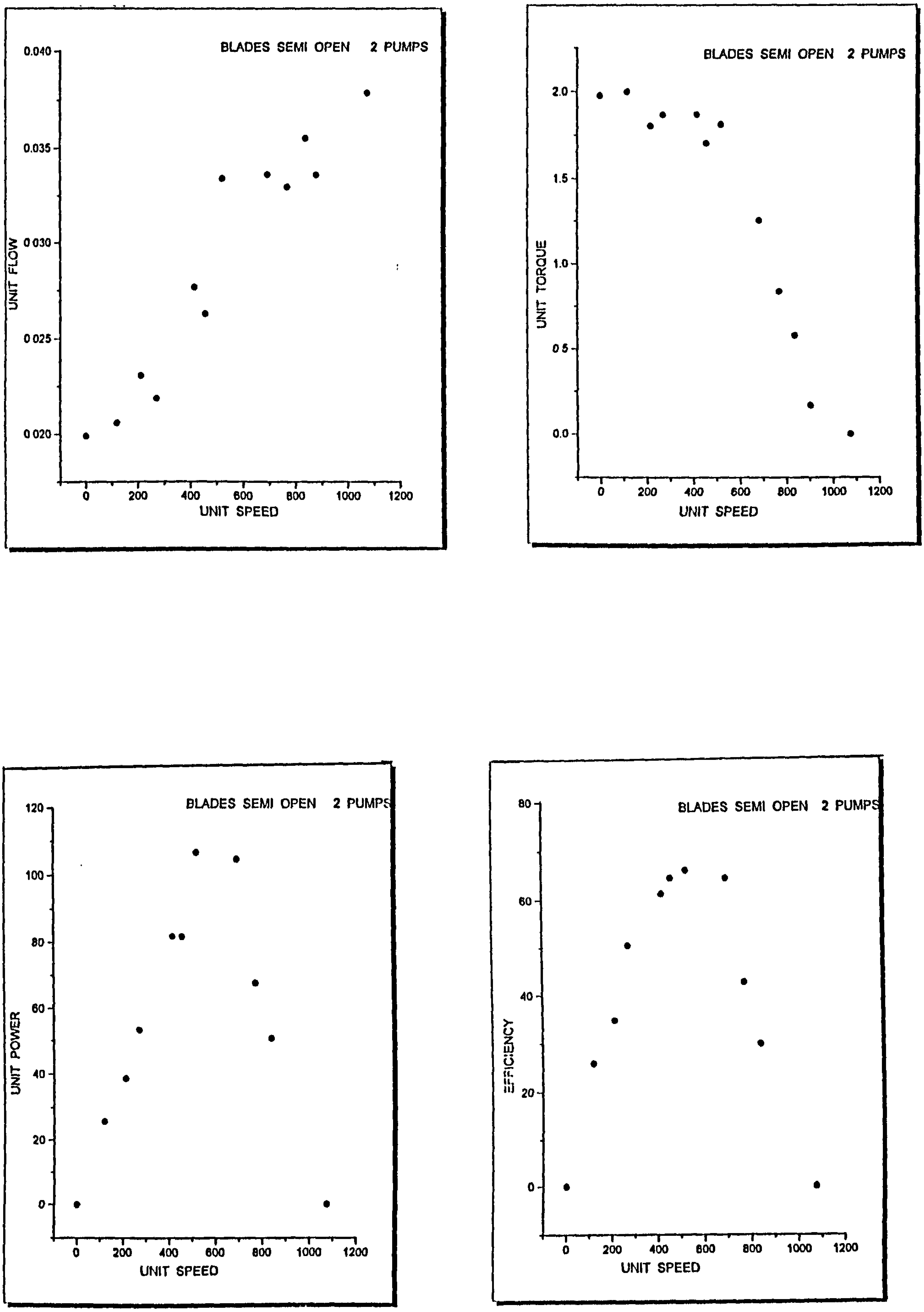
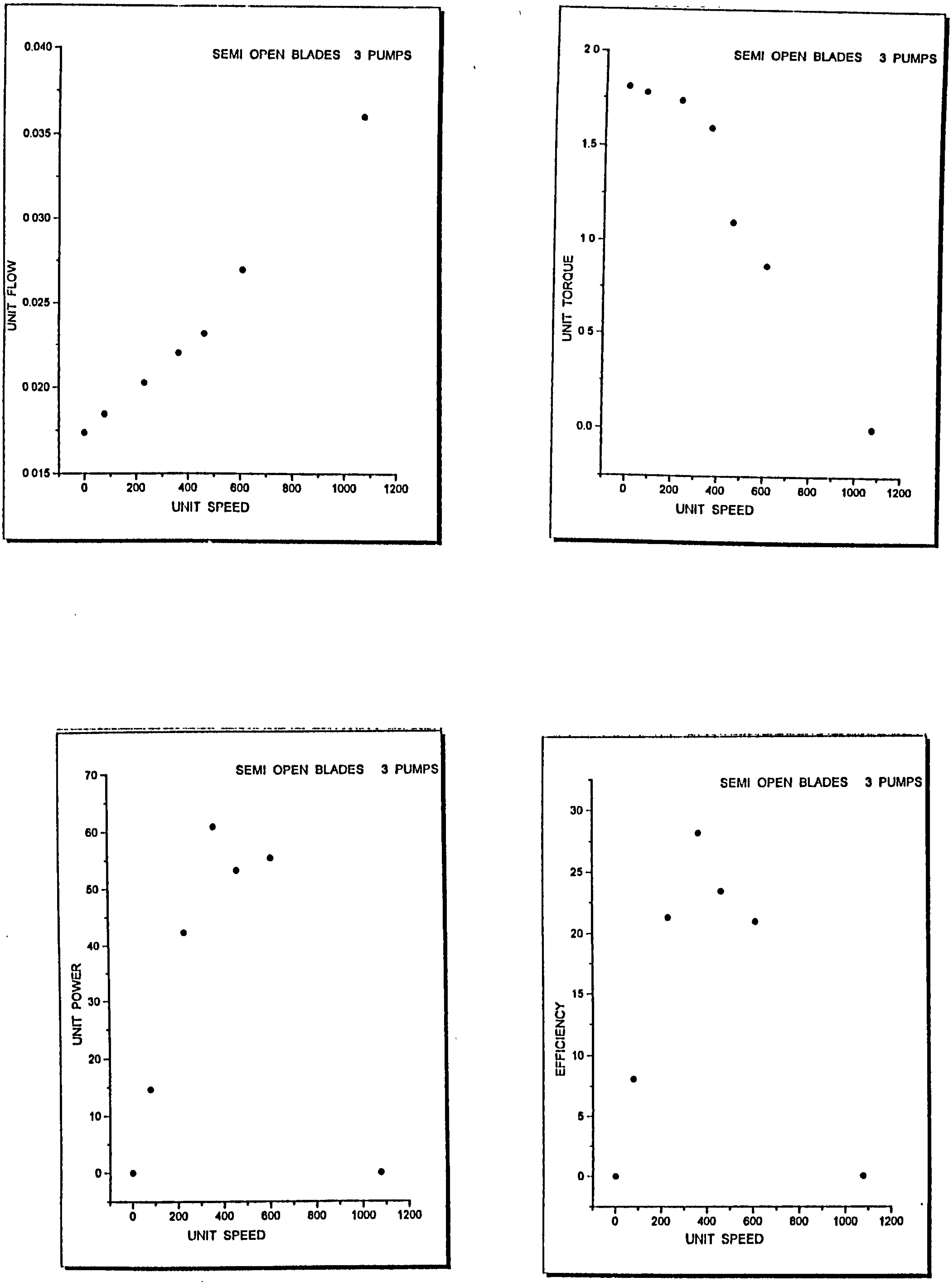
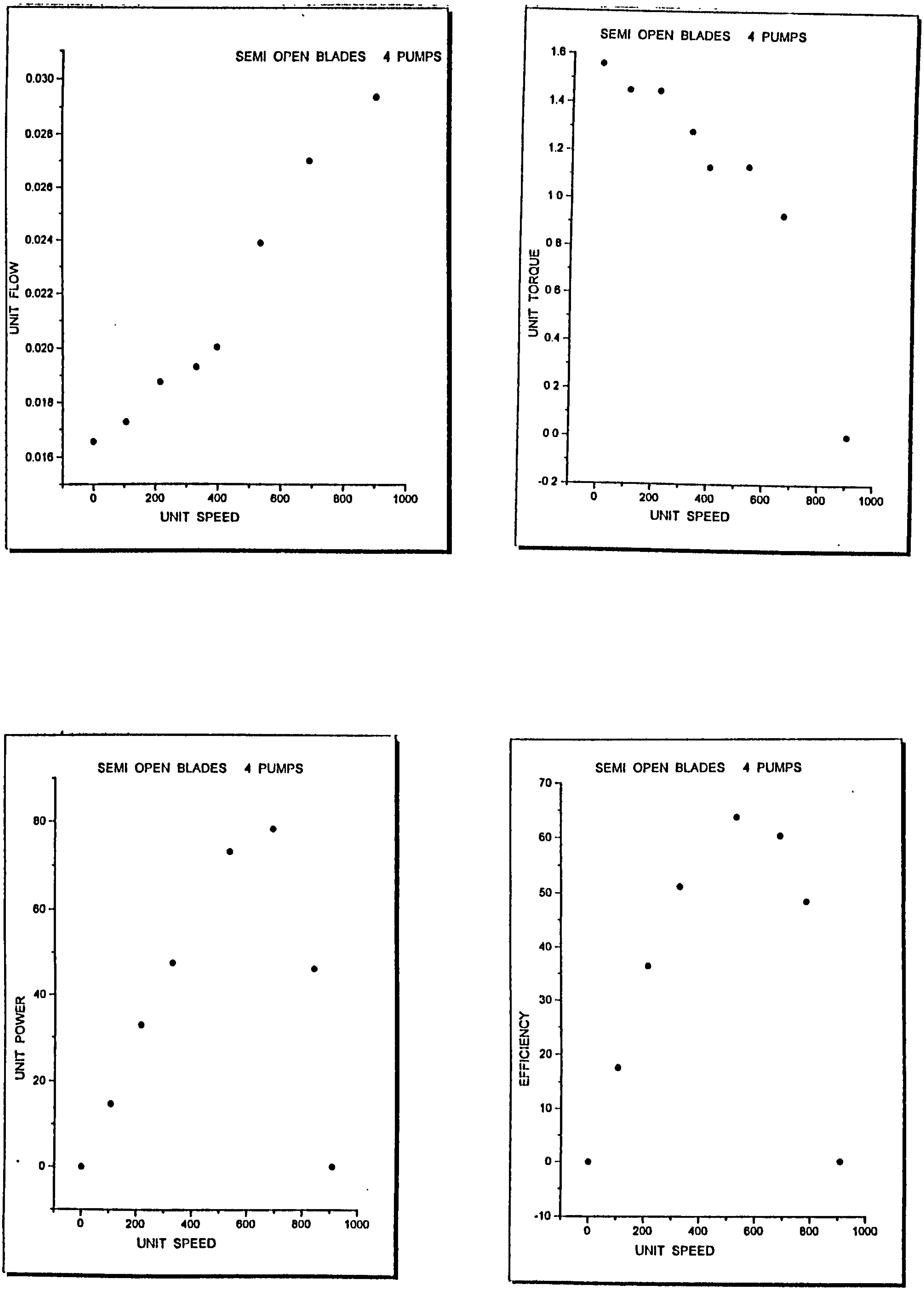

BLADES SEMI OPEN 2 PUMPS
UNIT FLOW
UNIT SPEED
BLADES SEMI OPEN 2 PUMPS
UNIT TORQUE
UNIT SPEED
BLADES SEMI OPEN 2 PUMPS
UNIT POWER
UNIT SPEED
BLADES SEMI OPEN 2 PUMPS
EFFICIENCY
UNIT SPEED

Figure ( 6.2.2 )



SEMI OPEN BLADES 3 PUMPS
UNIT FLOW
UNIT SPEED
0.040
0.035
0 030
0.025
0 020
0 015
0
200
400
600
800
1000
1200
SEMI OPEN BLADES 3 PUMPS
UNIT TORQUE
UNIT SPEED
2 0
1.5
1 0
0 5
0.0
SEMI OPEN BLADES 3 PUMPS
UNIT POWER
UNIT SPEED
70
60
50
40
30
20
10
0
SEMI OPEN BLADES 3 PUMPS
EFFICIENCY
UNIT SPEED
30
25
20
15
10
5
0


Figure ( 6.3.2 )



SEMI OPEN BLADES 4 PUMPS
UNIT FLOW
UNIT SPEED
SEMI OPEN BLADES 4 PUMPS
UNIT TORQUE
UNIT SPEED
SEMI OPEN BLADES 4 PUMPS
UNIT POWER
UNIT SPEED
SEMI OPEN BLADES 4 PUMPS
EFFICIENCY
UNIT SPEED


Figure ( 6.4.2 )

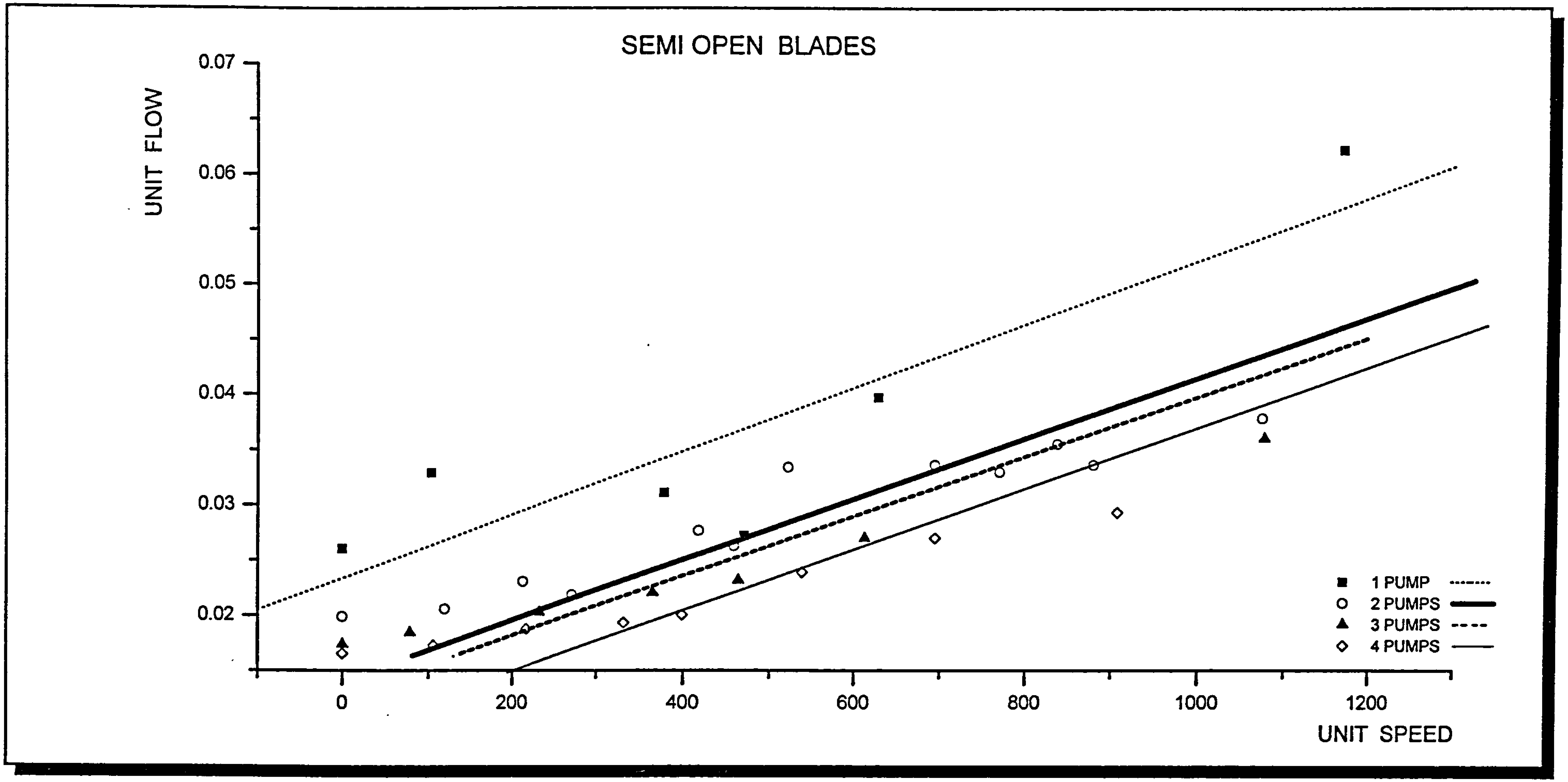

SEMI OPEN BLADES
UNIT FLOW
0.07
0.06
0.05
0.04
0.03
0.02
0
200
400
600
800
1000
1200
UNIT SPEED
1 PUMP
2 PUMPS
3 PUMPS
4 PUMPS

Figure ( 6.5.2 )

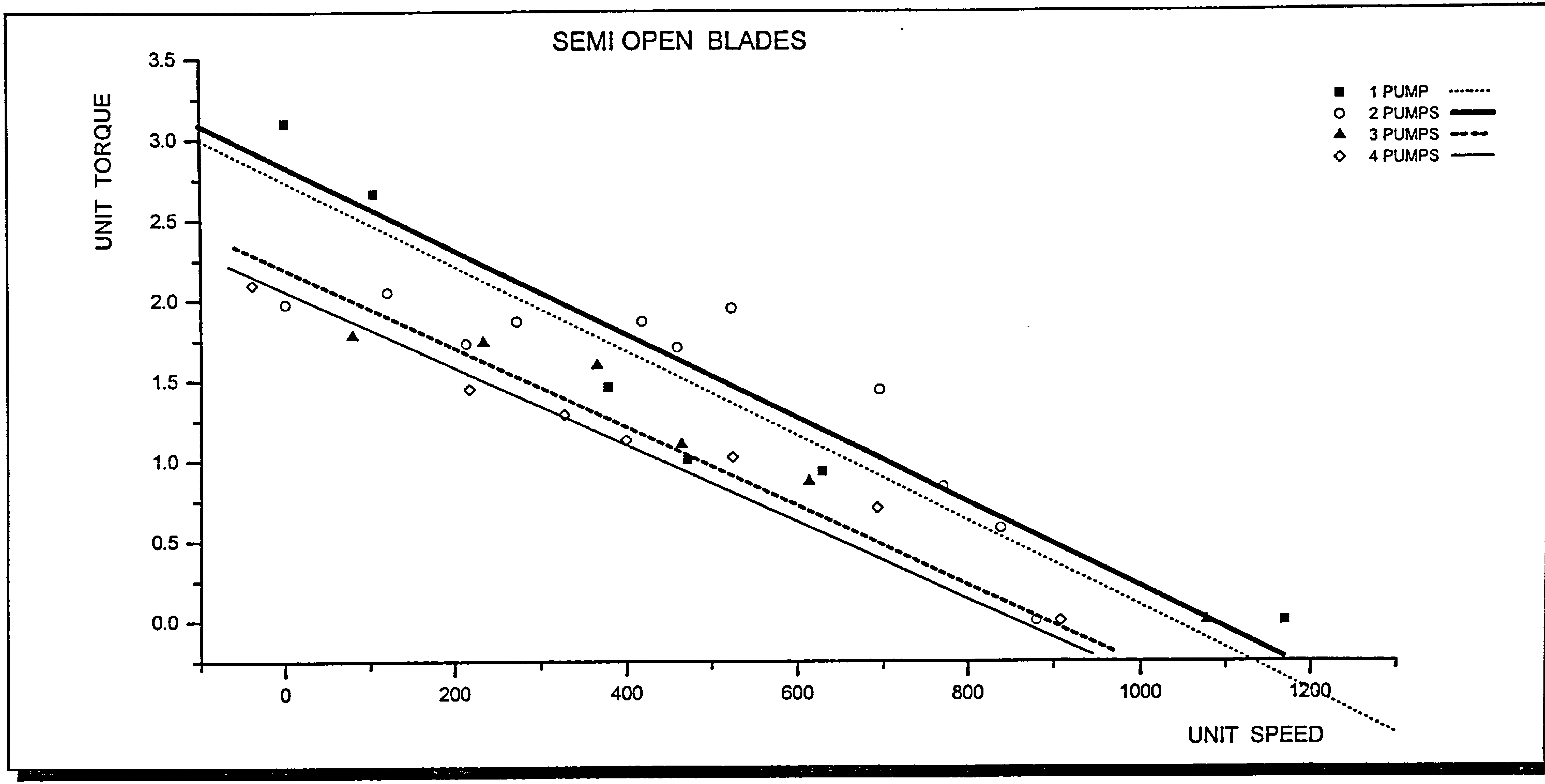

SEMI OPEN BLADES
UNIT TORQUE
3.5
3.0
2.5
2.0
1.5
1.0
0.5
0.0
0
200
400
600
800
1000
1200
UNIT SPEED
1 PUMP
2 PUMPS
3 PUMPS
4 PUMPS

Figure ( 6.6.2 )

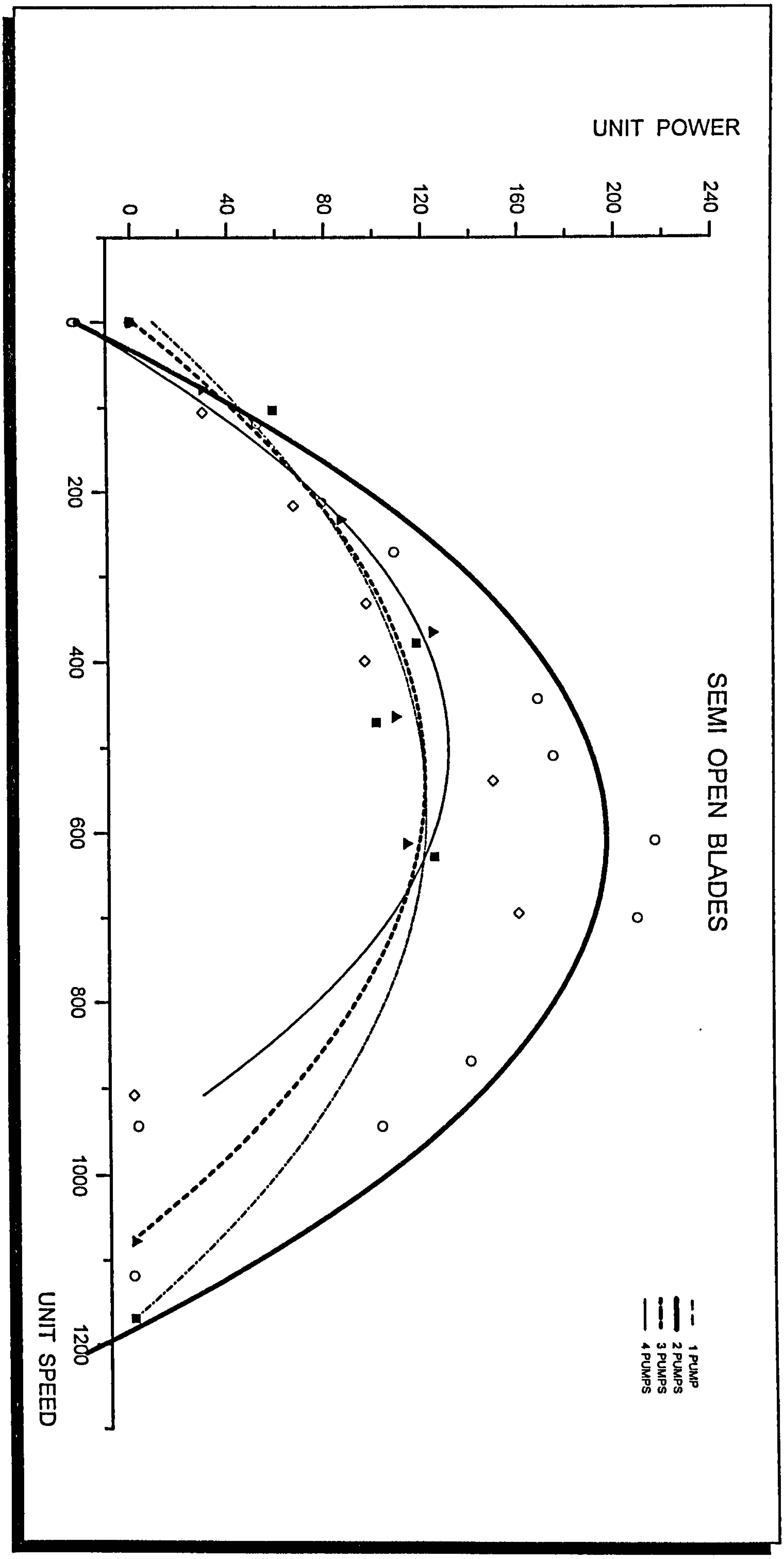

UNIT POWER
240
200
160
120
80
40
0
SEMI OPEN BLADES
1 PUMP
2 PUMPS
3 PUMPS
4 PUMPS
0
200
400
600
800
1000
1200
UNIT SPEED

Figure ( 6.7.2 )

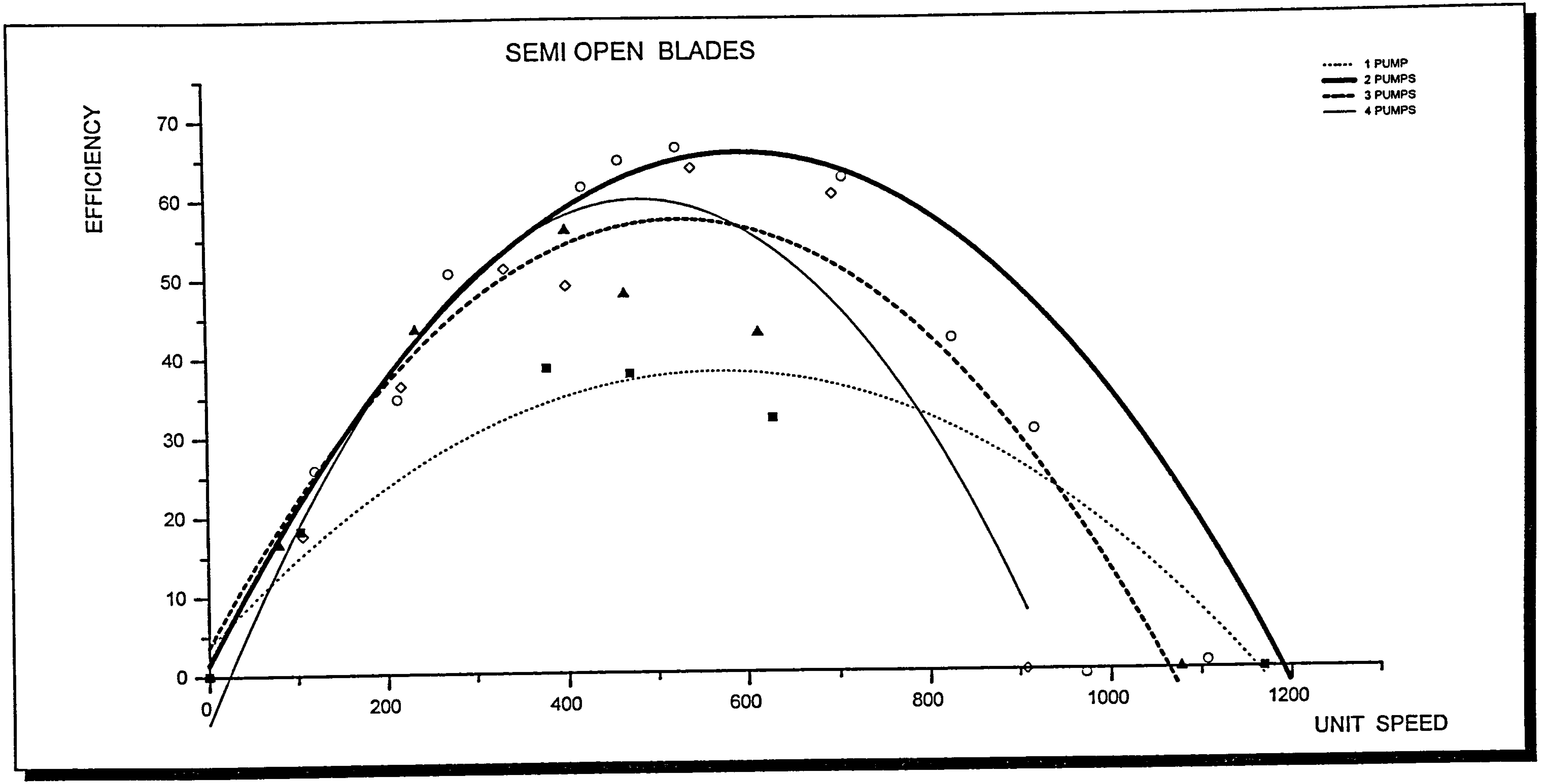

SEMI OPEN BLADES
EFFICIENCY
0
10
20
30
40
50
60
70
0
200
400
600
800
1000
1200
UNIT SPEED
1 PUMP
2 PUMPS
3 PUMPS
4 PUMPS

Figure ( 6.8.2 )

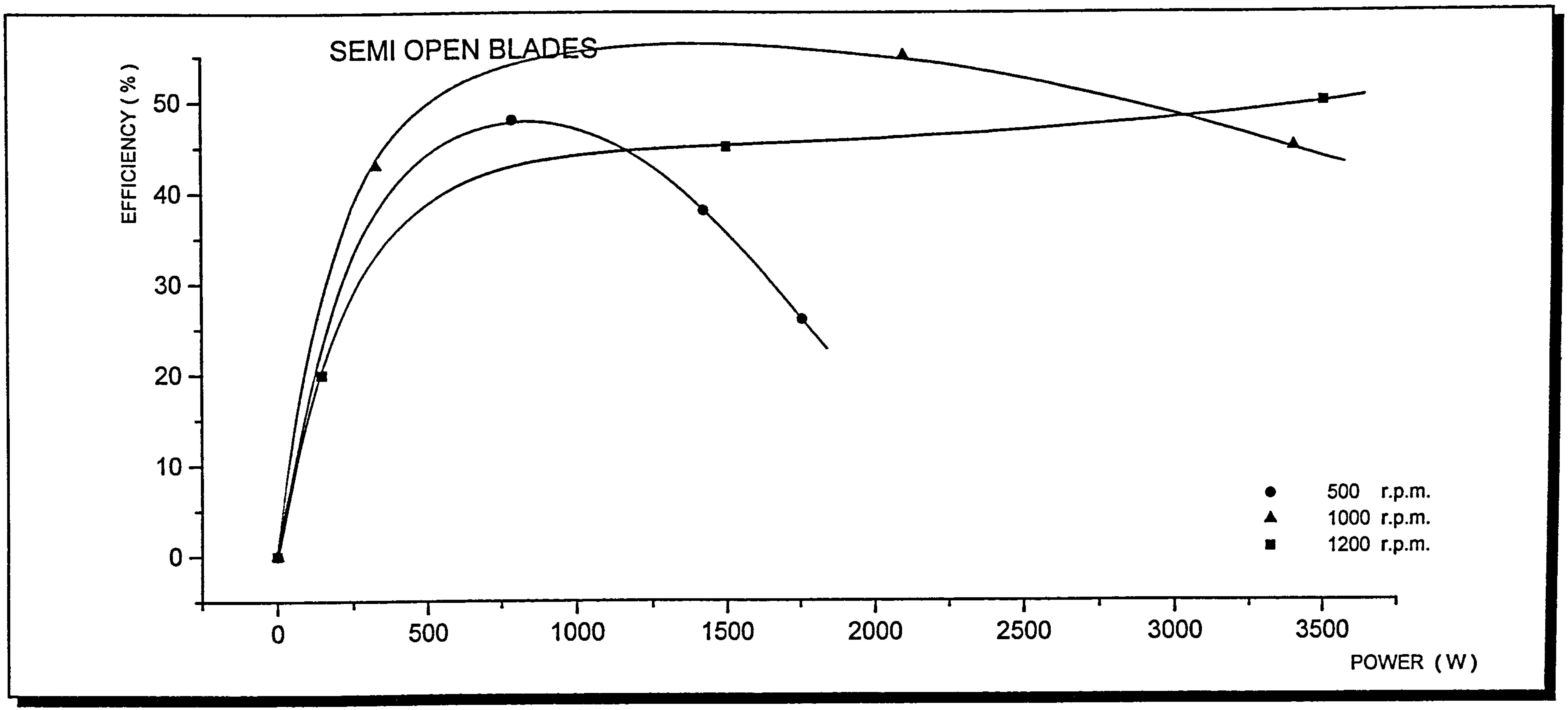
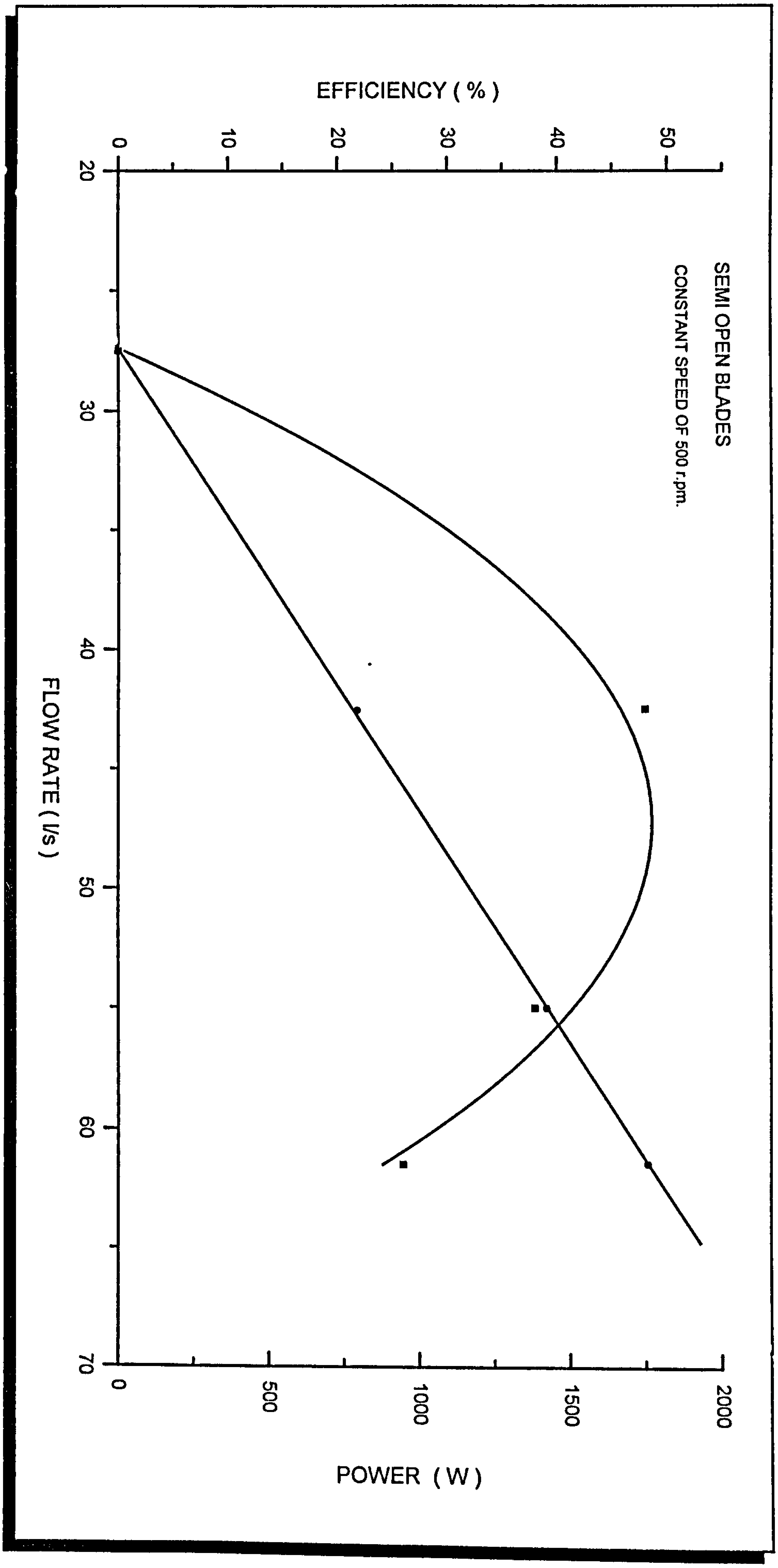

SEMI OPEN BLADES
EFFICIENCY ( % )
0
10
20
30
40
50
0
500
1000
1500
2000
2500
3000
3500
POWER ( W )
500 r.p.m.
1000 r.p.m.
1200 r.p.m.

Figure ( 6.9.2 )



SEMI OPEN BLADES
CONSTANT SPEED OF 500 r.pm.
EFFICIENCY ( % )
0
10
20
30
40
50
FLOW RATE ( l/s )
20
30
40
50
60
70
POWER ( W )
0
500
1000
1500
2000


Figure ( 6.10.2 )

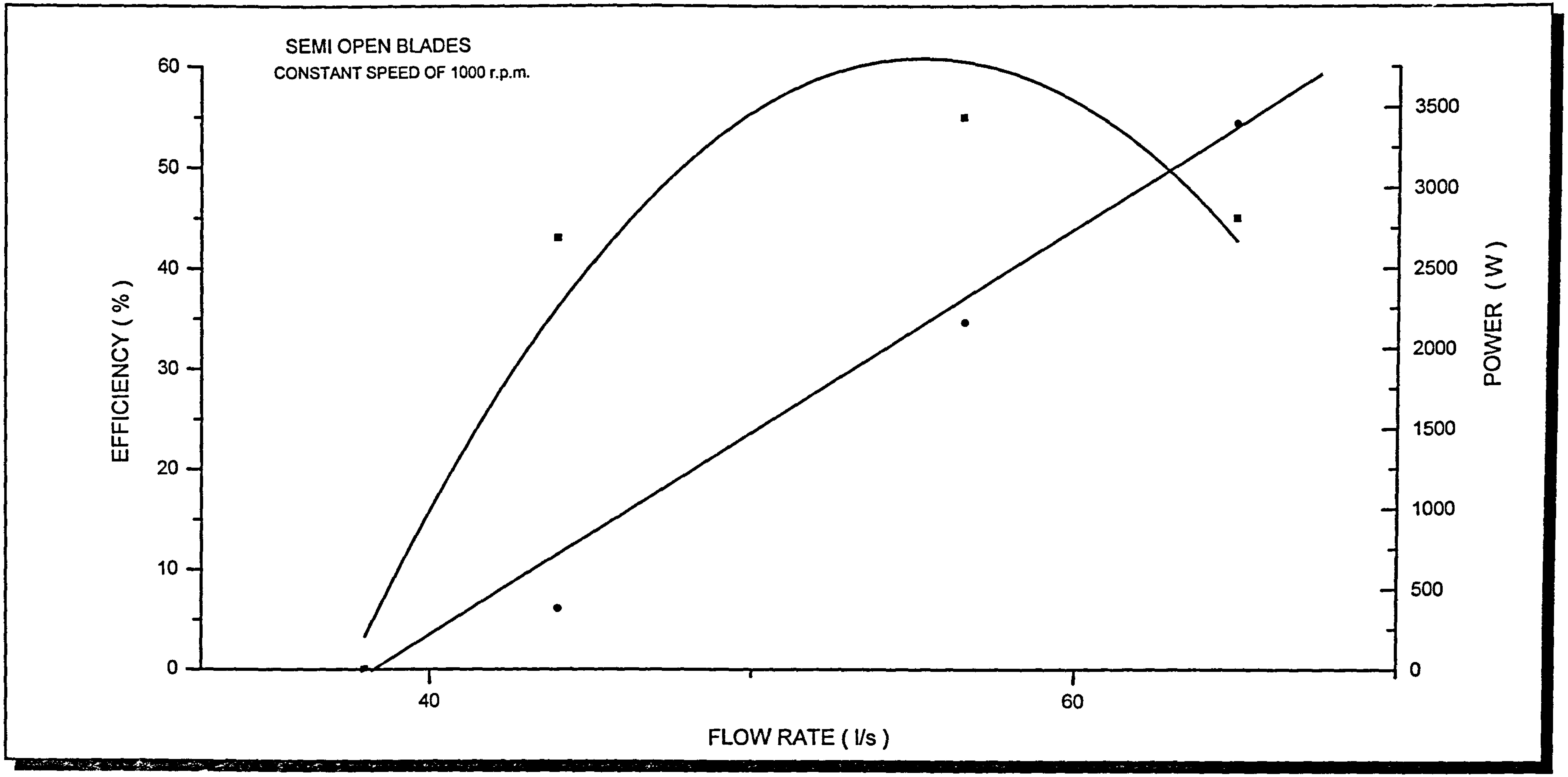

SEMI OPEN BLADES
CONSTANT SPEED OF 1000 r.p.m.
EFFICIENCY ( % )
0
10
20
30
40
50
60
40
60
FLOW RATE ( l/s )
POWER ( W )
0
500
1000
1500
2000
2500
3000
3500

Figure ( 6.11.2 )

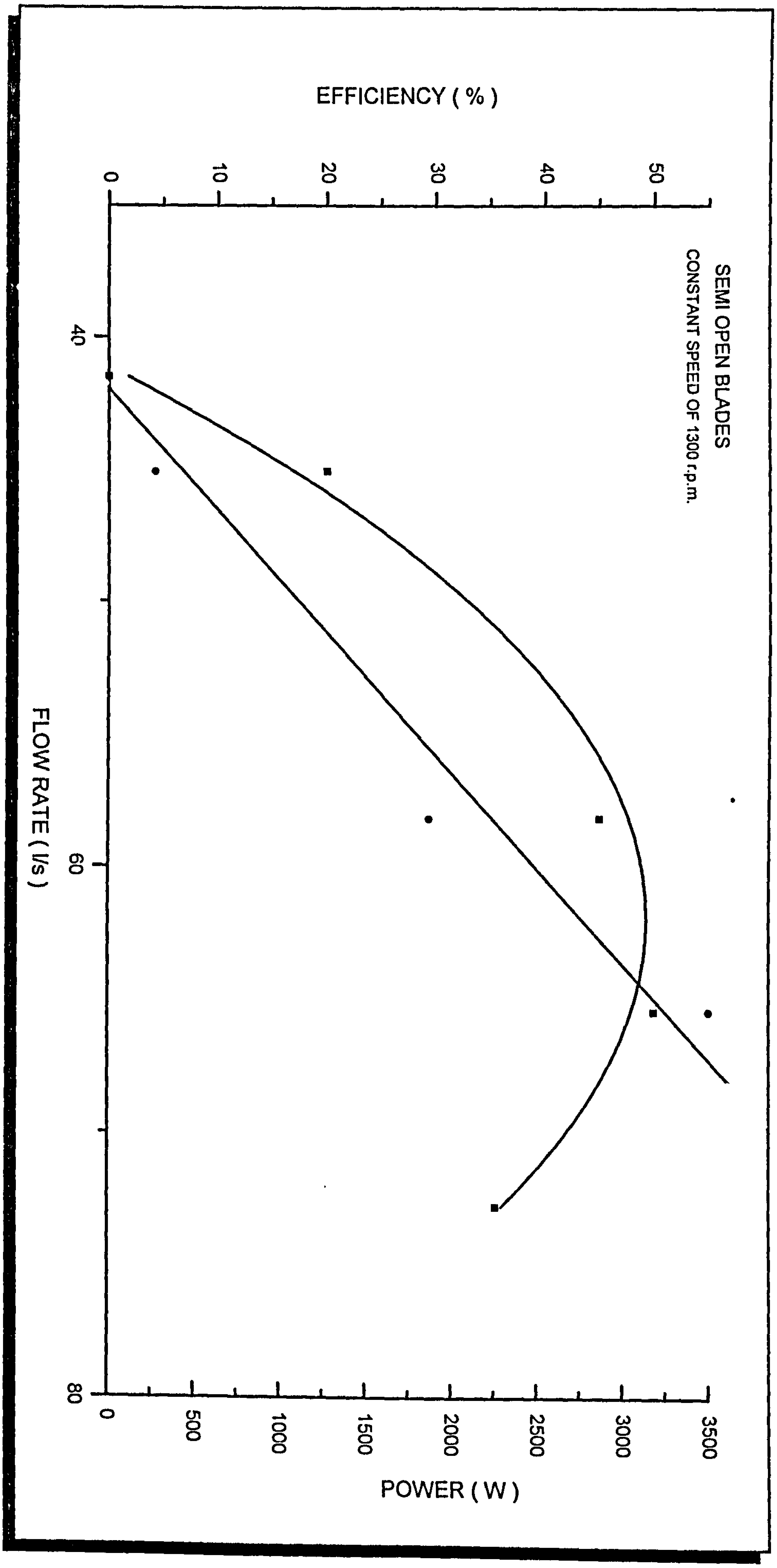

EFFICIENCY ( % )
0
10
20
30
40
50
SEMI OPEN BLADES
CONSTANT SPEED OF 1300 r.p.m.
40
60
80
FLOW RATE ( l/s )
0
500
1000
1500
2000
2500
3000
3500
POWER ( W )

Figure ( 6.12.2 )

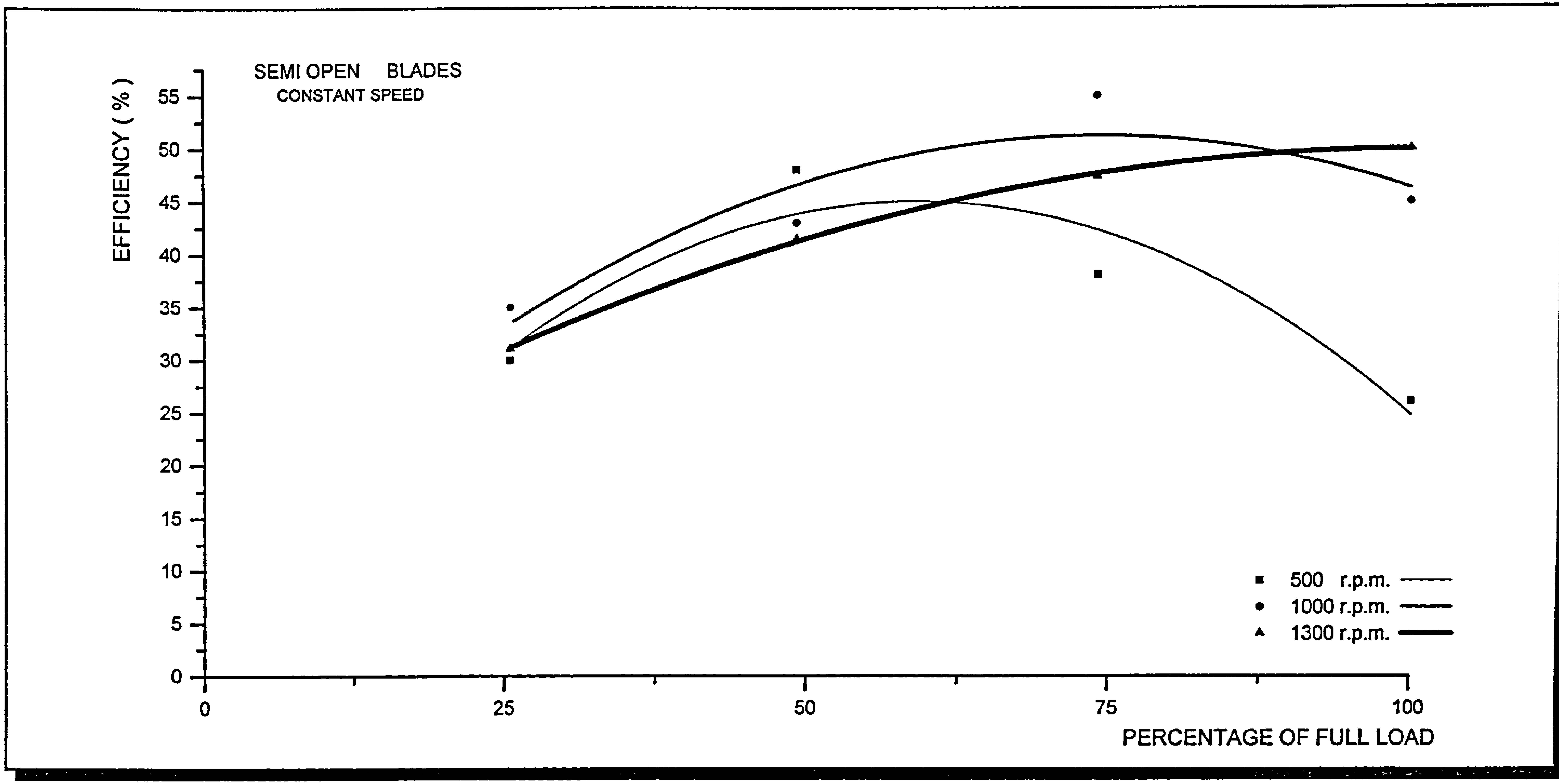

EFFICIENCY ( % )
55
50
45
40
35
30
25
20
15
10
5
0
SEMI OPEN BLADES
CONSTANT SPEED
0
25
50
75
100
PERCENTAGE OF FULL LOAD
500 r.p.m.
1000 r.p.m.
1300 r.p.m.

Figure ( 6.13.2 )

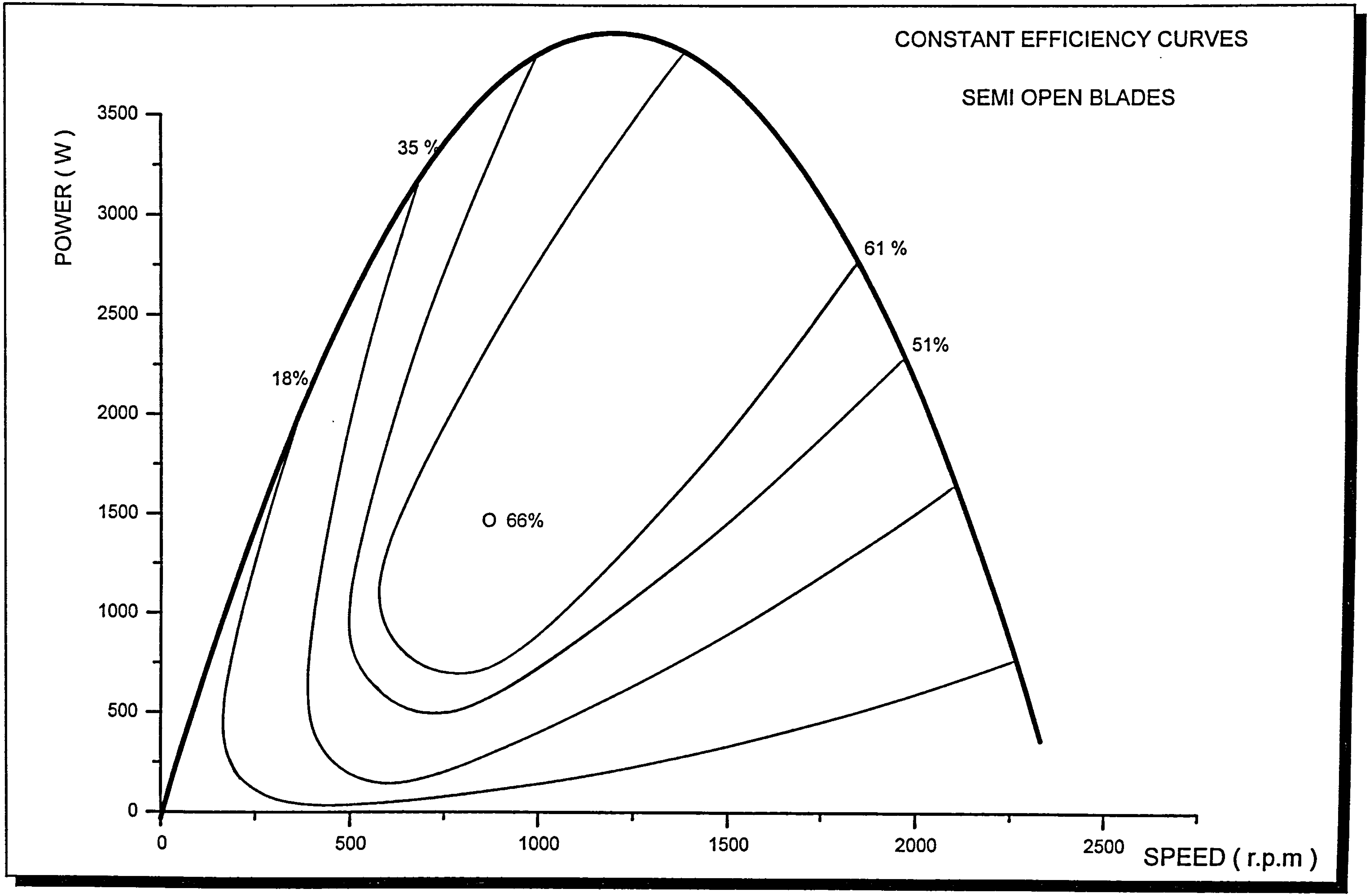

CONSTANT EFFICIENCY CURVES
SEMI OPEN BLADES
POWER ( W )
3500
3000
2500
2000
1500
1000
500
0
0
500
1000
1500
2000
2500
SPEED ( r.p.m )
35 %
18%
61 %
51%
O 66%

Figure ( 6.14.2 )

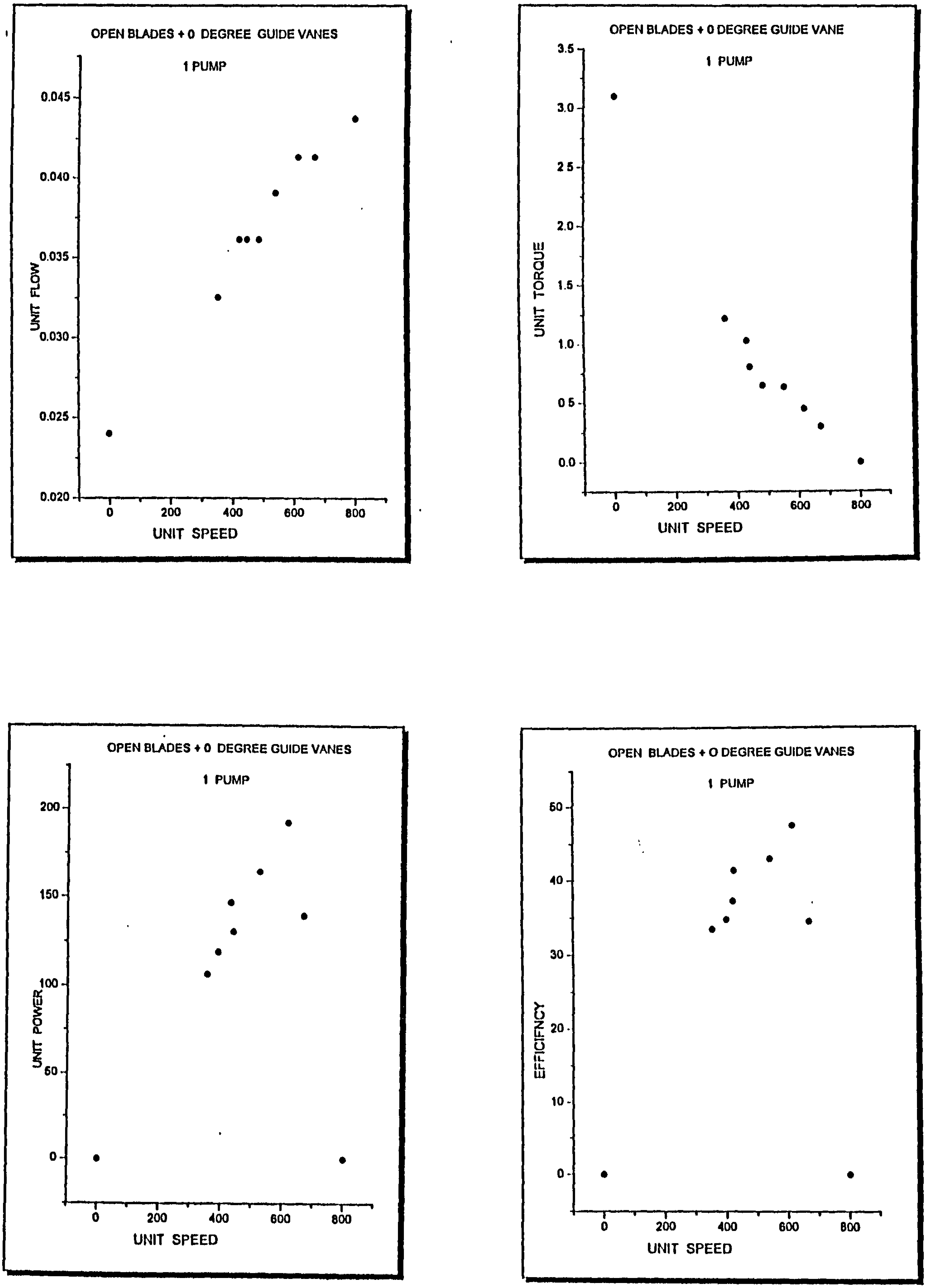

OPEN BLADES + 0 DEGREE GUIDE VANES
1 PUMP
UNIT FLOW
0.045
0.040
0.035
0.030
0.025
0.020
0
200
400
600
800
UNIT SPEED
OPEN BLADES + 0 DEGREE GUIDE VANE
1 PUMP
UNIT TORQUE
3.5
3.0
2.5
2.0
1.5
1.0
0.5
0.0
0
200
400
600
800
UNIT SPEED
OPEN BLADES + 0 DEGREE GUIDE VANES
1 PUMP
UNIT POWER
200
150
100
50
0
0
200
400
600
800
UNIT SPEED
OPEN BLADES + O DEGREE GUIDE VANES
1 PUMP
EFFICIFNCY
50
40
30
20
10
0
0
200
400
600
800
UNIT SPEED

Figure ( 6.1.3 )

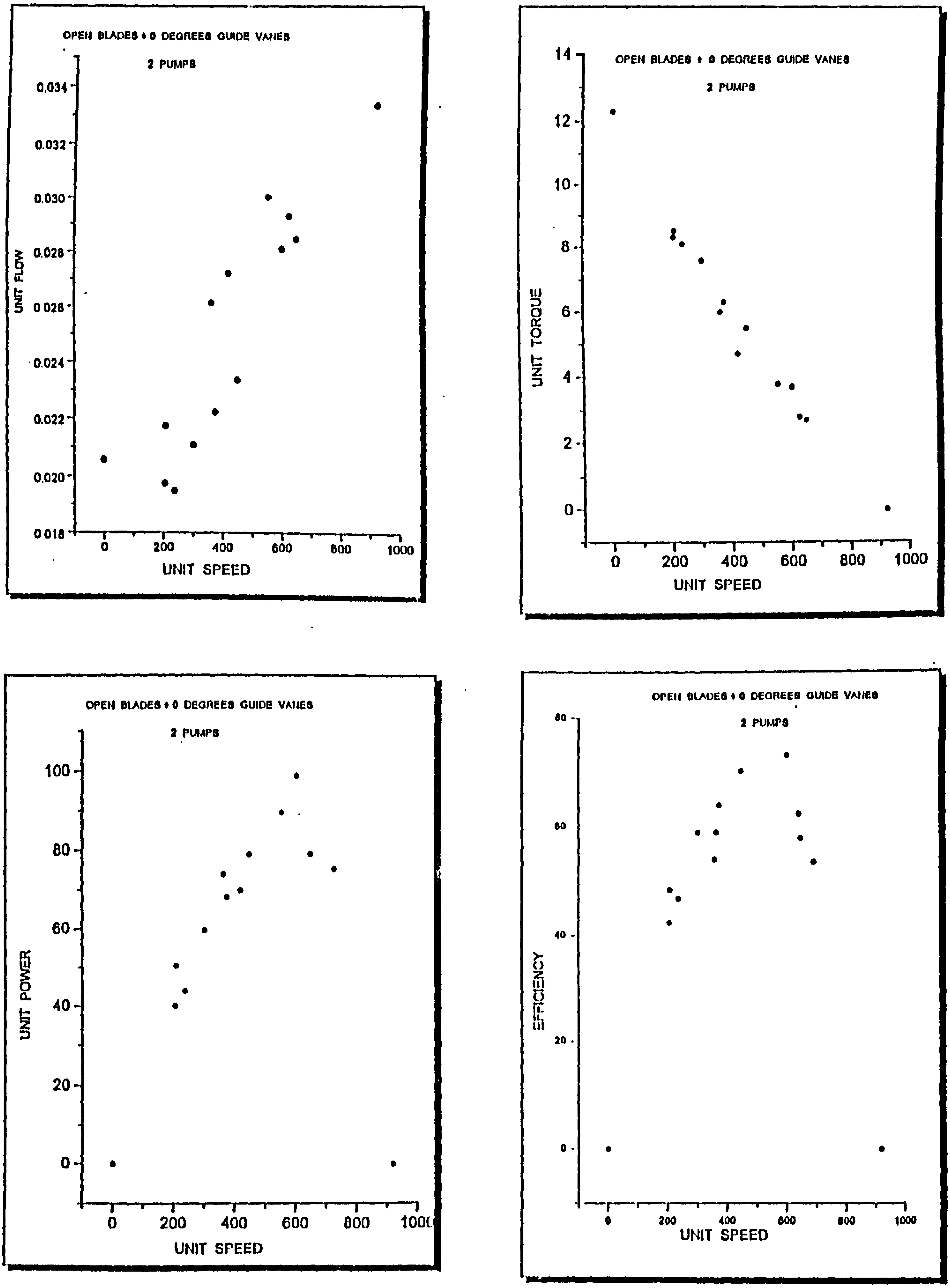

OPEN BLADES + 0 DEGREES GUIDE VANES
2 PUMPS
UNIT FLOW
UNIT SPEED
UNIT TORQUE
UNIT POWER
EFFICIENCY

Figure ( 6.2.3 )

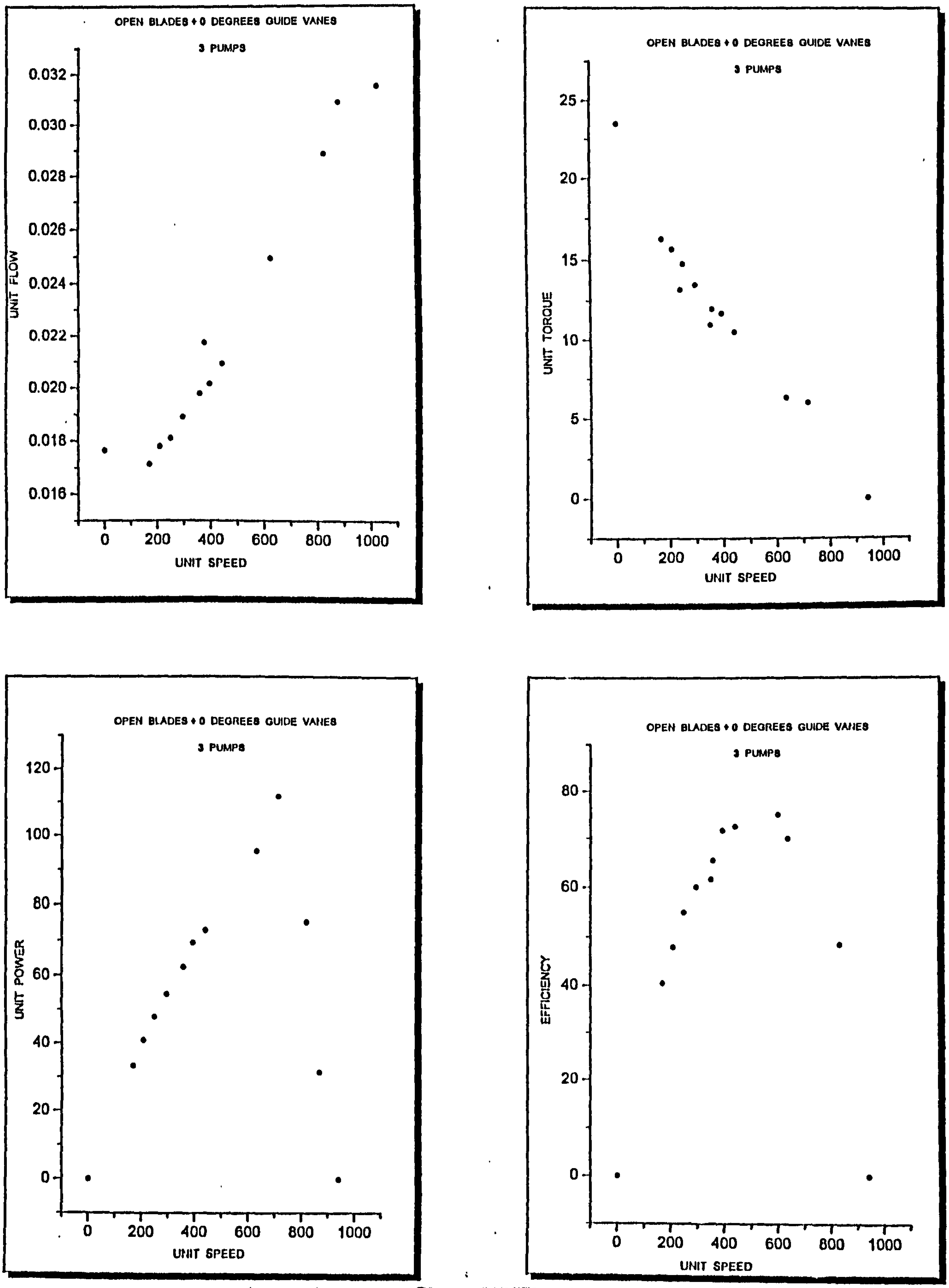

OPEN BLADES + 0 DEGREES GUIDE VANES
3 PUMPS
UNIT FLOW
0.032
0.030
0.028
0.026
0.024
0.022
0.020
0.018
0.016
0 200 400 600 800 1000
UNIT SPEED
OPEN BLADES + 0 DEGREES GUIDE VANES
3 PUMPS
UNIT TORQUE
25
20
15
10
5
0
0 200 400 600 800 1000
UNIT SPEED
OPEN BLADES + 0 DEGREES GUIDE VANES
3 PUMPS
UNIT POWER
120
100
80
60
40
20
0
0 200 400 600 800 1000
UNIT SPEED
OPEN BLADES + 0 DEGREES GUIDE VANES
3 PUMPS
EFFICIENCY
80
60
40
20
0
0 200 400 600 800 1000
UNIT SPEED

Figure ( 6.3.3 )

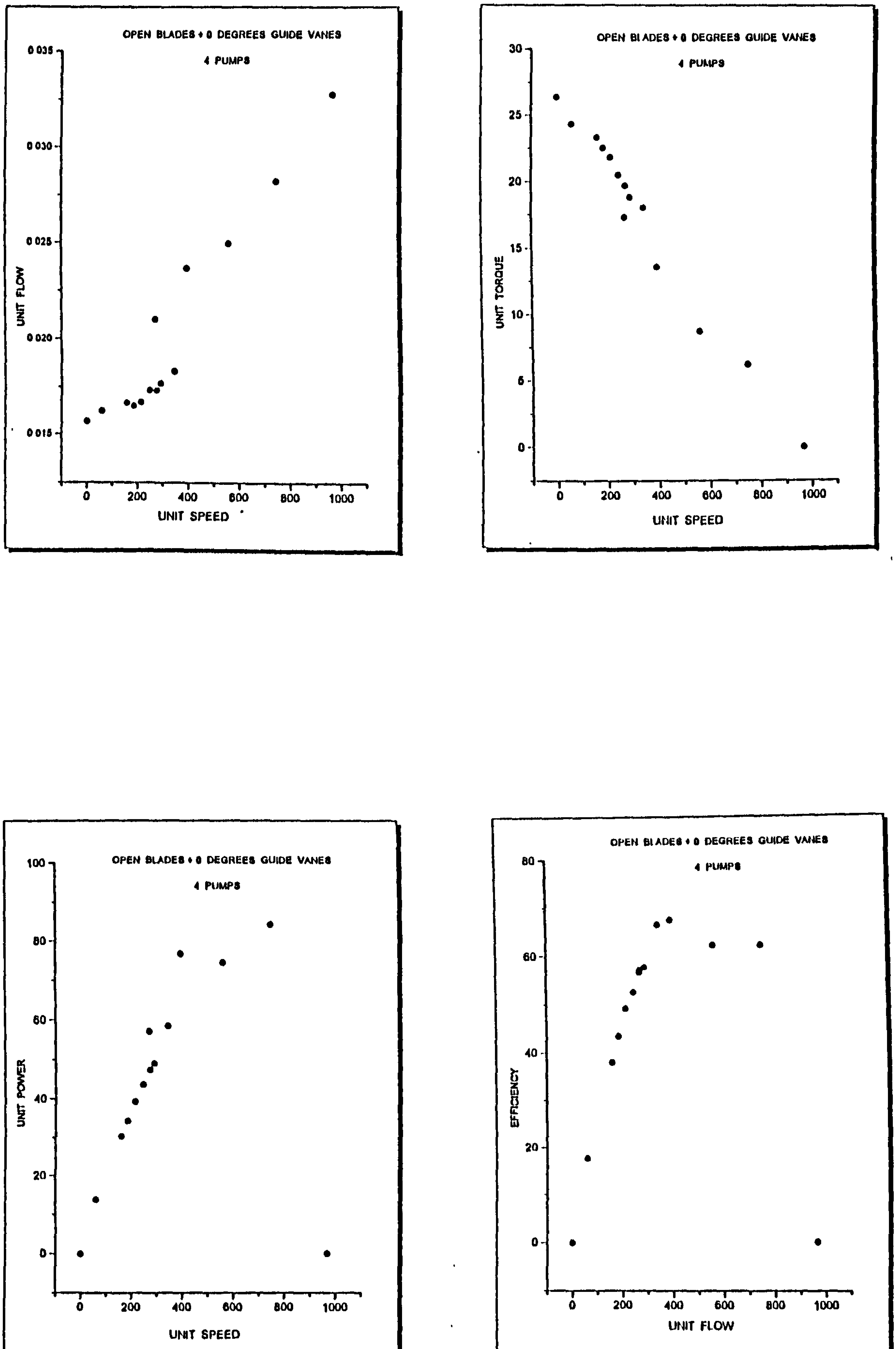
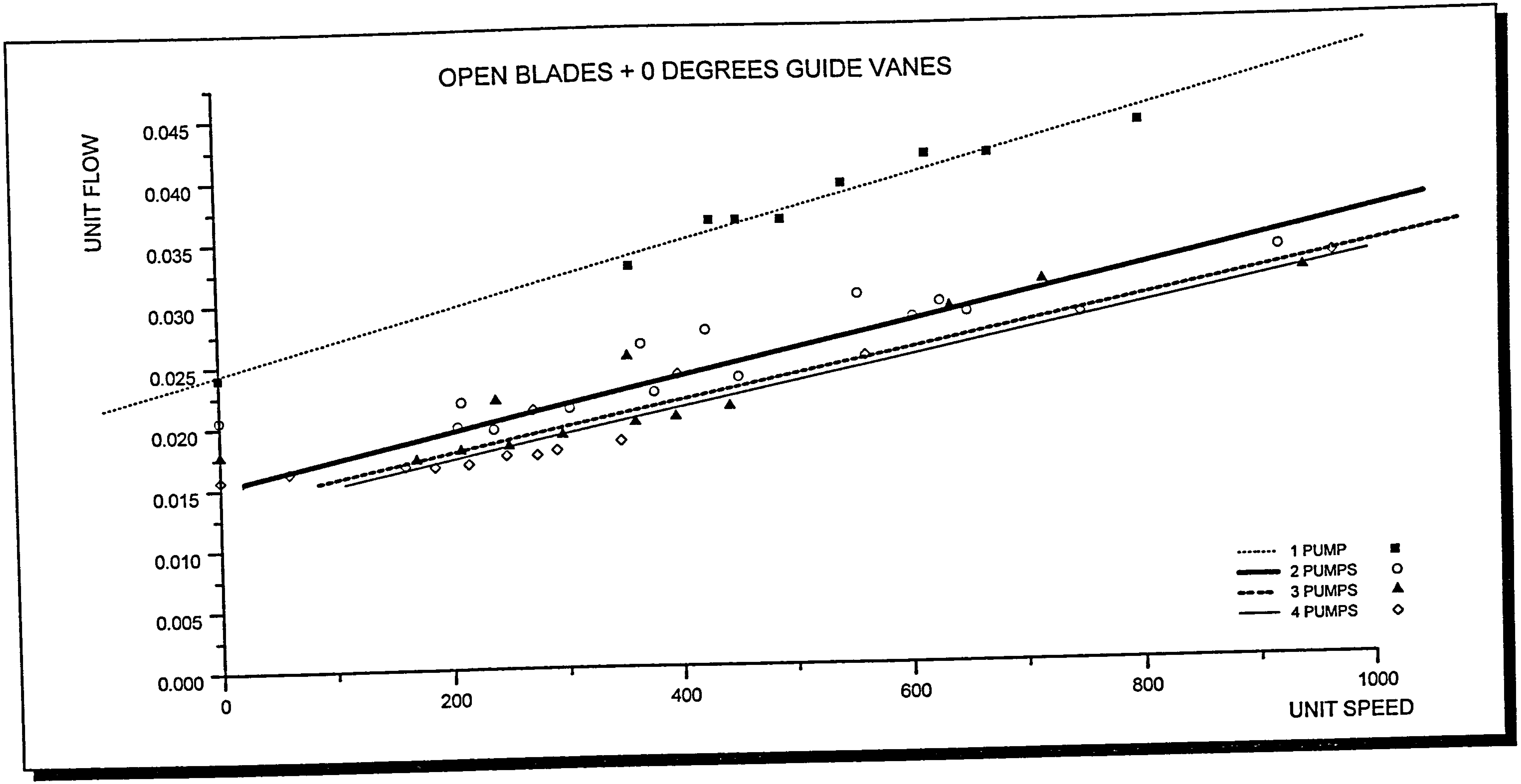

OPEN BLADES + 0 DEGREES GUIDE VANES
4 PUMPS
UNIT FLOW
UNIT SPEED
UNIT TORQUE
UNIT POWER
EFFICIENCY
UNIT FLOW

Figure ( 6.4.3 )



OPEN BLADES + 0 DEGREES GUIDE VANES
UNIT FLOW
0.045
0.040
0.035
0.030
0.025
0.020
0.015
0.010
0.005
0.000
0
200
400
600
800
1000
UNIT SPEED
1 PUMP
2 PUMPS
3 PUMPS
4 PUMPS


Figure ( 6.5.3 )

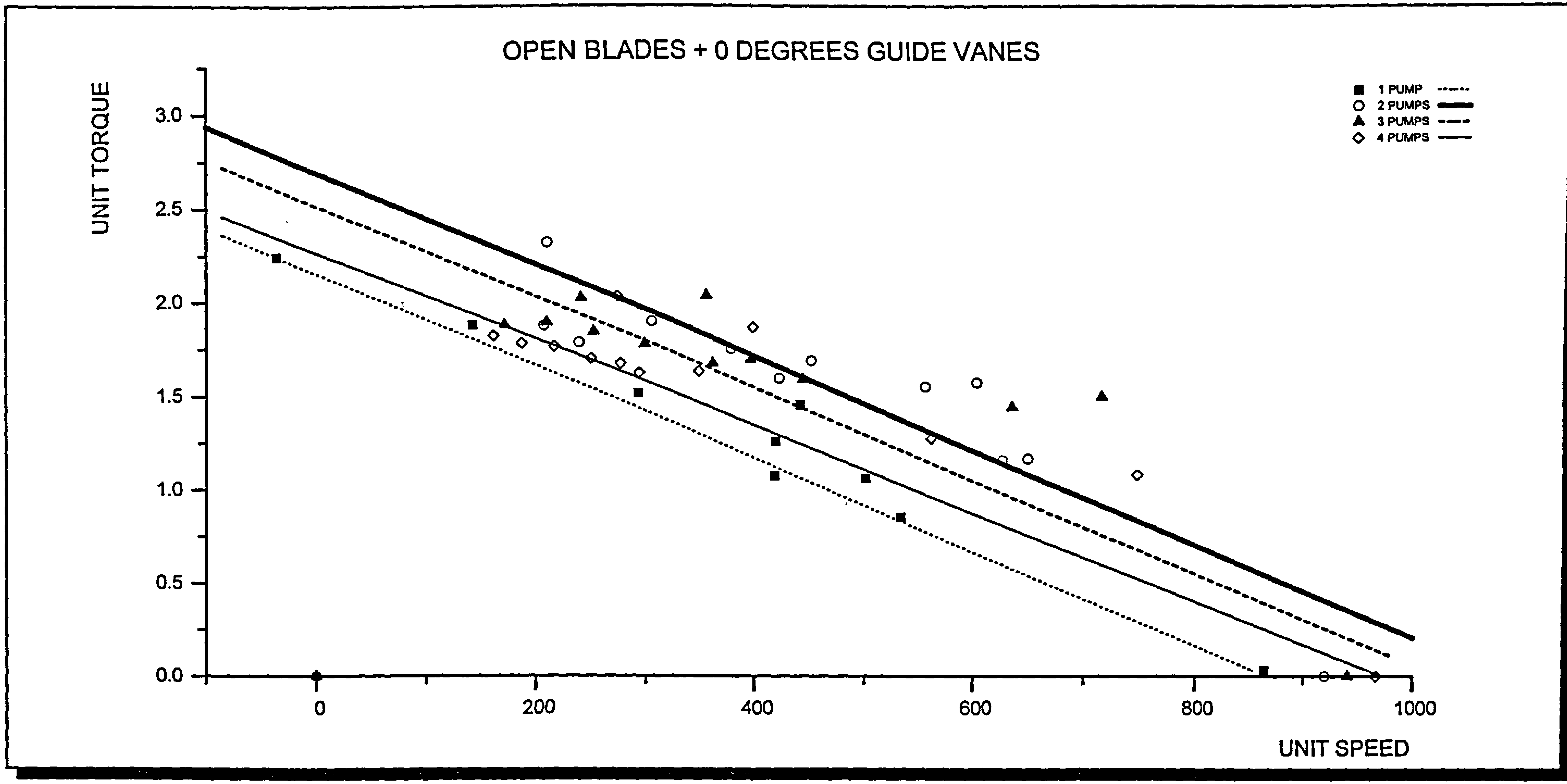

OPEN BLADES + 0 DEGREES GUIDE VANES
UNIT TORQUE
3.0
2.5
2.0
1.5
1.0
0.5
0.0
0
200
400
600
800
1000
UNIT SPEED
1 PUMP
2 PUMPS
3 PUMPS
4 PUMPS

Figure ( 6.6.3 )

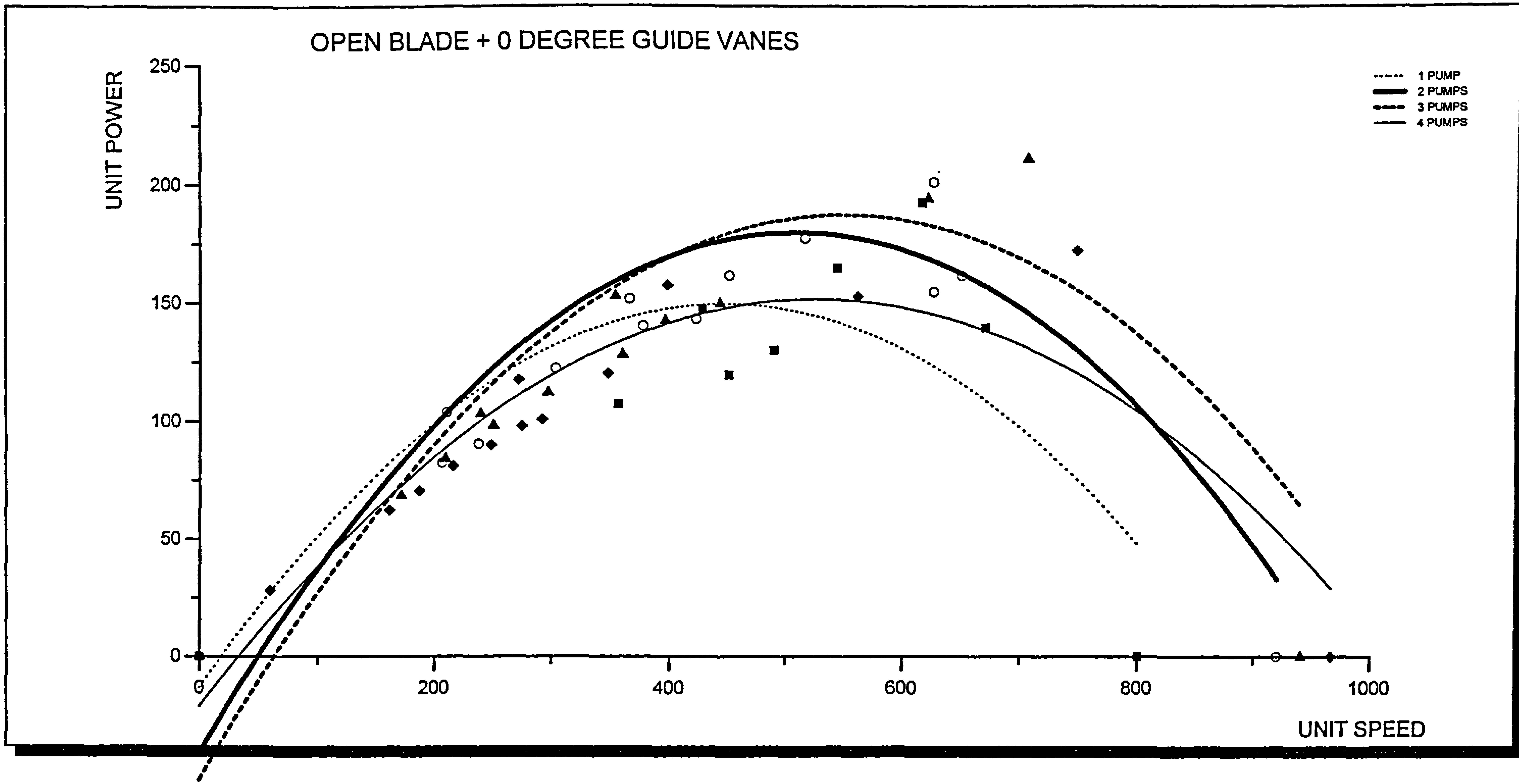

OPEN BLADE + 0 DEGREE GUIDE VANES
UNIT POWER
250
200
150
100
50
0
0
200
400
600
800
1000
UNIT SPEED
1 PUMP
2 PUMPS
3 PUMPS
4 PUMPS

Figure ( 6.7.3 )

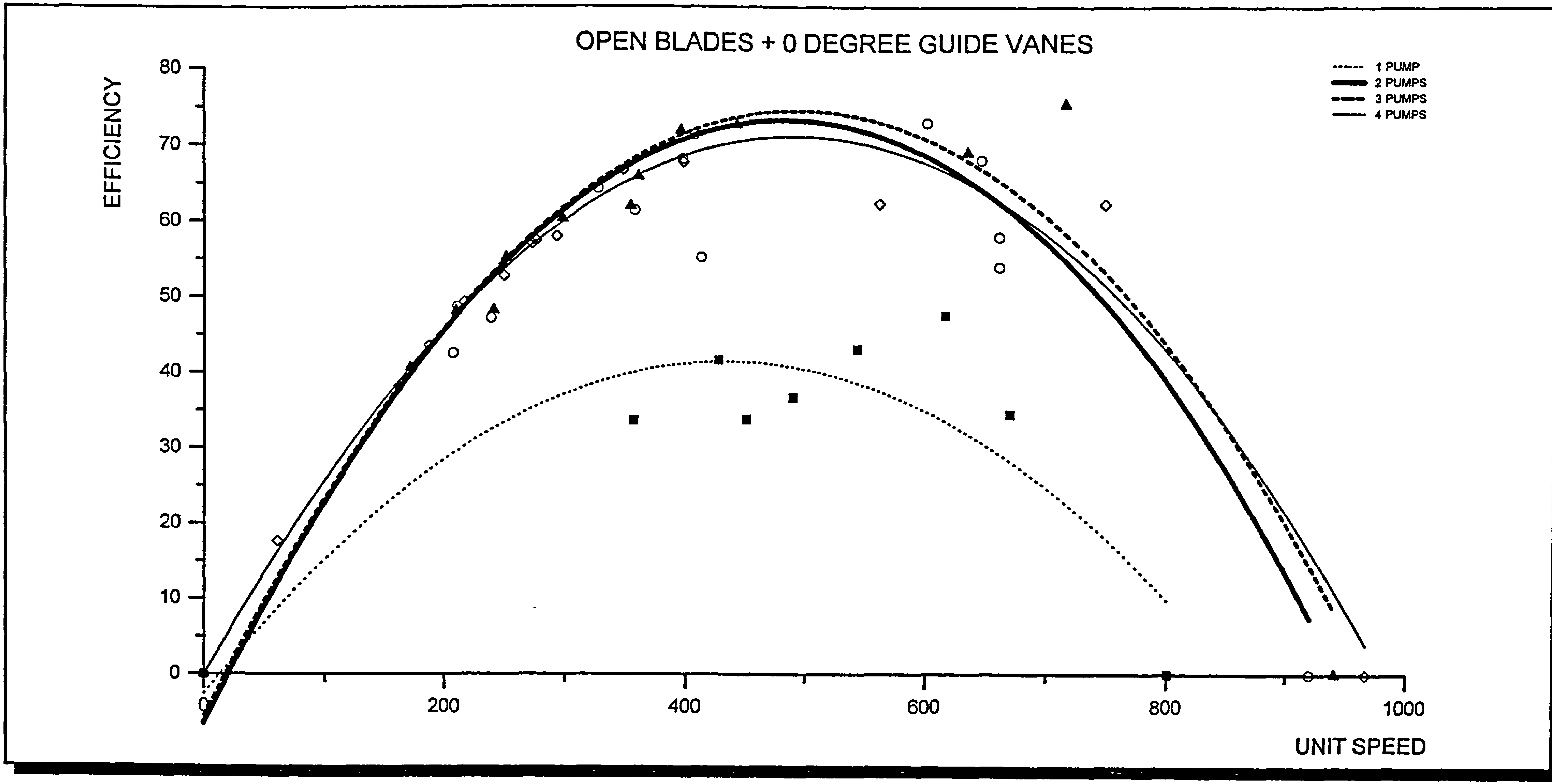

OPEN BLADES + 0 DEGREE GUIDE VANES
EFFICIENCY
80
70
60
50
40
30
20
10
0
0
200
400
600
800
1000
UNIT SPEED
1 PUMP
2 PUMPS
3 PUMPS
4 PUMPS

Figure ( 6.8.3 )

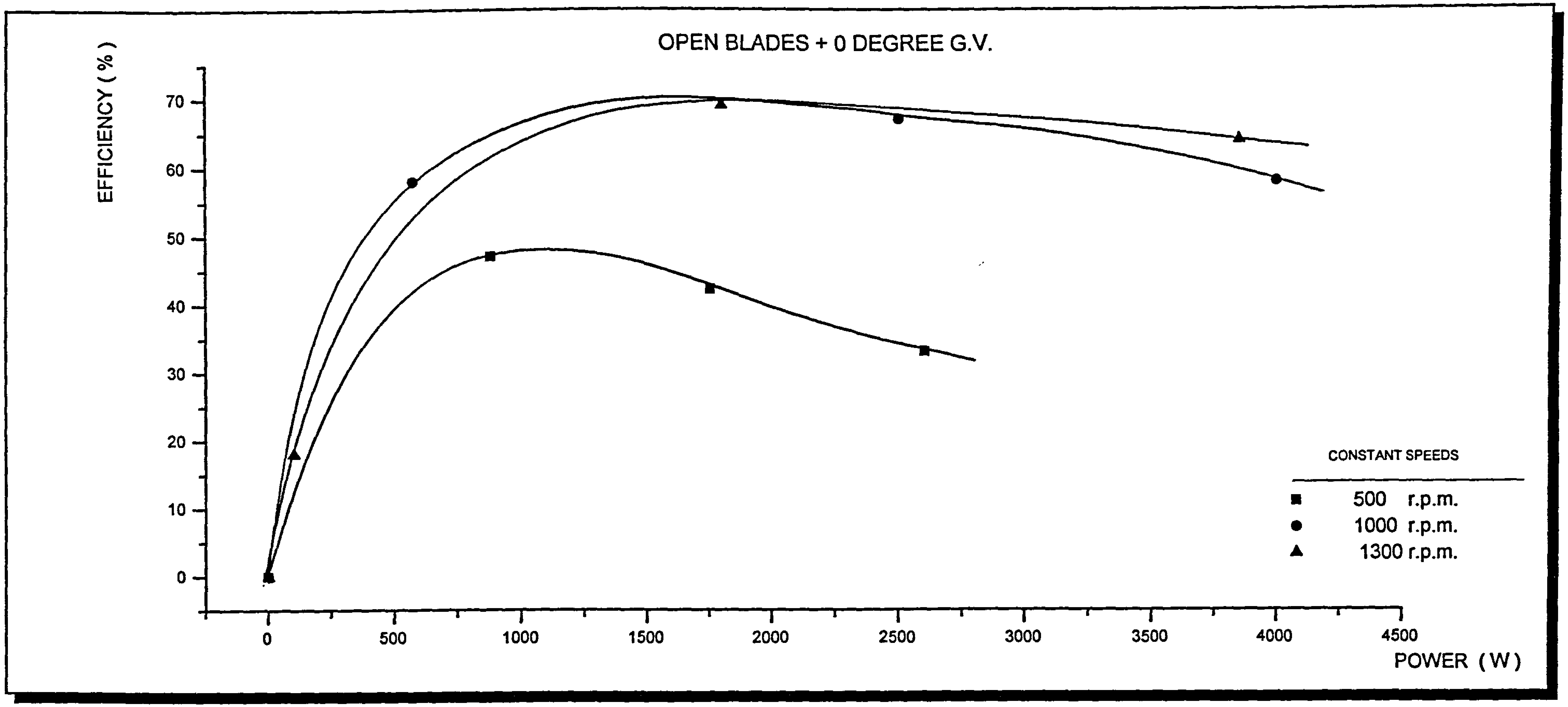
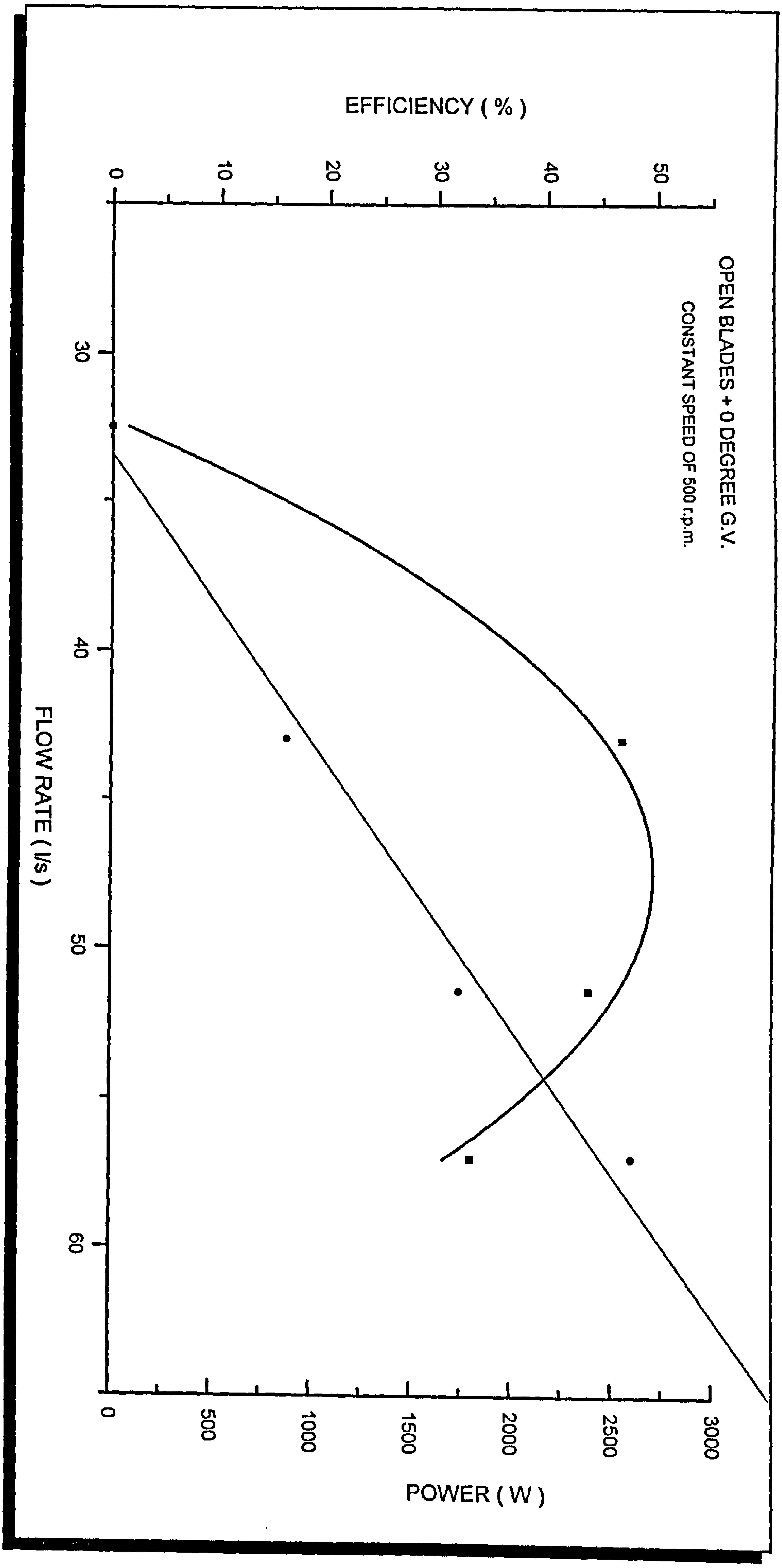
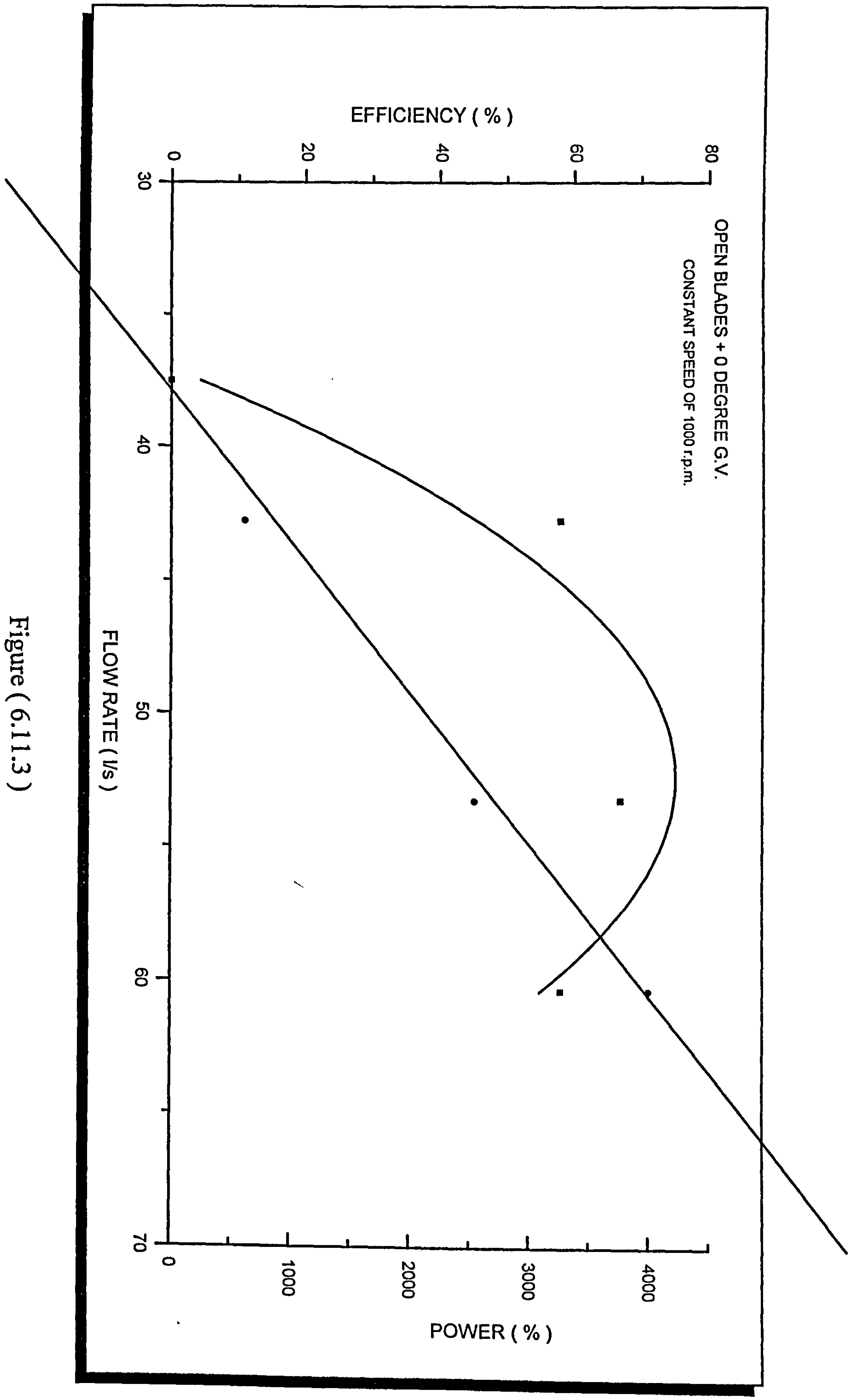

OPEN BLADES + 0 DEGREE G.V.
EFFICIENCY ( % )
0
10
20
30
40
50
60
70
0
500
1000
1500
2000
2500
3000
3500
4000
4500
POWER ( W )
CONSTANT SPEEDS
500 r.p.m.
1000 r.p.m.
1300 r.p.m.

Figure ( 6.9.3 )



OPEN BLADES + 0 DEGREE G.V.
CONSTANT SPEED OF 500 r.p.m.
EFFICIENCY ( % )
0
10
20
30
40
50
FLOW RATE ( l/s )
30
40
50
60
POWER ( W )
0
500
1000
1500
2000
2500
3000


Figure ( 6.10.3 )



OPEN BLADES + 0 DEGREE G.V.
CONSTANT SPEED OF 1000 r.p.m.
EFFICIENCY ( % )
0
20
40
60
80
30
40
50
60
70
FLOW RATE ( l/s )
0
1000
2000
3000
4000
POWER ( % )


Figure ( 6.11.3 )

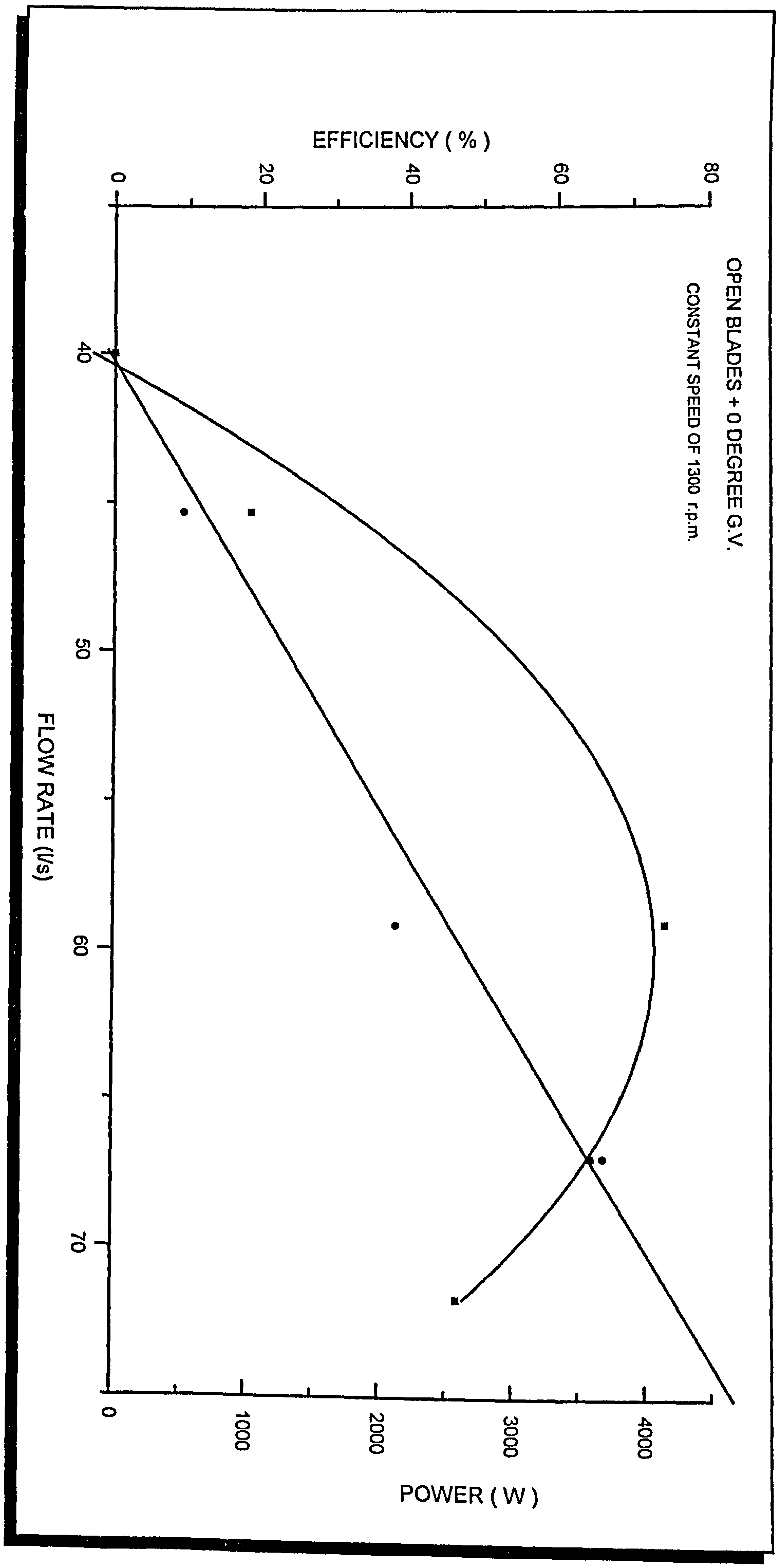

OPEN BLADES + 0 DEGREE G.V.
CONSTANT SPEED OF 1300 r.p.m.
EFFICIENCY ( % )
0
20
40
60
80
40
50
60
70
FLOW RATE (l/s)
0
1000
2000
3000
4000
POWER ( W )

Figure ( 6.12.3 )

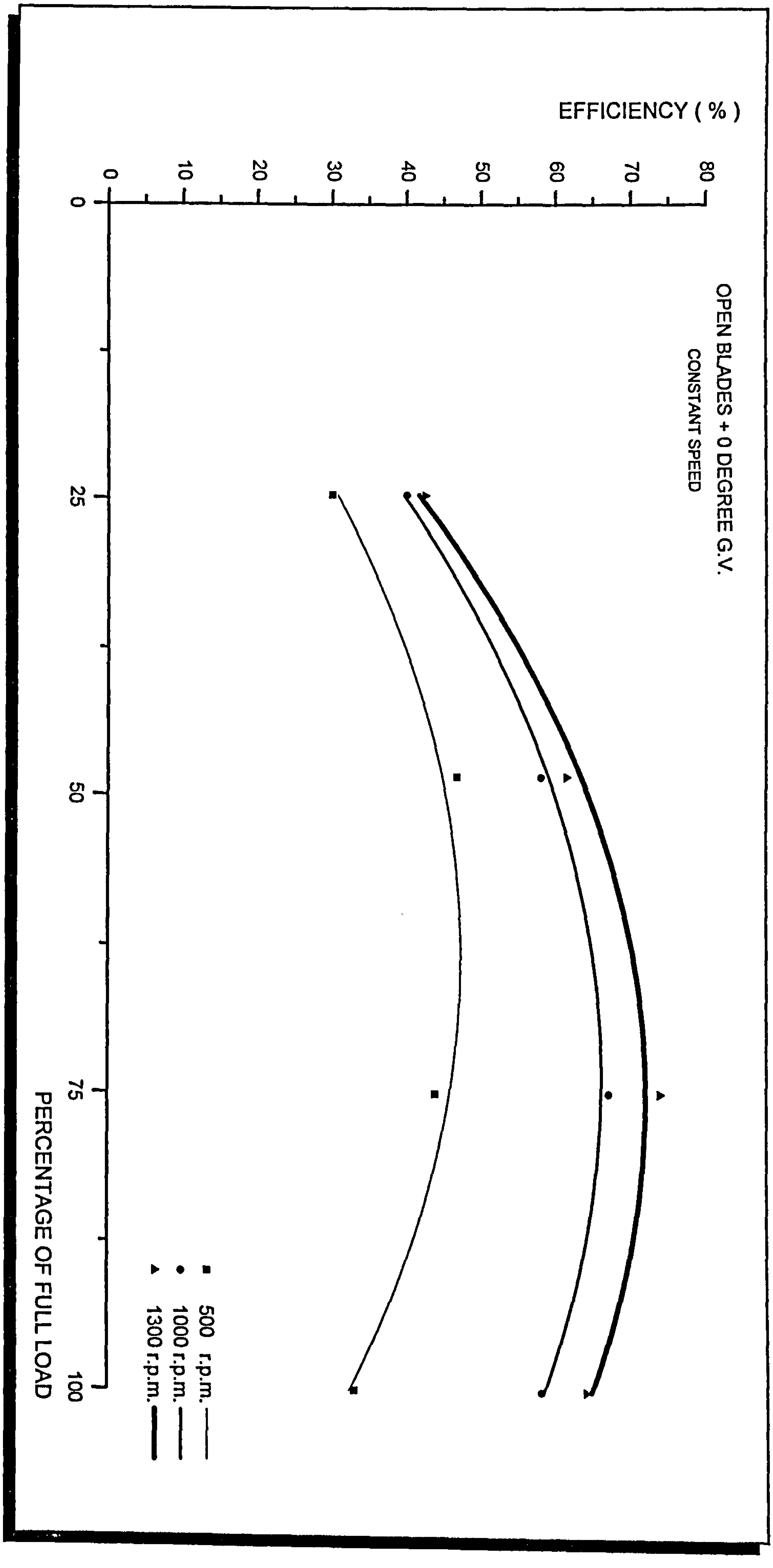

EFFICIENCY ( % )
OPEN BLADES + 0 DEGREE G.V.
CONSTANT SPEED
0
10
20
30
40
50
60
70
80
0
25
50
75
100
PERCENTAGE OF FULL LOAD
500 r.p.m.
1000 r.p.m.
1300 r.p.m.

Figure ( 6.13.3 )

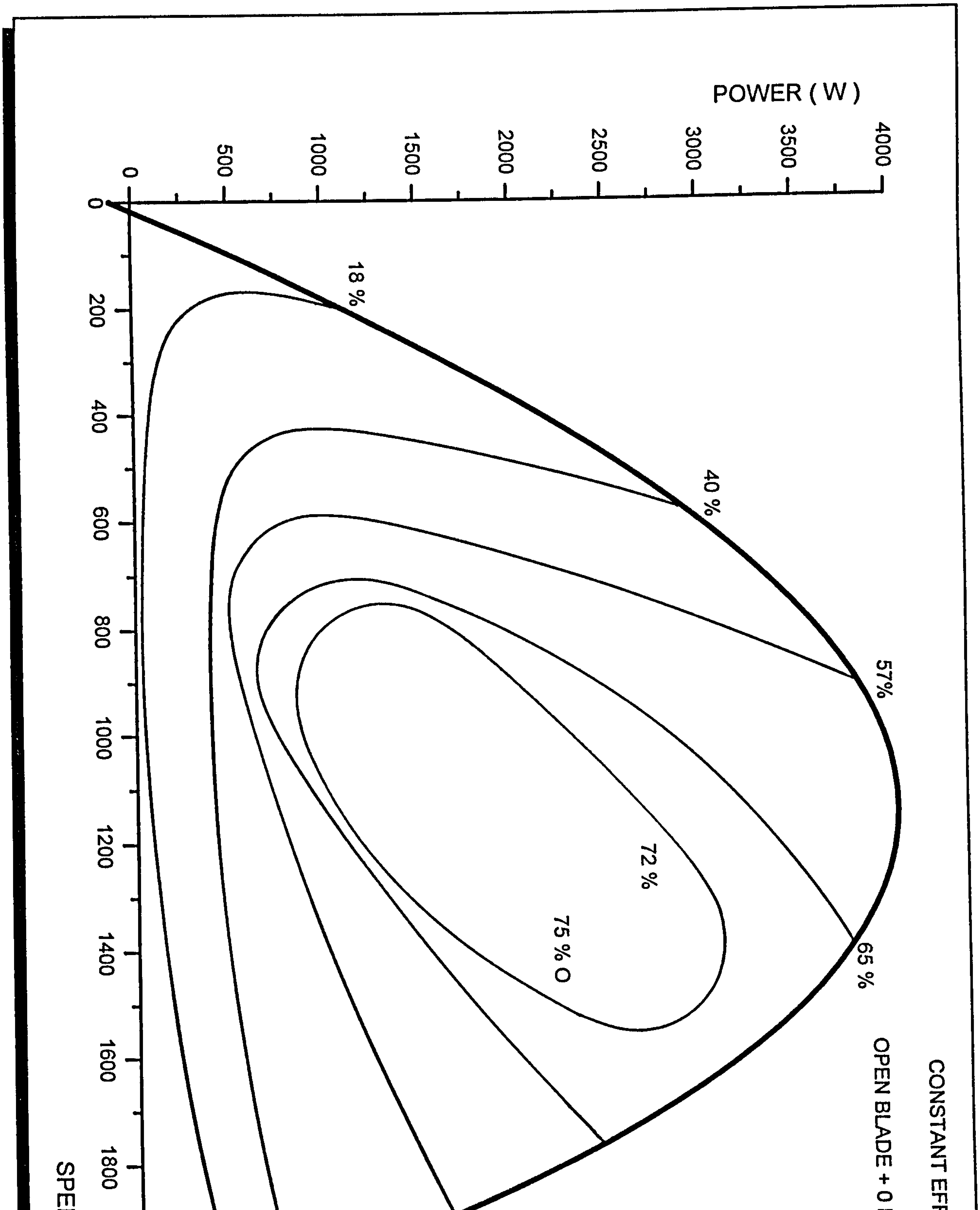

POWER ( W )
4000
3500
3000
2500
2000
1500
1000
500
0
0
200
400
600
800
1000
1200
1400
1600
1800
2000
SPEED ( r.p.m )
CONSTANT EFFICIENCY CURVES
OPEN BLADE + 0 DEGREE GUIDE VANES
18 %
40 %
57%
65 %
72 %
75 % O

Figure ( 6.14.3 )

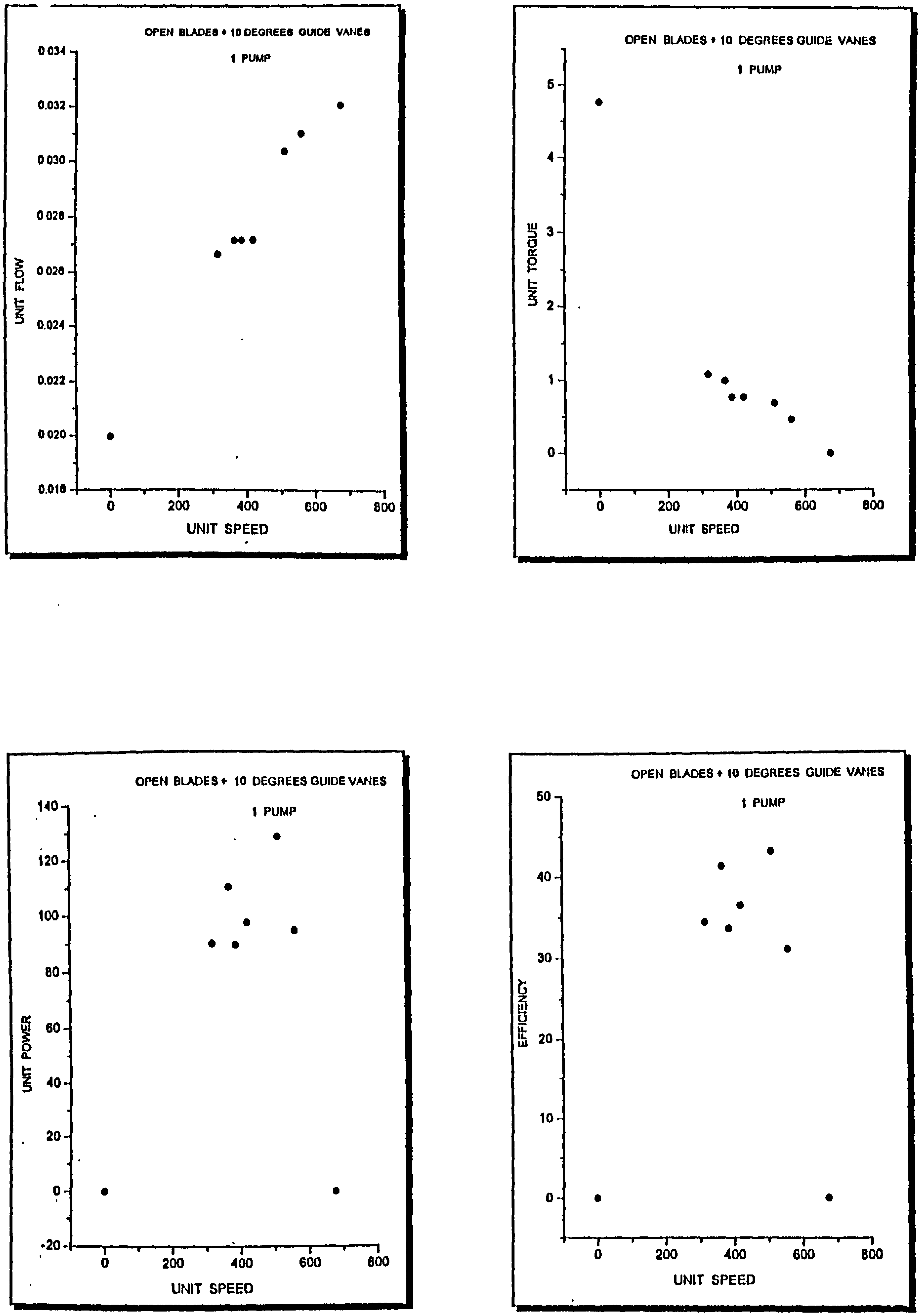

OPEN BLADES + 10 DEGREES GUIDE VANES
1 PUMP
UNIT FLOW
UNIT SPEED
OPEN BLADES + 10 DEGREES GUIDE VANES
1 PUMP
UNIT TORQUE
UNIT SPEED
OPEN BLADES + 10 DEGREES GUIDE VANES
1 PUMP
UNIT POWER
UNIT SPEED
OPEN BLADES + 10 DEGREES GUIDE VANES
1 PUMP
EFFICIENCY
UNIT SPEED

Figure ( 6.1.4 )

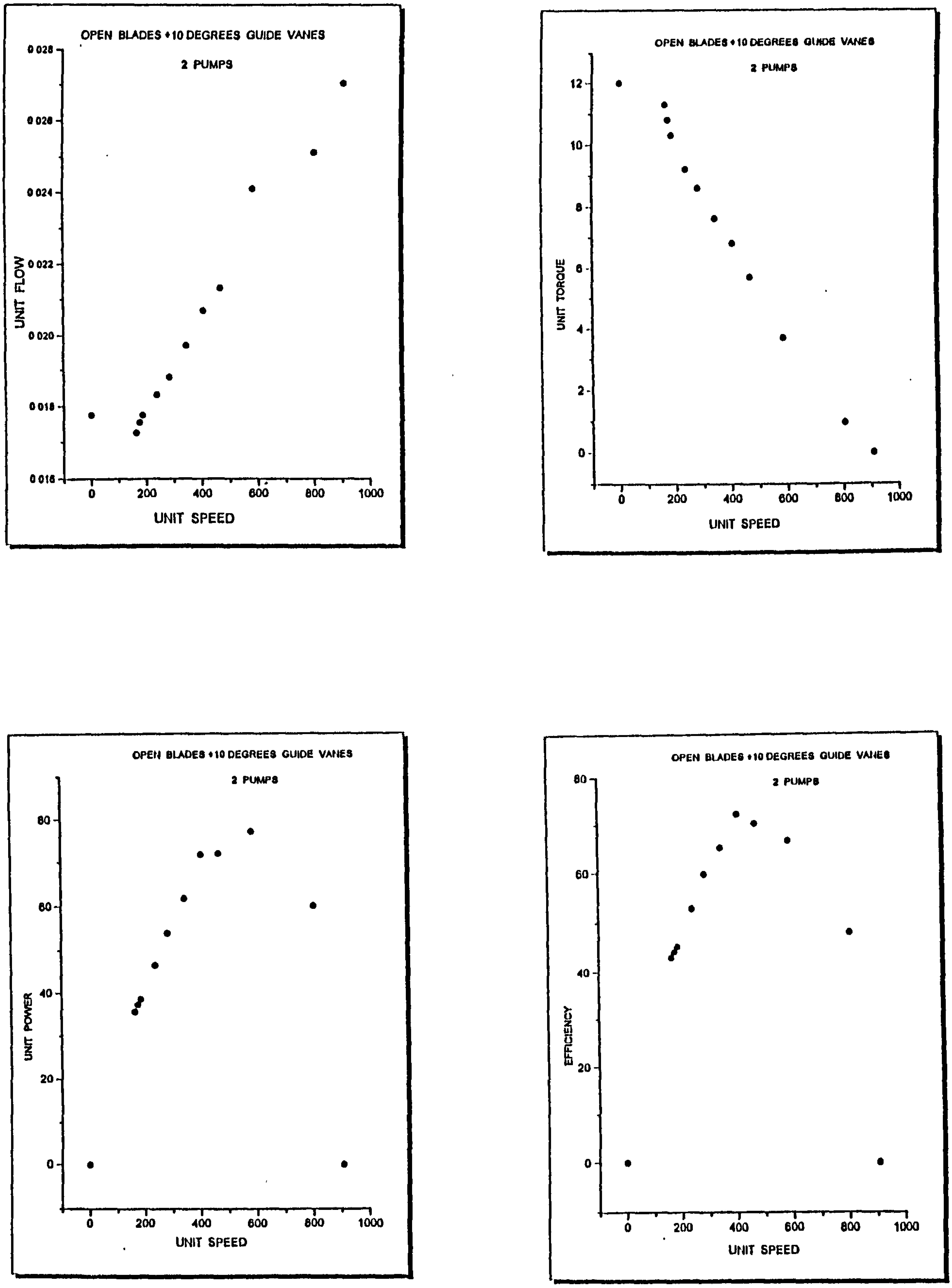
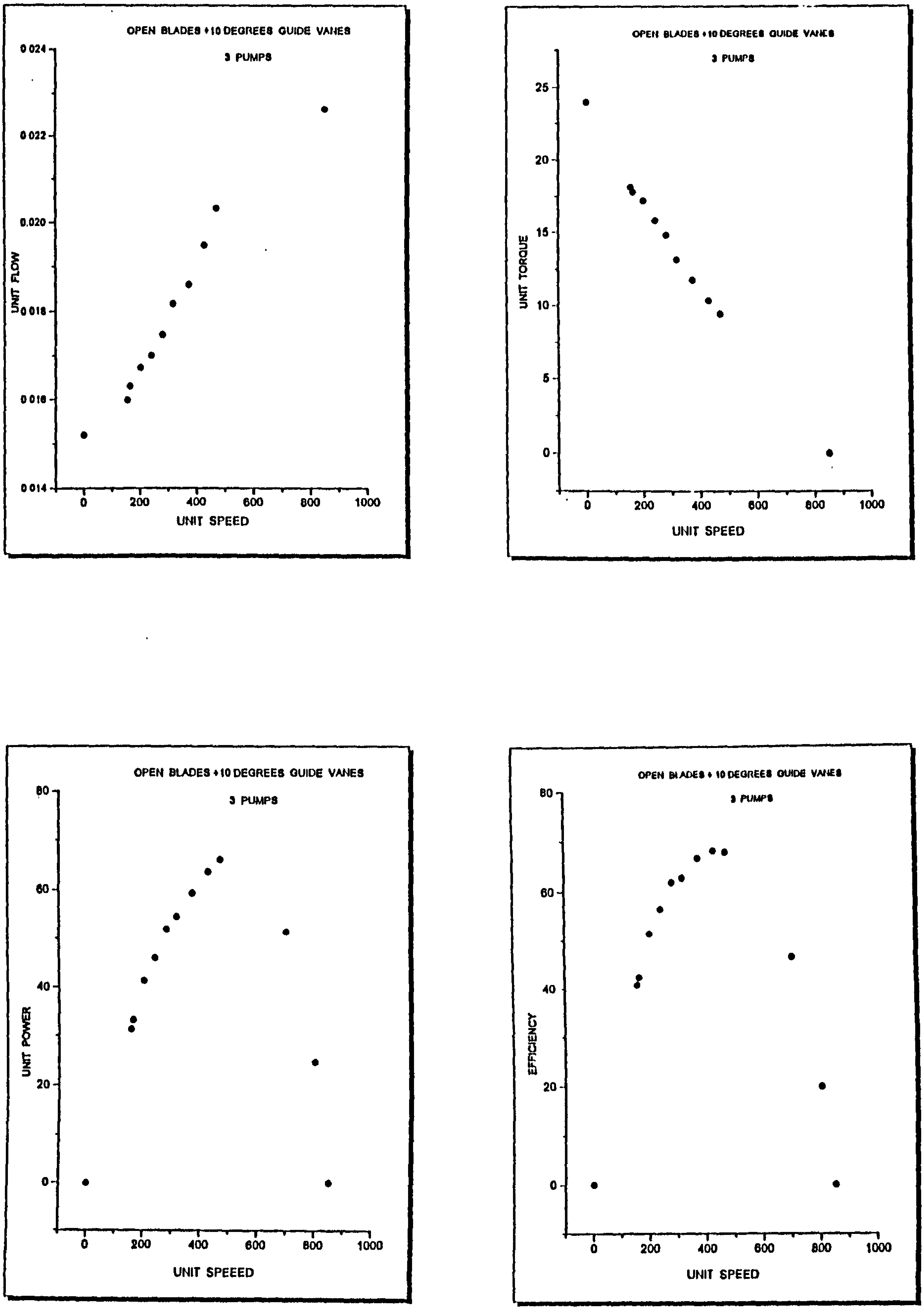

OPEN BLADES +10 DEGREES GUIDE VANES
2 PUMPS
UNIT FLOW
UNIT SPEED
UNIT TORQUE
UNIT POWER
EFFICIENCY

Figure ( 6.2.4 )



OPEN BLADES +10 DEGREES GUIDE VANES
3 PUMPS
UNIT FLOW
UNIT SPEED
OPEN BLADES +10 DEGREES GUIDE VANES
3 PUMPS
UNIT TORQUE
UNIT SPEED
OPEN BLADES +10 DEGREES GUIDE VANES
3 PUMPS
UNIT POWER
UNIT SPEED
OPEN BLADES +10 DEGREES GUIDE VANES
3 PUMPS
EFFICIENCY
UNIT SPEED


Figure ( 6.3.4 )

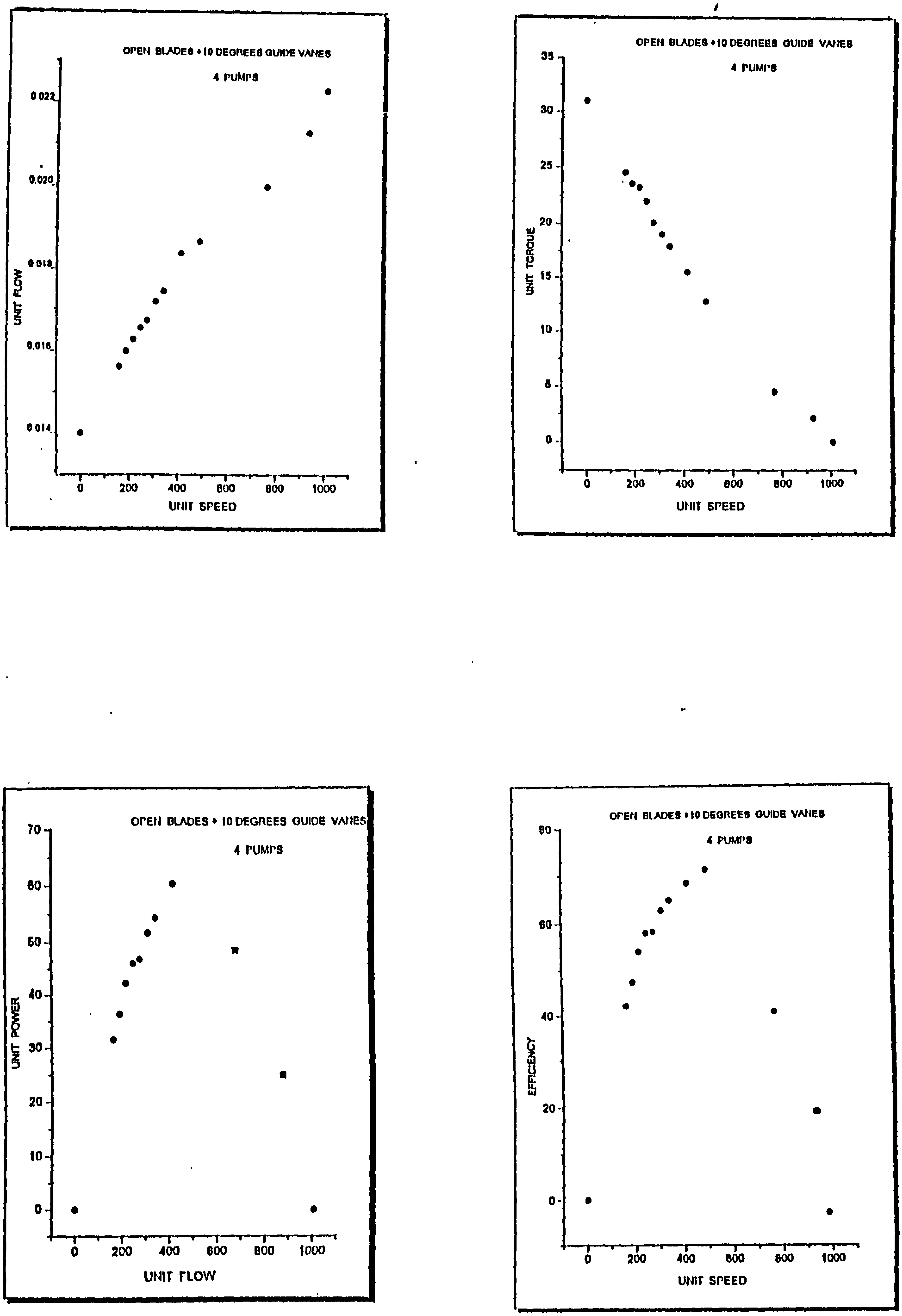
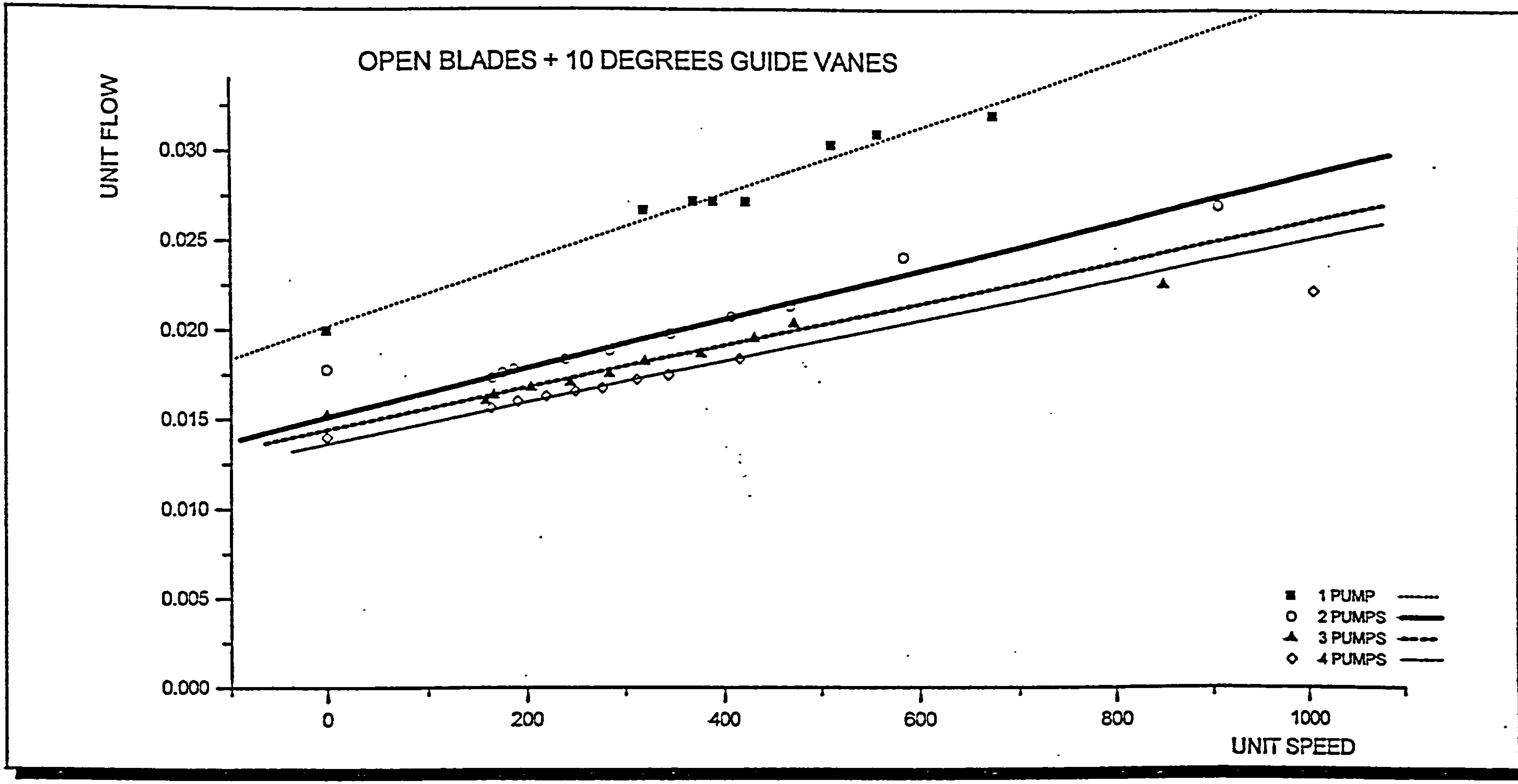

OPEN BLADES + 10 DEGREES GUIDE VANES
4 PUMPS
UNIT FLOW
0.022
0.020
0.018
0.016
0.014
0
200
400
600
800
1000
UNIT SPEED
OPEN BLADES + 10 DEGREES GUIDE VANES
4 PUMPS
UNIT TORQUE
35
30
25
20
15
10
5
0
UNIT SPEED
OPEN BLADES + 10 DEGREES GUIDE VANES
4 PUMPS
UNIT POWER
70
60
50
40
30
20
10
0
UNIT FLOW
OPEN BLADES + 10 DEGREES GUIDE VANES
4 PUMPS
EFFICIENCY
80
60
40
20
0
UNIT SPEED

Figure ( 6.4.4 )



OPEN BLADES + 10 DEGREES GUIDE VANES
UNIT FLOW
0.030
0.025
0.020
0.015
0.010
0.005
0.000
0
200
400
600
800
1000
UNIT SPEED
1 PUMP
2 PUMPS
3 PUMPS
4 PUMPS


Figure ( 6.5.4 )

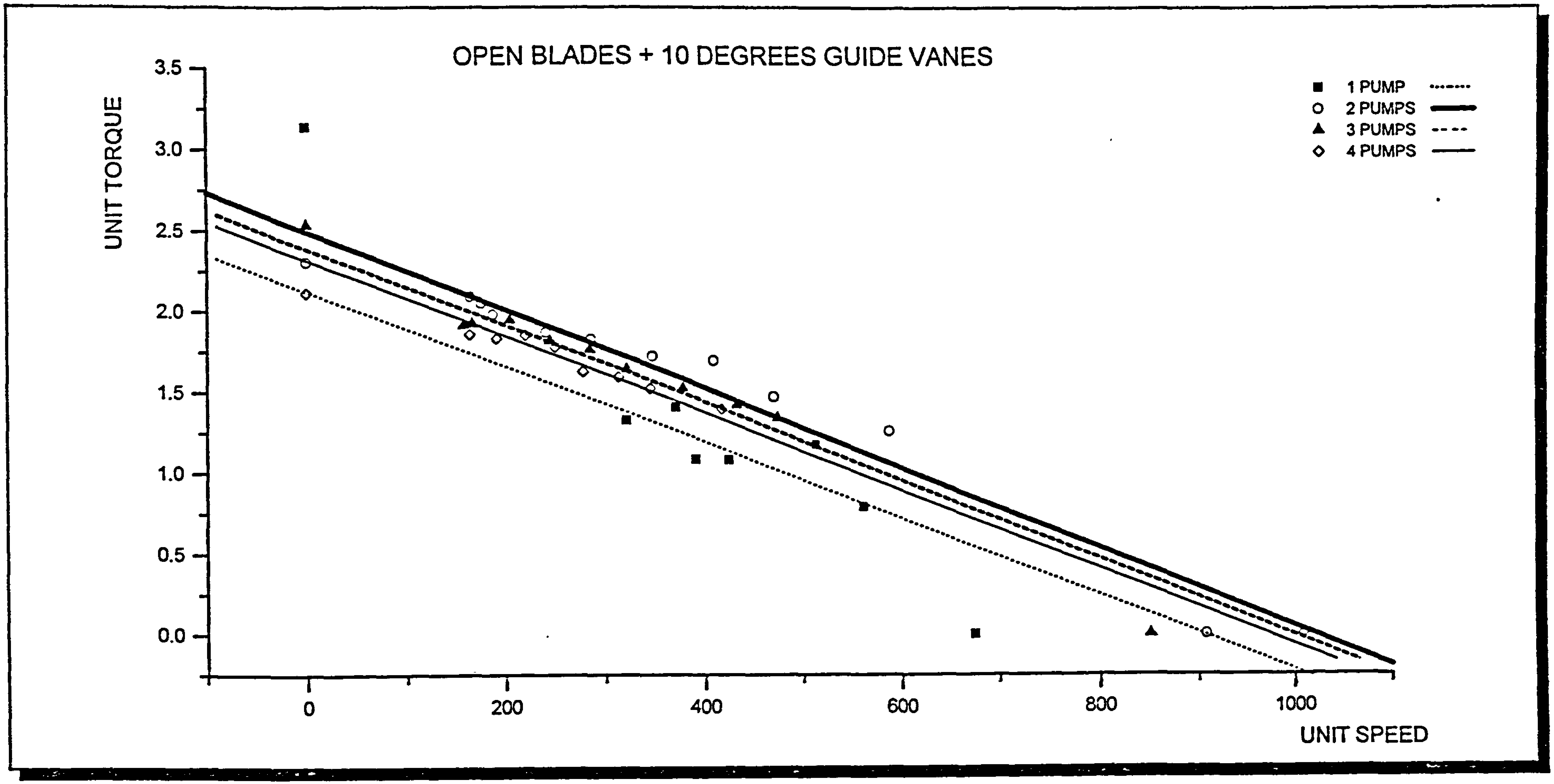

OPEN BLADES + 10 DEGREES GUIDE VANES
UNIT TORQUE
3.5
3.0
2.5
2.0
1.5
1.0
0.5
0.0
0
200
400
600
800
1000
UNIT SPEED
1 PUMP
2 PUMPS
3 PUMPS
4 PUMPS

Figure ( 6.6.4 )

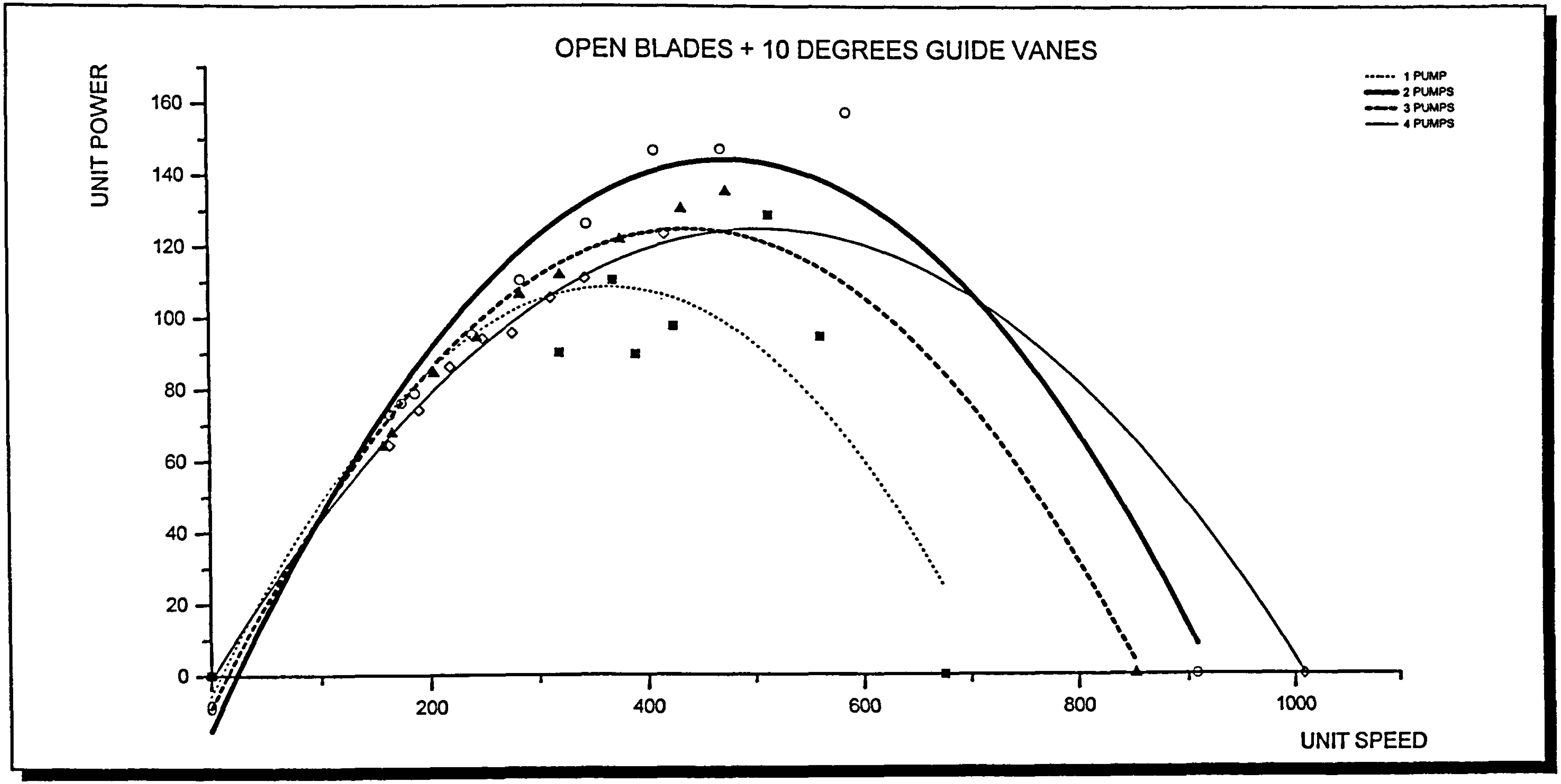
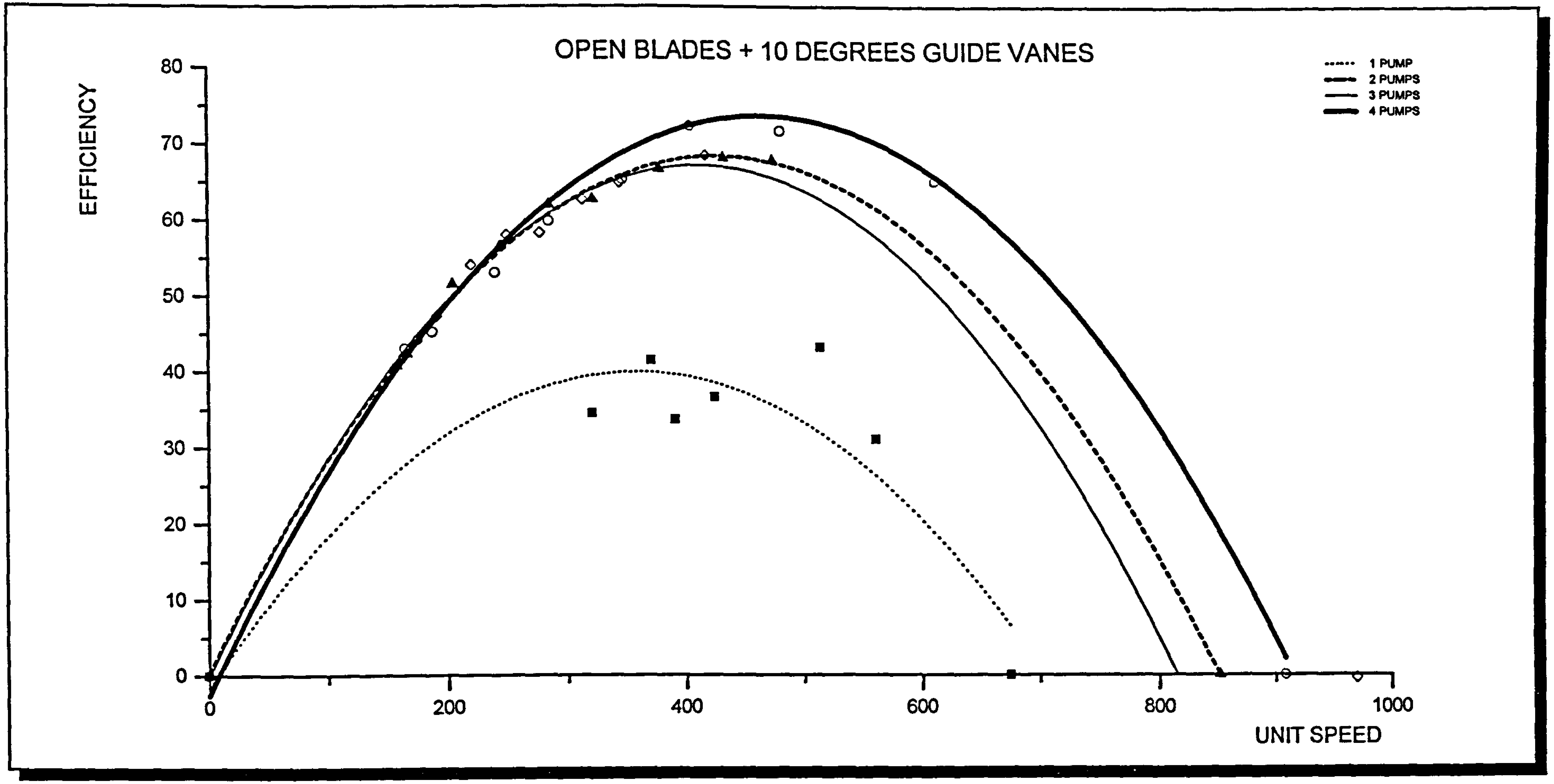
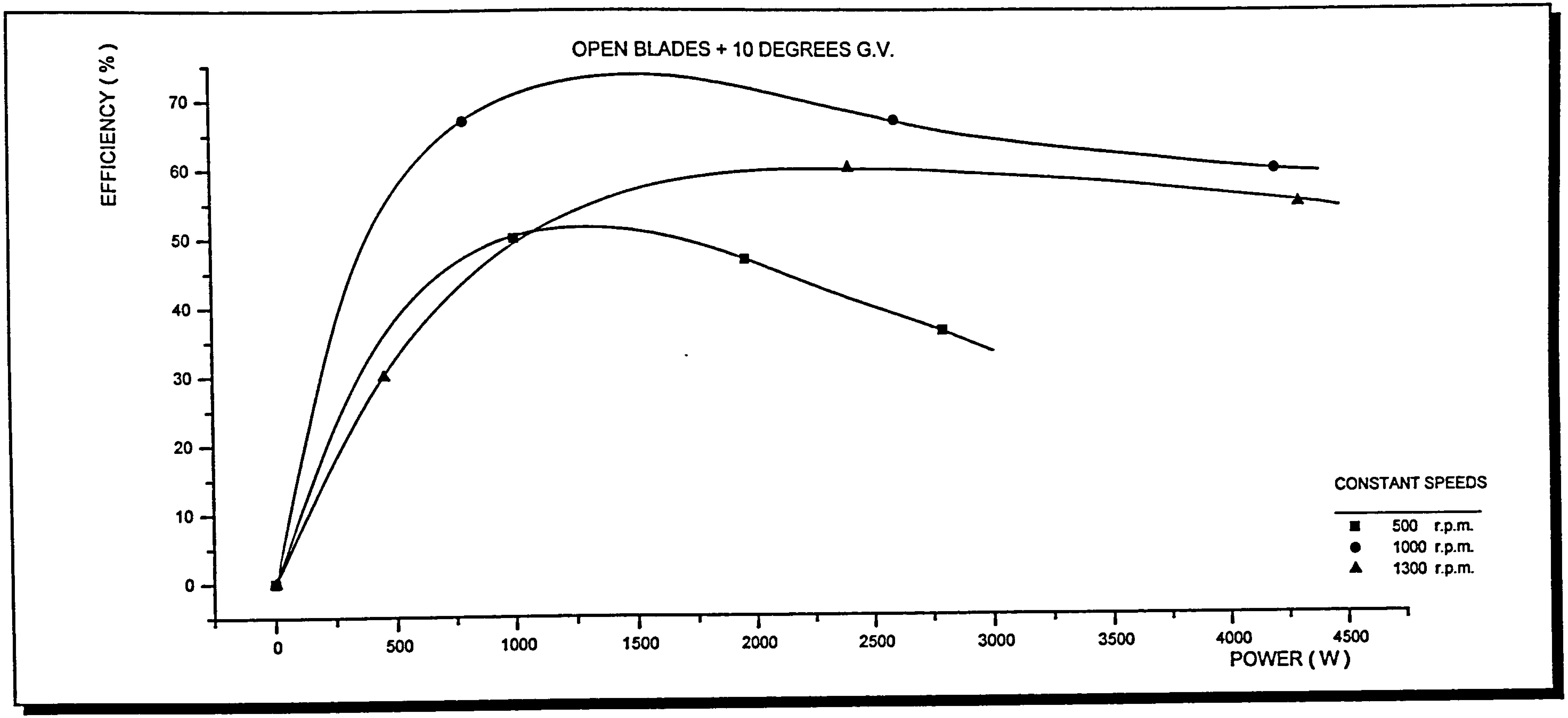

OPEN BLADES + 10 DEGREES GUIDE VANES
UNIT POWER
160
140
120
100
80
60
40
20
0
200
400
600
800
1000
UNIT SPEED
1 PUMP
2 PUMPS
3 PUMPS
4 PUMPS

Figure ( 6.7.4 )

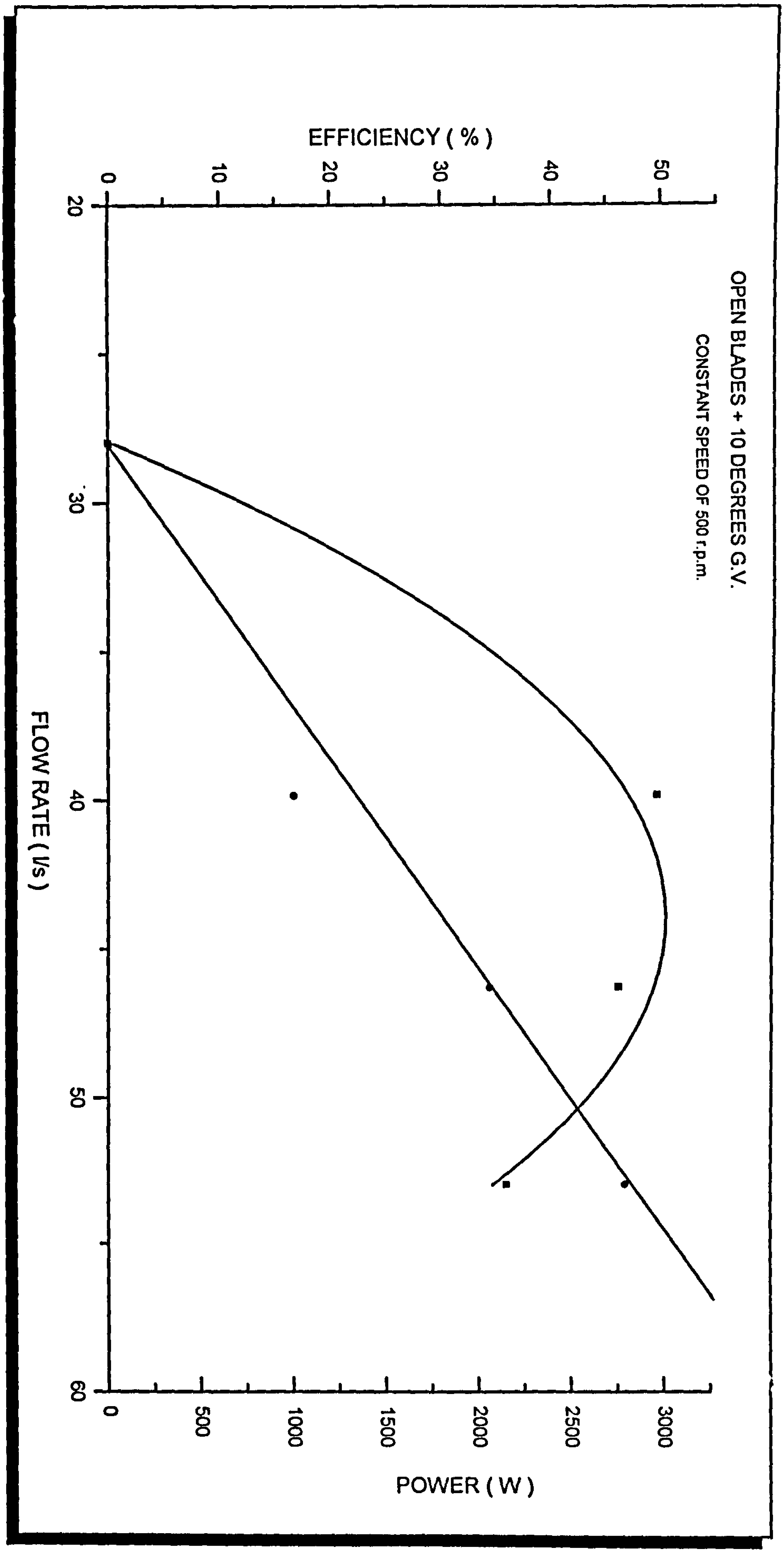

OPEN BLADES + 10 DEGREES GUIDE VANES
EFFICIENCY
80
70
60
50
40
30
20
10
0
0
200
400
600
800
1000
UNIT SPEED
1 PUMP
2 PUMPS
3 PUMPS
4 PUMPS

|Figure ( 6.8.4 )



OPEN BLADES + 10 DEGREES G.V.
EFFICIENCY ( % )
0
10
20
30
40
50
60
70
0
500
1000
1500
2000
2500
3000
3500
4000
4500
POWER ( W )
CONSTANT SPEEDS
500 r.p.m.
1000 r.p.m.
1300 r.p.m.


Figure ( 6.9.4 )



OPEN BLADES + 10 DEGREES G.V.
CONSTANT SPEED OF 500 r.p.m.
EFFICIENCY ( % )
0
10
20
30
40
50
FLOW RATE ( l/s )
20
30
40
50
60
POWER ( W )
0
500
1000
1500
2000
2500
3000


Figure ( 6.10.4 )

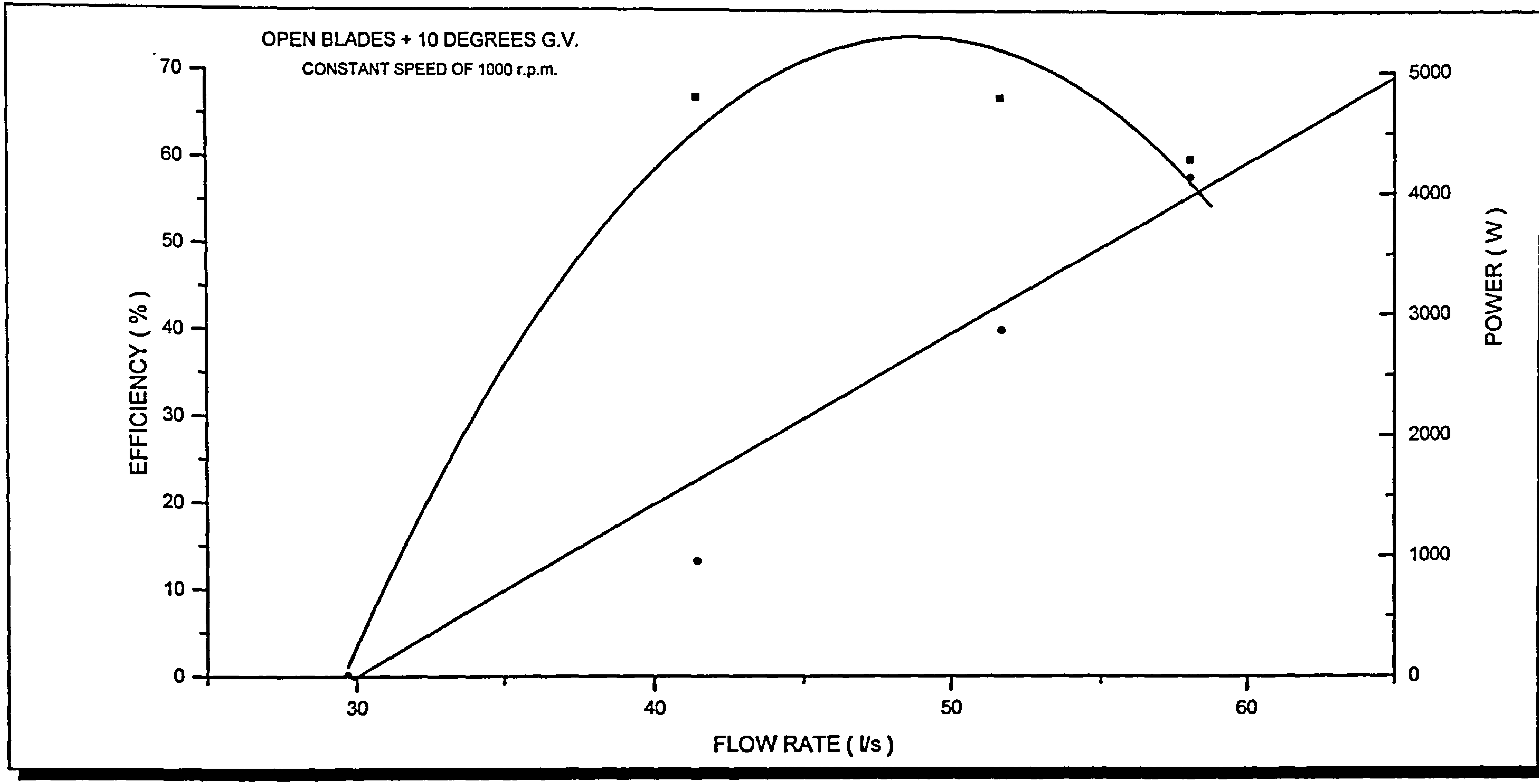

OPEN BLADES + 10 DEGREES G.V.
CONSTANT SPEED OF 1000 r.p.m.
EFFICIENCY ( % )
0
10
20
30
40
50
60
70
FLOW RATE ( l/s )
30
40
50
60
POWER ( W )
0
1000
2000
3000
4000
5000

Figure ( 6.11.4 )

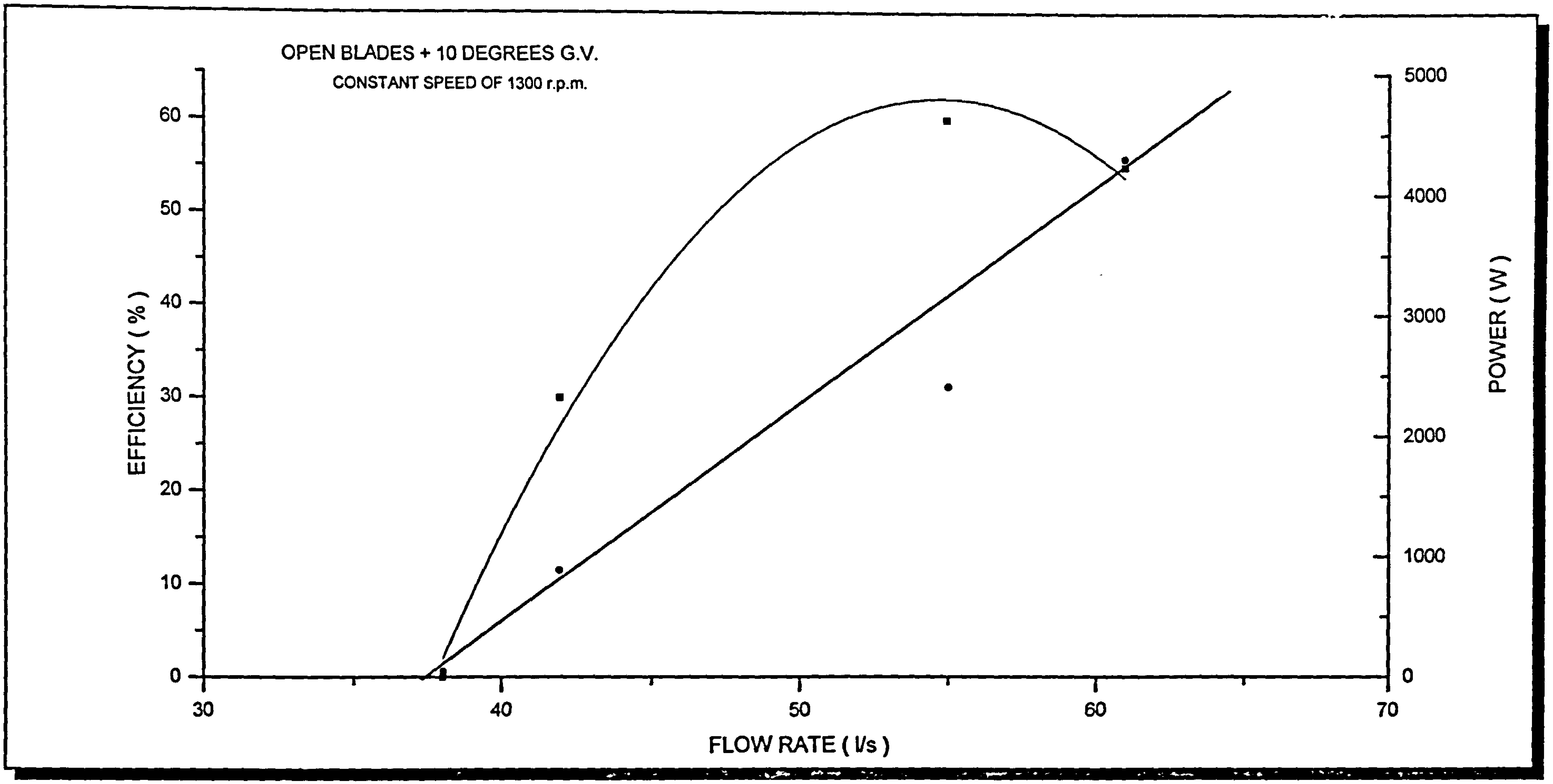

OPEN BLADES + 10 DEGREES G.V.
CONSTANT SPEED OF 1300 r.p.m.
EFFICIENCY ( % )
0
10
20
30
40
50
60
POWER ( W )
0
1000
2000
3000
4000
5000
FLOW RATE ( l/s )
30
40
50
60
70

Figure ( 6.12.4 )

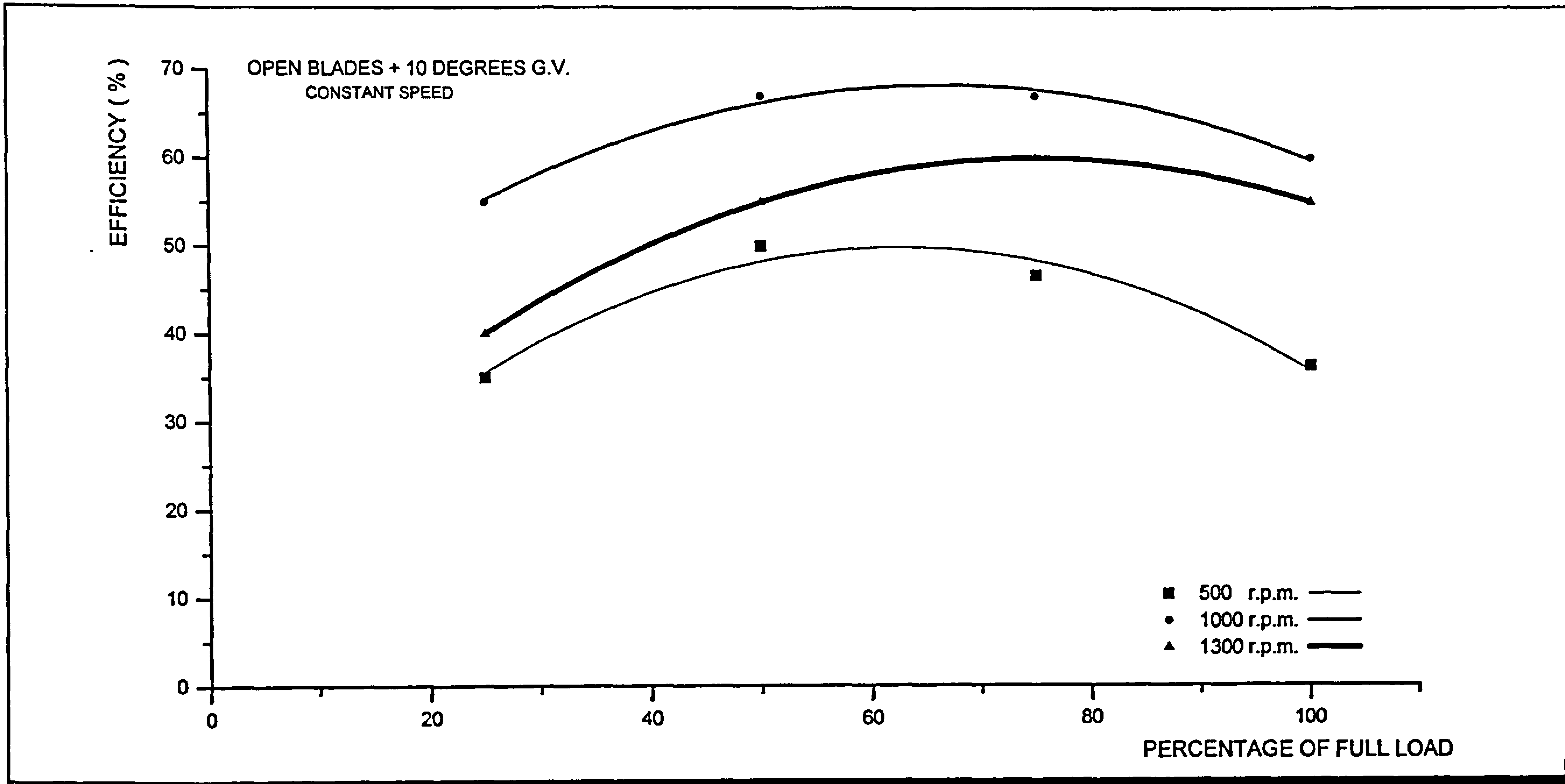

EFFICIENCY ( % )
70
60
50
40
30
20
10
0
OPEN BLADES + 10 DEGREES G.V.
CONSTANT SPEED
500 r.p.m.
1000 r.p.m.
1300 r.p.m.
0
20
40
60
80
100
PERCENTAGE OF FULL LOAD

Figure ( 6.13.4 )

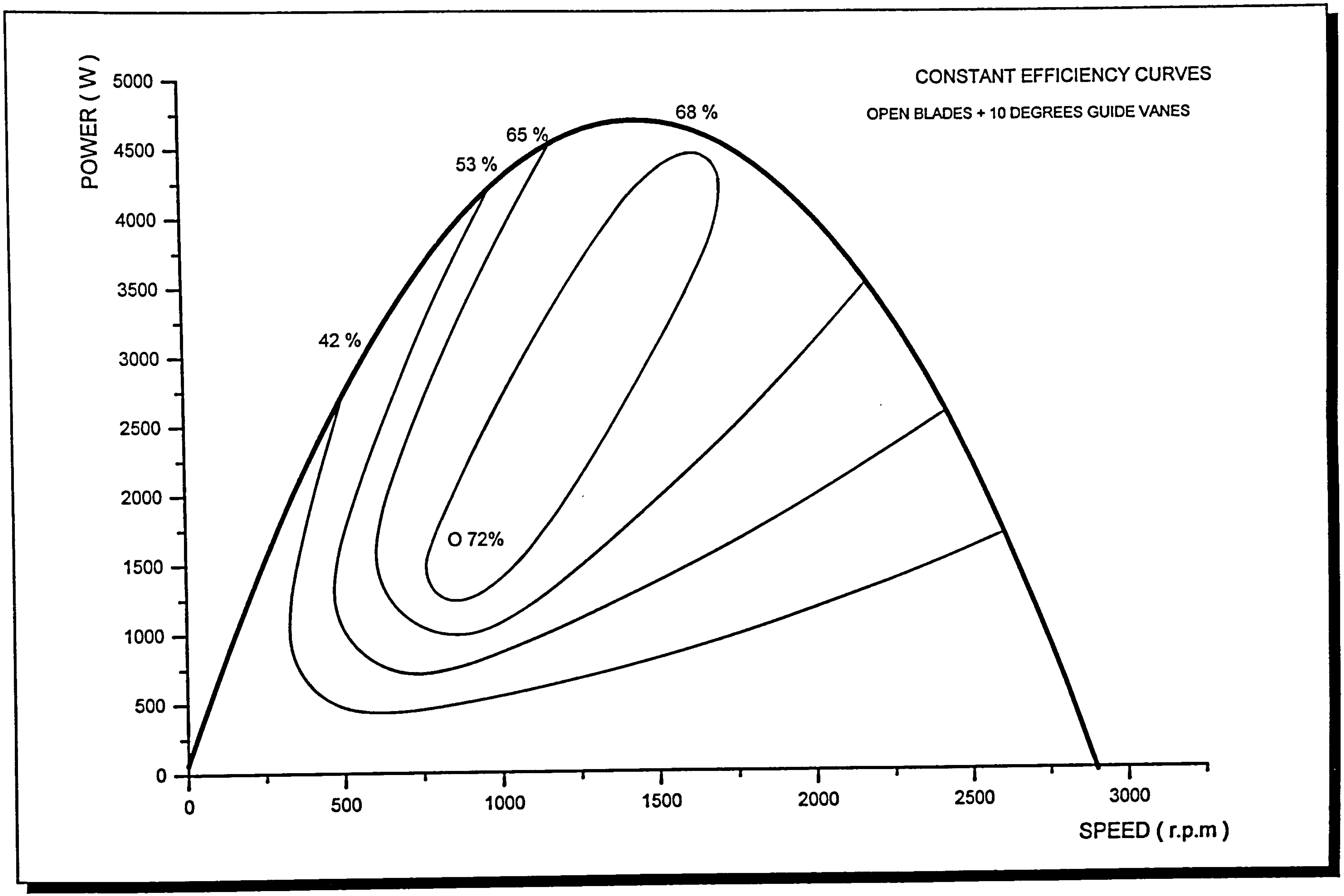

CONSTANT EFFICIENCY CURVES
OPEN BLADES + 10 DEGREES GUIDE VANES
POWER ( W )
5000
4500
4000
3500
3000
2500
2000
1500
1000
500
0
0
500
1000
1500
2000
2500
3000
SPEED ( r.p.m )
42 %
53 %
65 %
68 %
O 72%

Figure ( 6.14.4 )

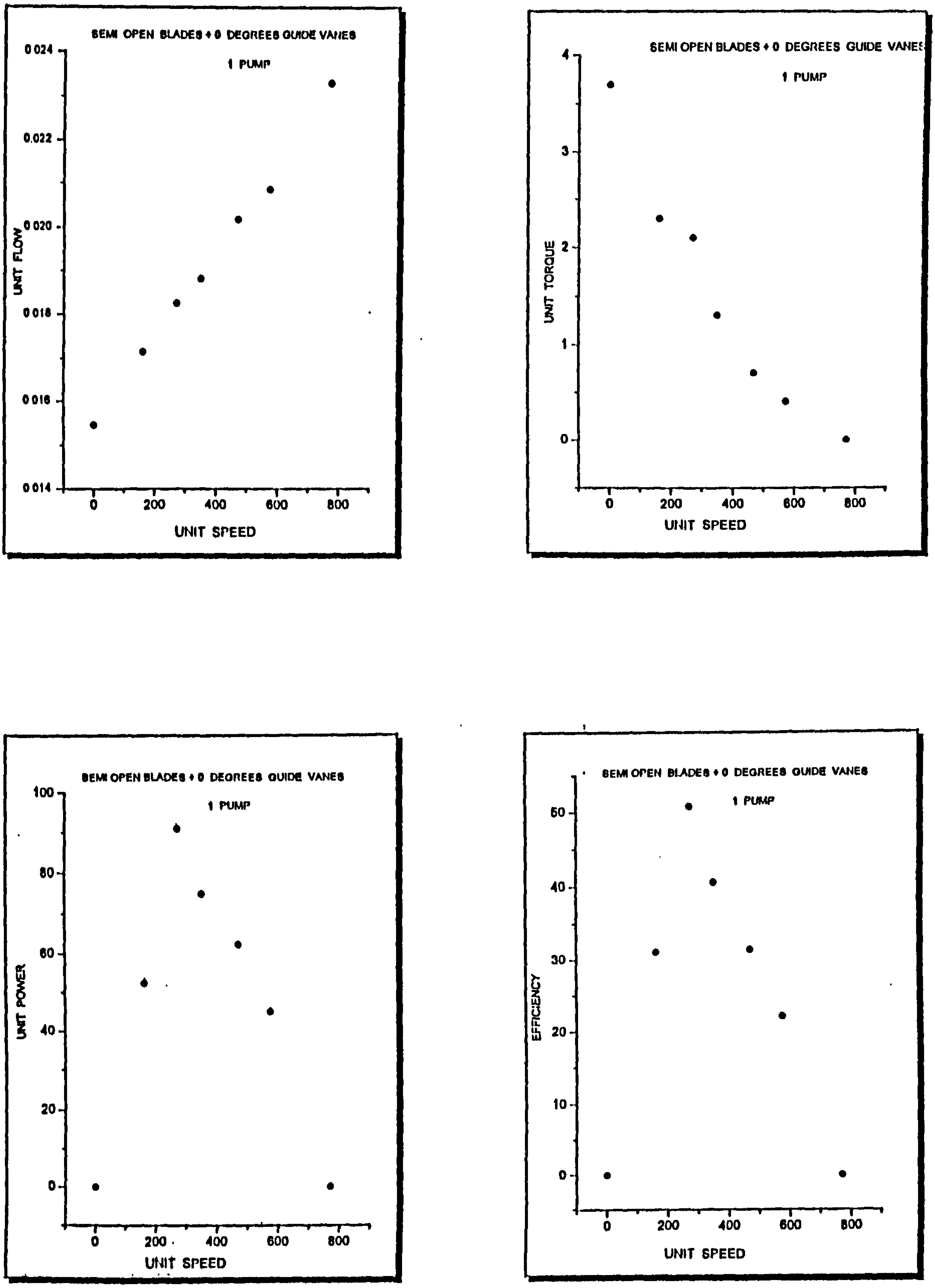

SEMI OPEN BLADES + 0 DEGREES GUIDE VANES
1 PUMP
UNIT FLOW
UNIT SPEED
SEMI OPEN BLADES + 0 DEGREES GUIDE VANES
1 PUMP
UNIT TORQUE
UNIT SPEED
SEMI OPEN BLADES + 0 DEGREES GUIDE VANES
1 PUMP
UNIT POWER
UNIT SPEED
SEMI OPEN BLADES + 0 DEGREES GUIDE VANES
1 PUMP
EFFICIENCY
UNIT SPEED

Figure ( 6.1.5 )

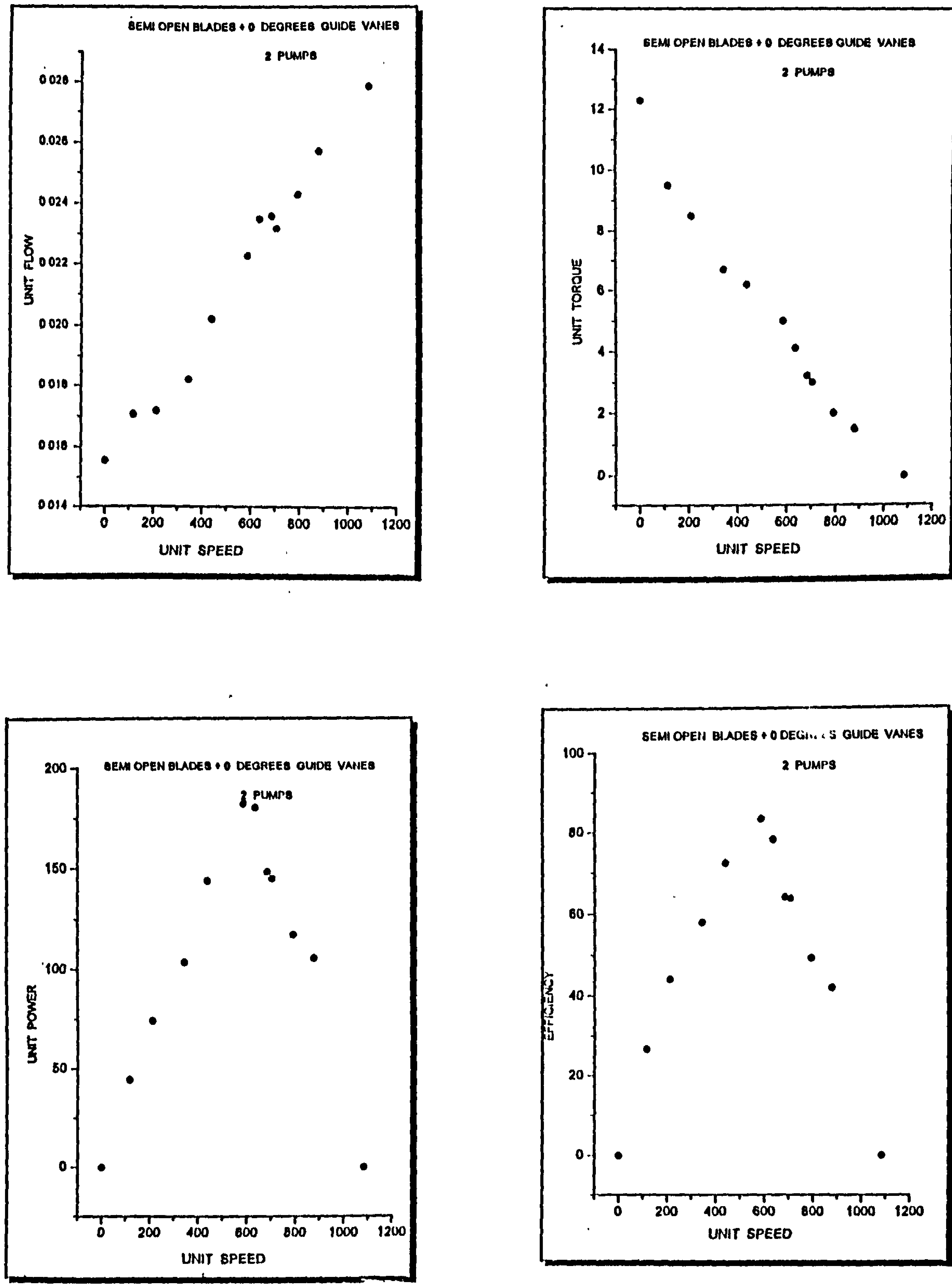

SEMI OPEN BLADES + 0 DEGREES GUIDE VANES
2 PUMPS
UNIT FLOW
UNIT SPEED
SEMI OPEN BLADES + 0 DEGREES GUIDE VANES
2 PUMPS
UNIT TORQUE
UNIT SPEED
SEMI OPEN BLADES + 0 DEGREES GUIDE VANES
2 PUMPS
UNIT POWER
UNIT SPEED
2 PUMPS
UNIT SPEED

Figure ( 6.2.5 )

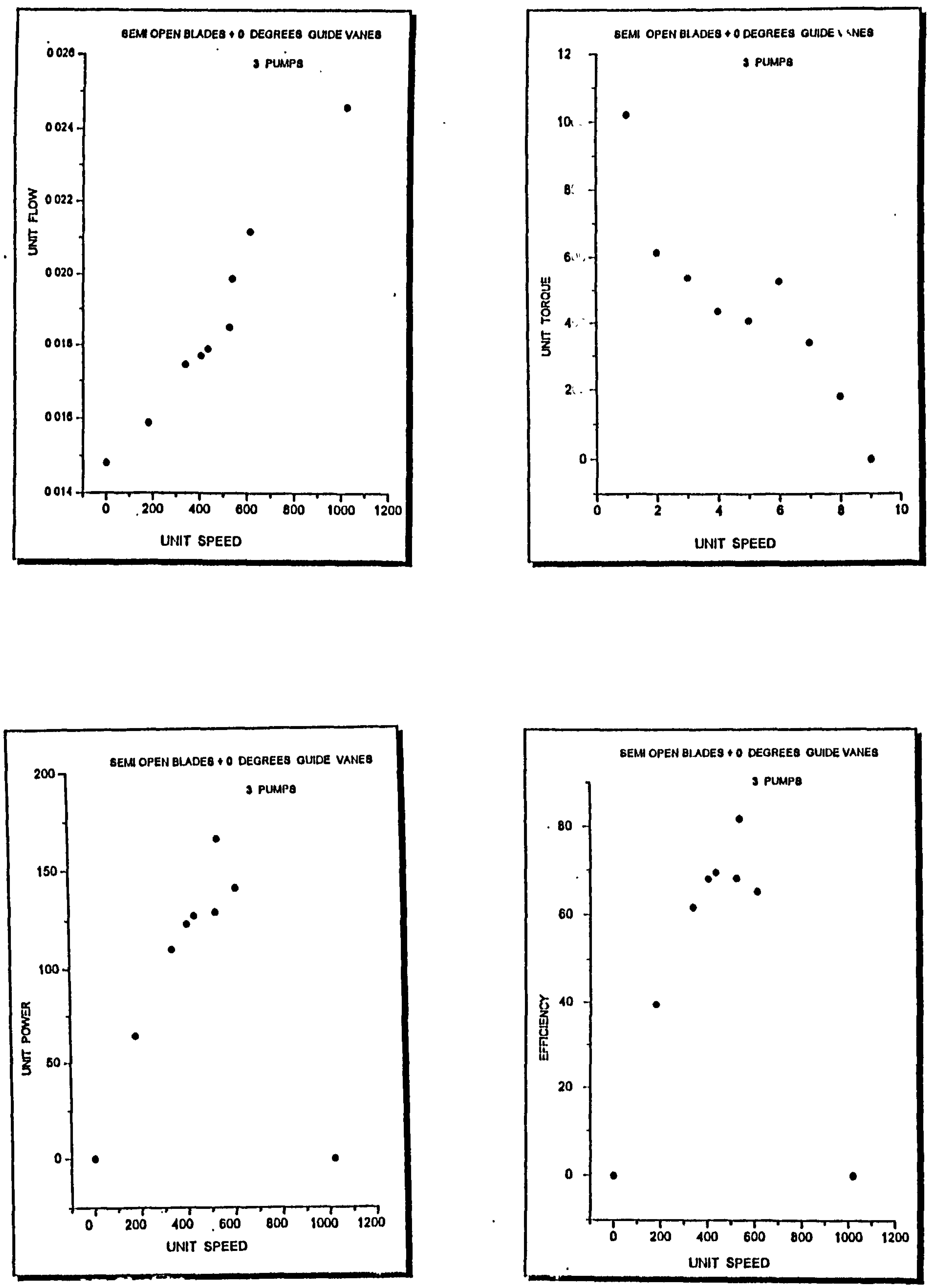

SEMI OPEN BLADES + 0 DEGREES GUIDE VANES
3 PUMPS
UNIT FLOW
UNIT SPEED
UNIT TORQUE
UNIT POWER
EFFICIENCY

Figure ( 6.3.5 )

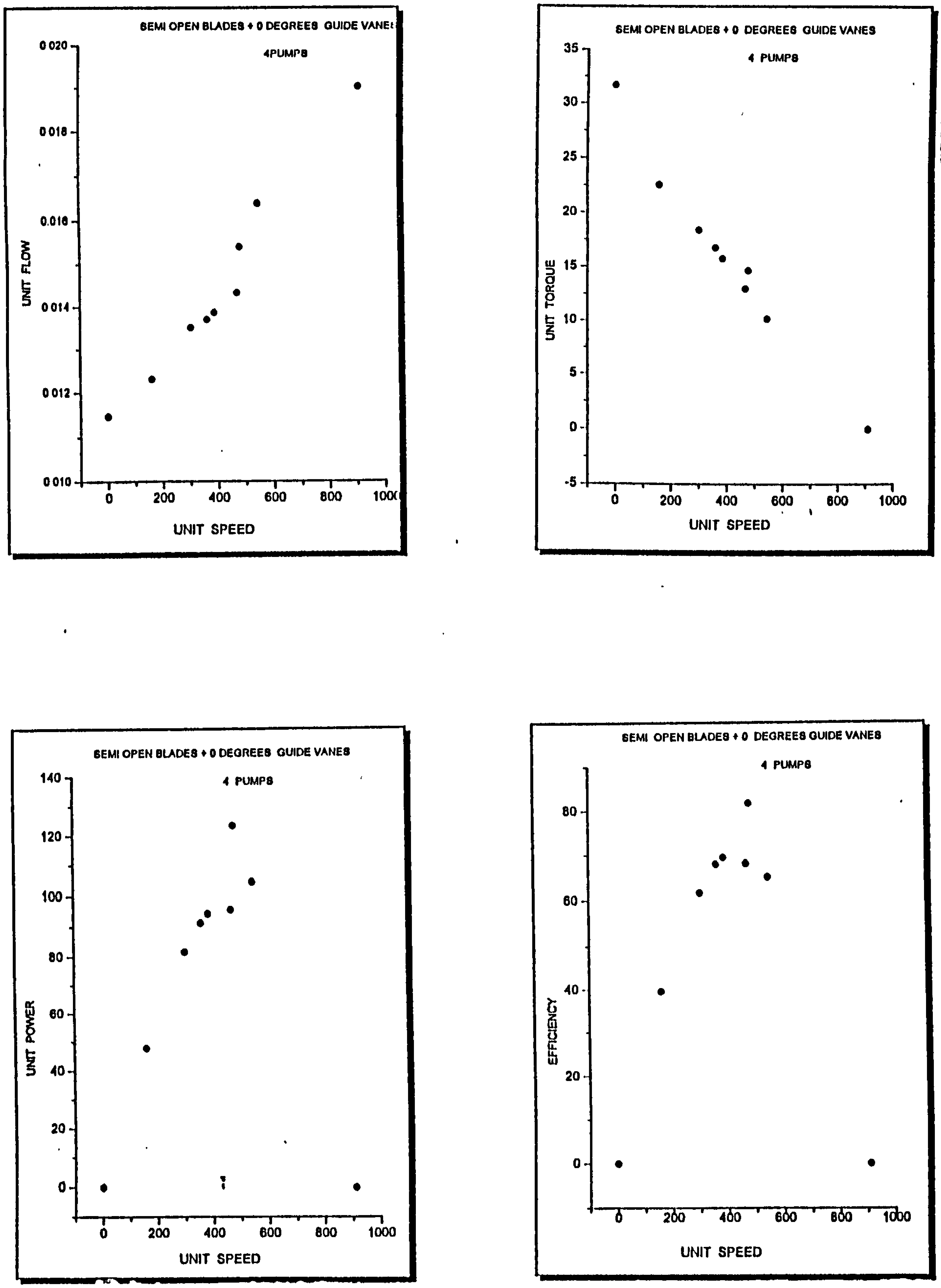

SEMI OPEN BLADES + 0 DEGREES GUIDE VANES
4PUMPS
UNIT FLOW
UNIT SPEED
SEMI OPEN BLADES + 0 DEGREES GUIDE VANES
4 PUMPS
UNIT TORQUE
UNIT SPEED
SEMI OPEN BLADES + 0 DEGREES GUIDE VANES
4 PUMPS
UNIT POWER
UNIT SPEED
SEMI OPEN BLADES + 0 DEGREES GUIDE VANES
4 PUMPS
EFFICIENCY
UNIT SPEED

Figure ( 6.4.5 )

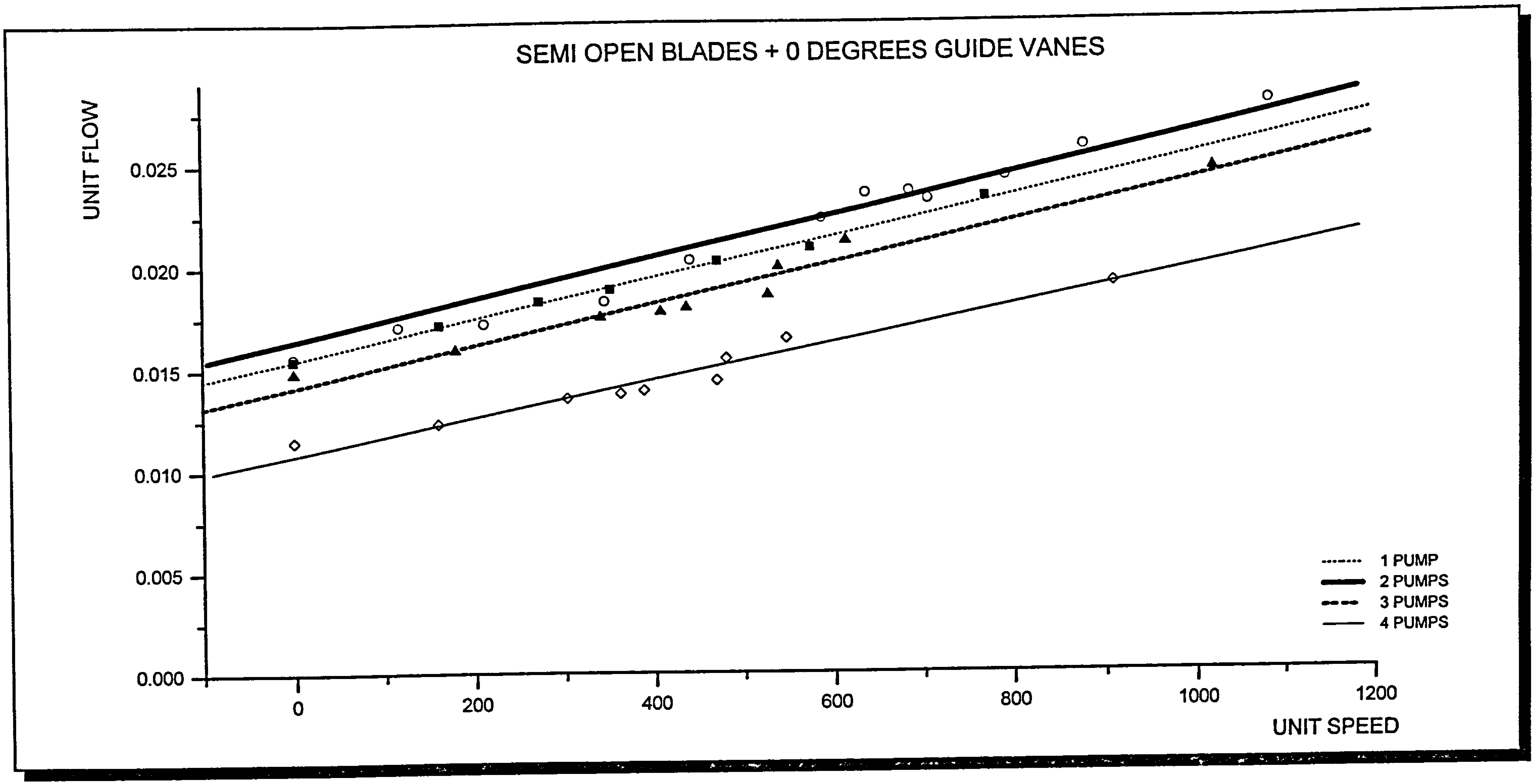

SEMI OPEN BLADES + 0 DEGREES GUIDE VANES
UNIT FLOW
0.000
0.005
0.010
0.015
0.020
0.025
0
200
400
600
800
1000
1200
UNIT SPEED
1 PUMP
2 PUMPS
3 PUMPS
4 PUMPS

Figure ( 6.5.5 )

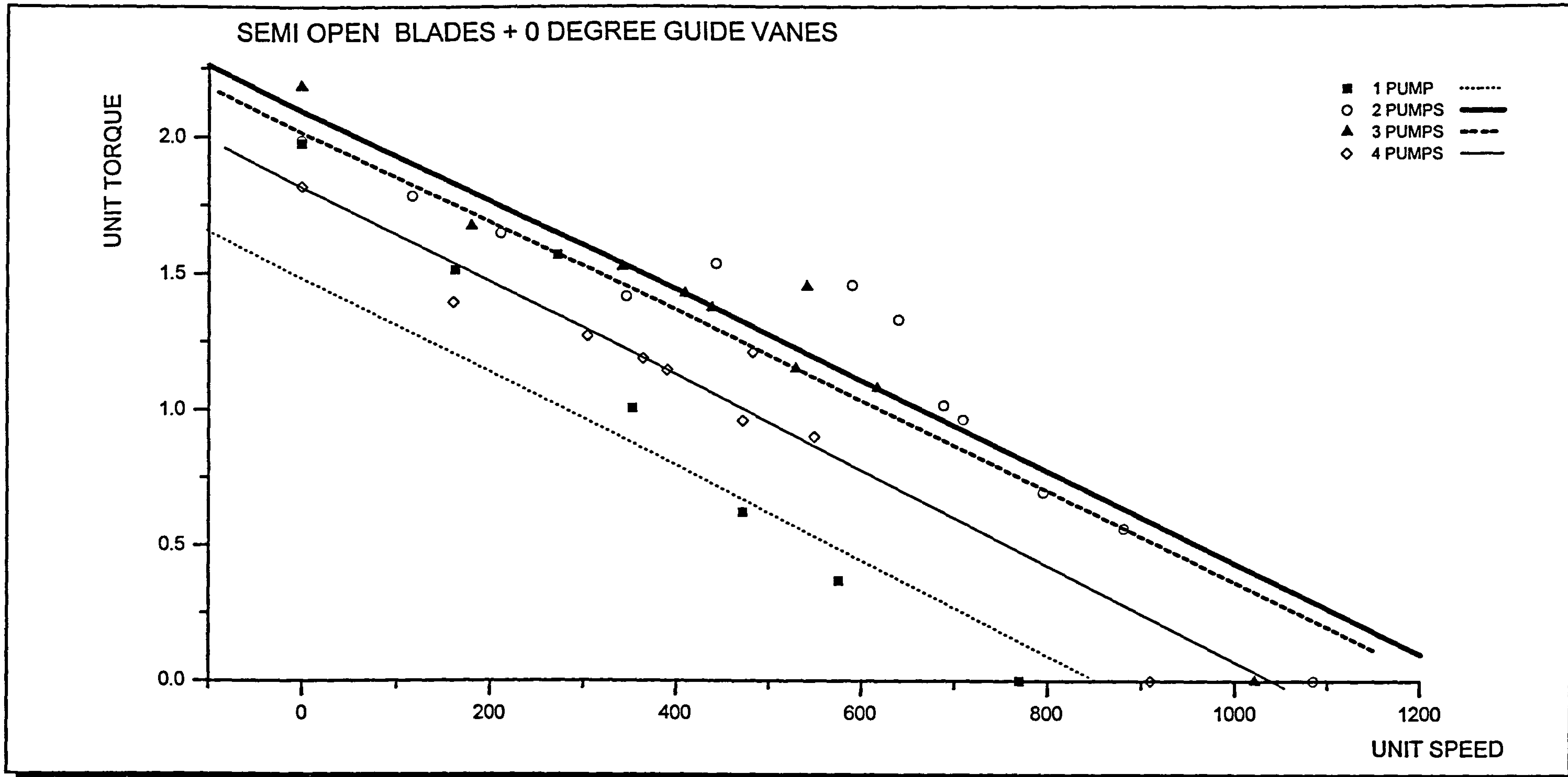
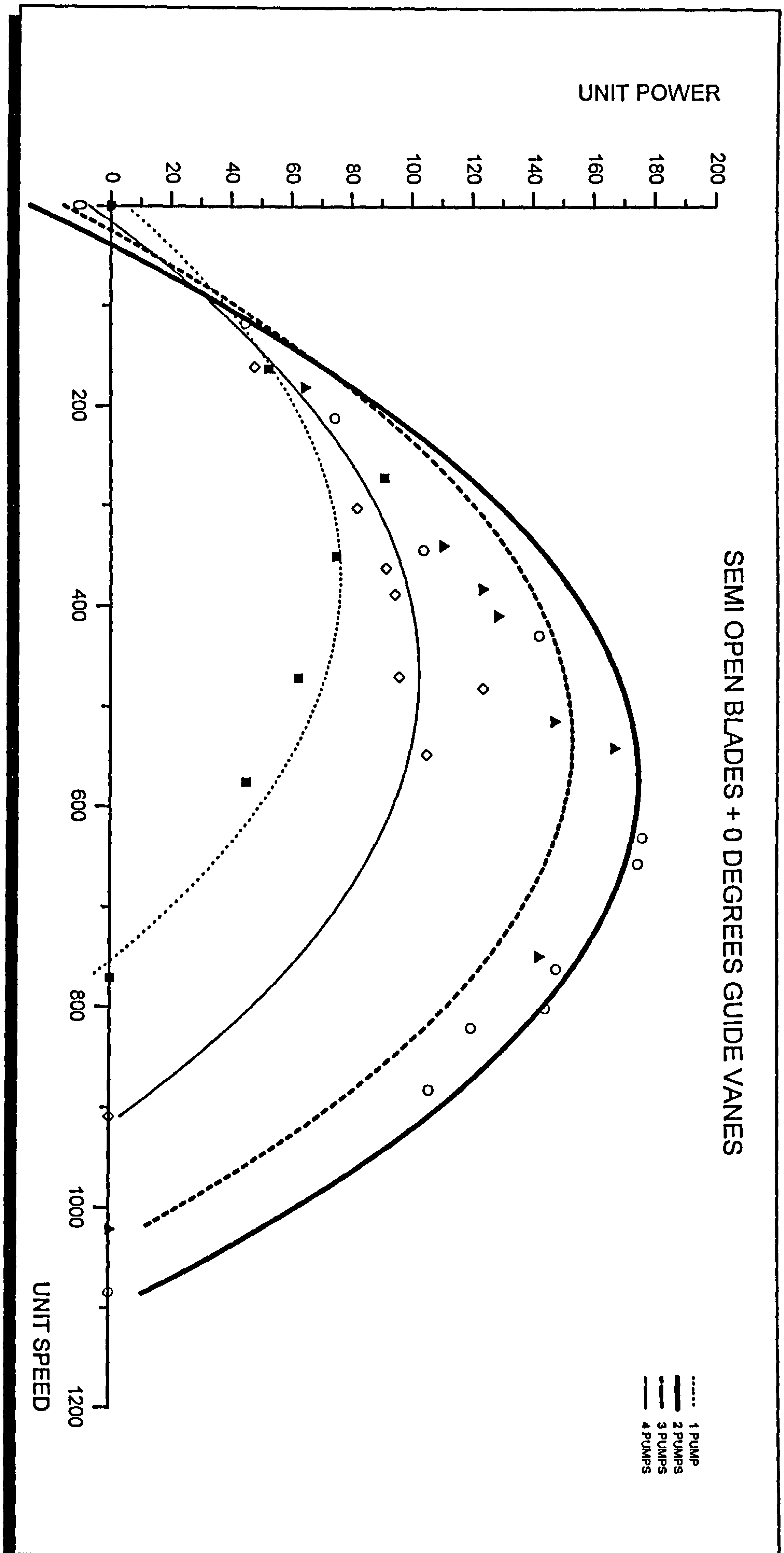
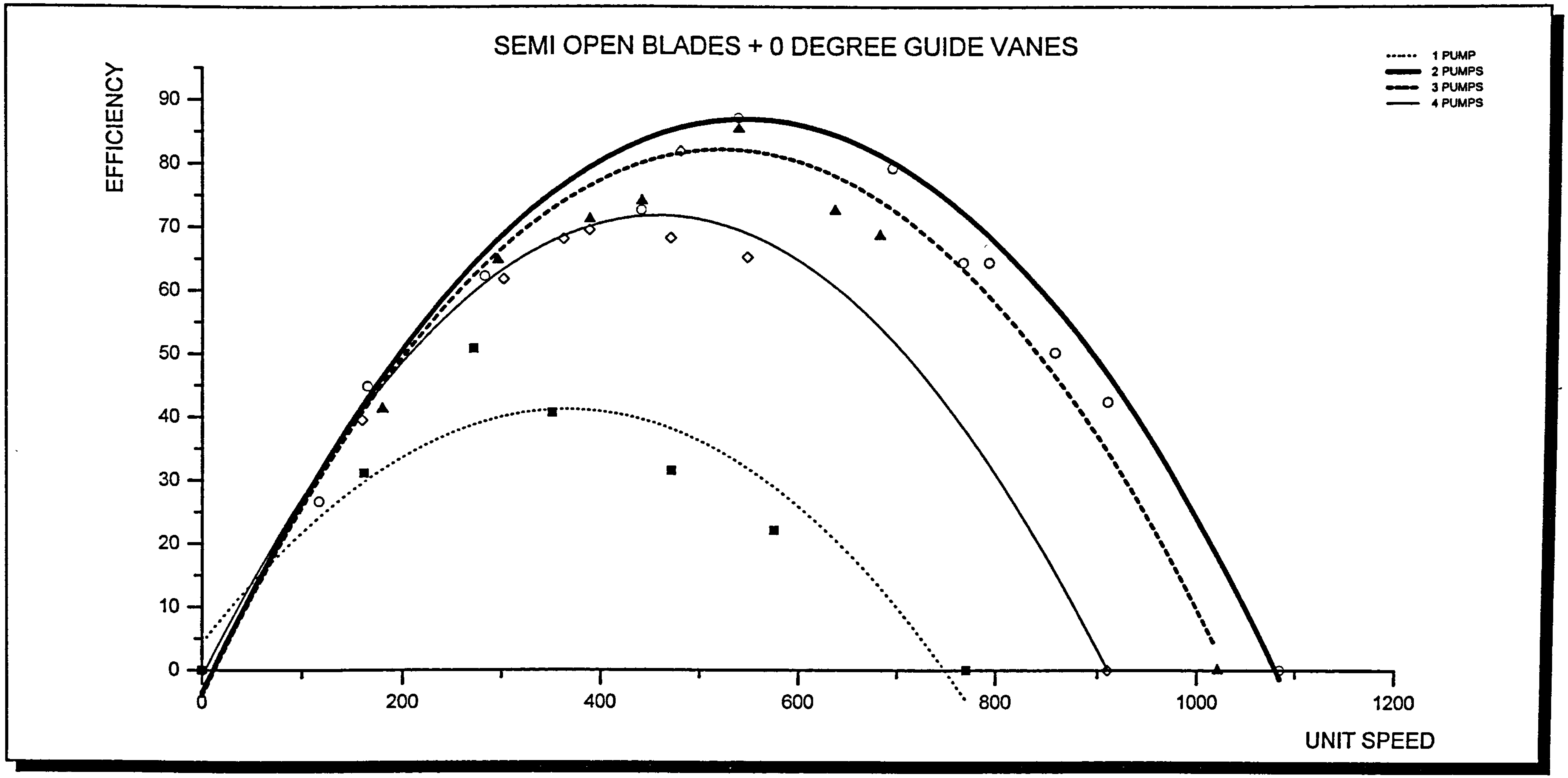

SEMI OPEN BLADES + 0 DEGREE GUIDE VANES
UNIT TORQUE
2.0
1.5
1.0
0.5
0.0
0
200
400
600
800
1000
1200
UNIT SPEED
1 PUMP
2 PUMPS
3 PUMPS
4 PUMPS

Figure ( 6.6.5 )



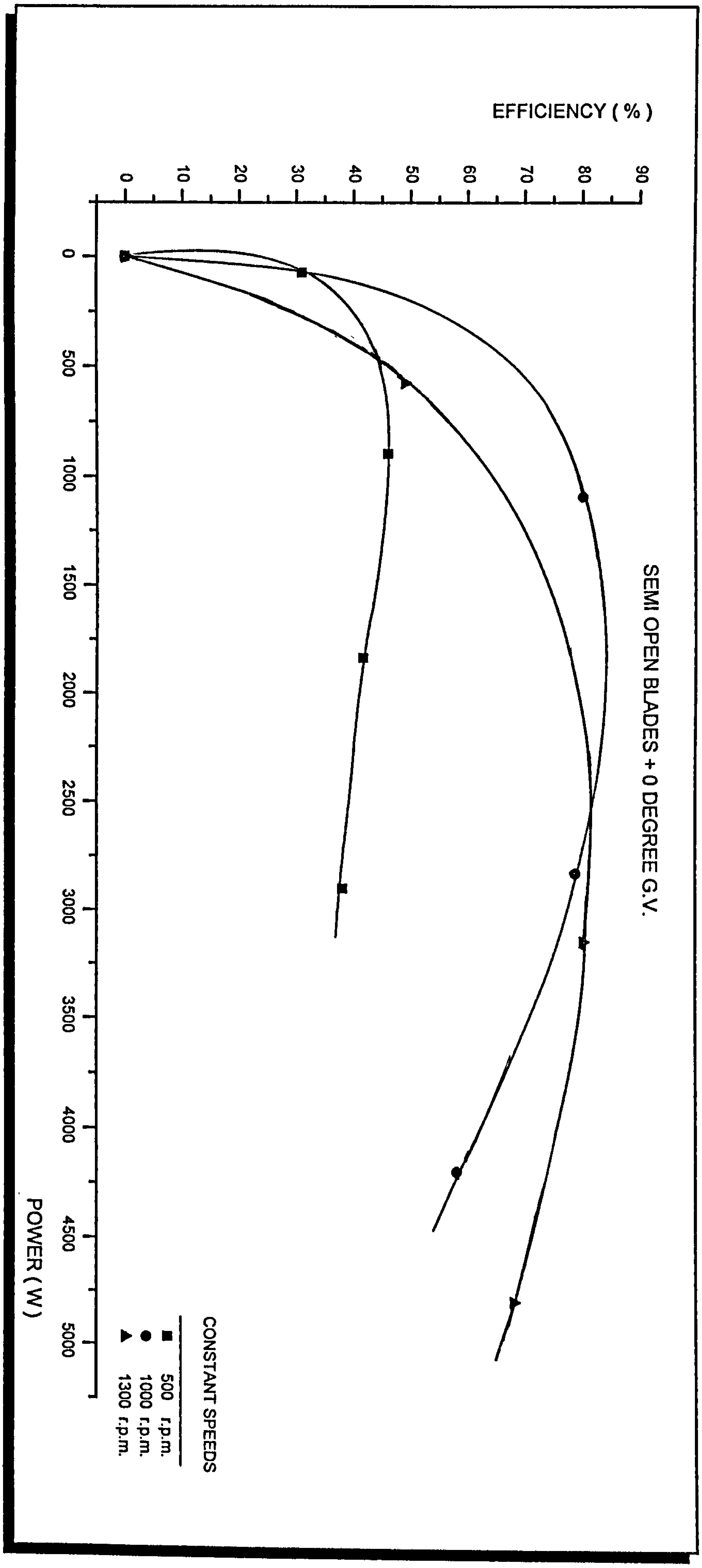
SEMI OPEN BLADES + 0 DEGREES GUIDE VANES
UNIT POWER
0
20
40
60
80
100
120
140
160
180
200
UNIT SPEED
0
200
400
600
800
1000
1200
1 PUMP
2 PUMPS
3 PUMPS
4 PUMPS


Figure ( 6.7.5 )



SEMI OPEN BLADES + 0 DEGREE GUIDE VANES
EFFICIENCY
0
10
20
30
40
50
60
70
80
90
0
200
400
600
800
1000
1200
UNIT SPEED
1 PUMP
2 PUMPS
3 PUMPS
4 PUMPS


Figure ( 6.8.5 )



EFFICIENCY ( % )
0
10
20
30
40
50
60
70
80
90
SEMI OPEN BLADES + 0 DEGREE G.V.
0
500
1000
1500
2000
2500
3000
3500
4000
4500
5000
POWER ( W )
CONSTANT SPEEDS
500 r.p.m.
1000 r.p.m.
1300 r.p.m.


Figure ( 6.9.5 )

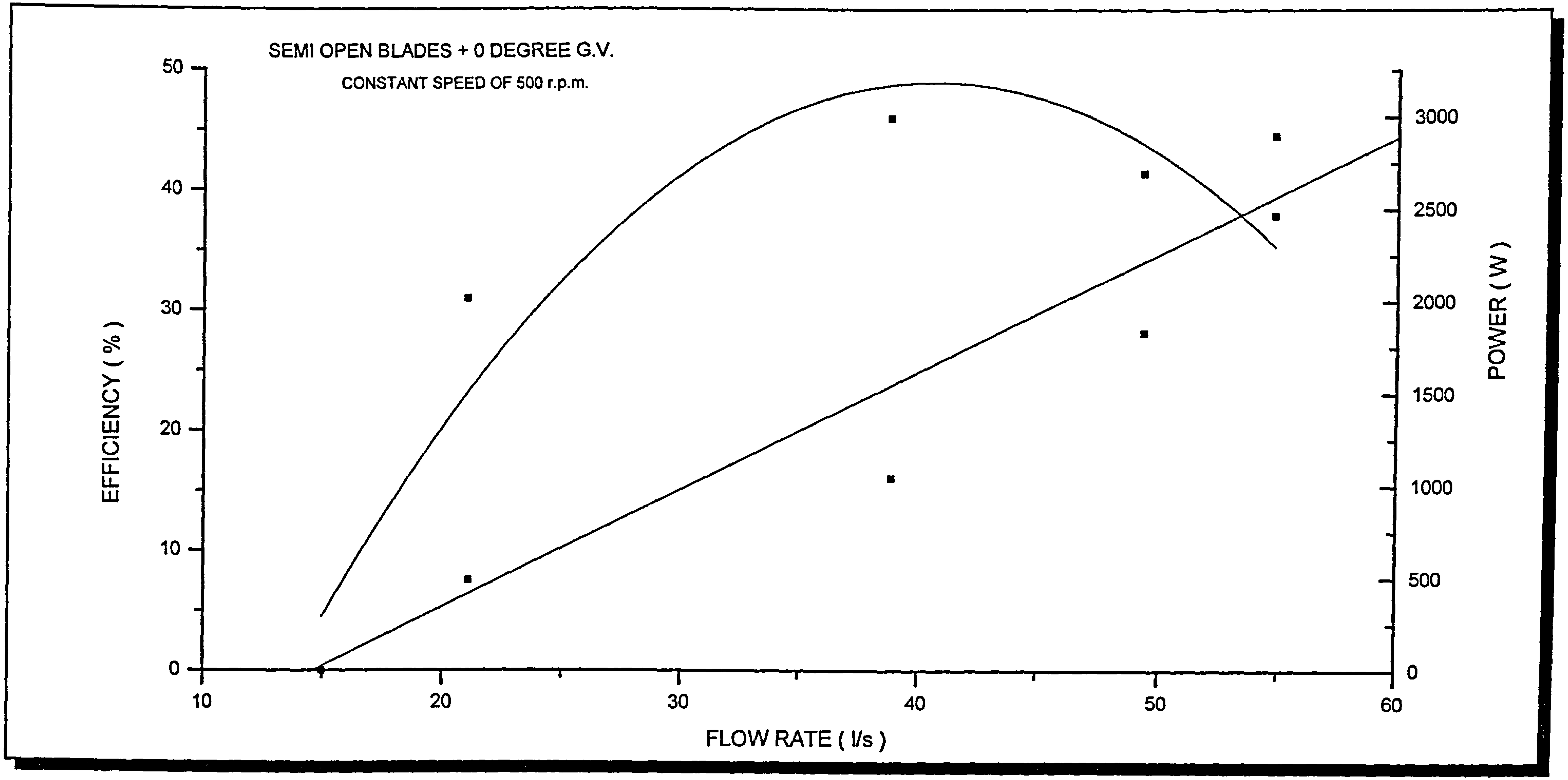

SEMI OPEN BLADES + 0 DEGREE G.V.
CONSTANT SPEED OF 500 r.p.m.
EFFICIENCY ( % )
0
10
20
30
40
50
POWER ( W )
0
500
1000
1500
2000
2500
3000
FLOW RATE ( l/s )
10
20
30
40
50
60

Figure ( 6.10.5 )

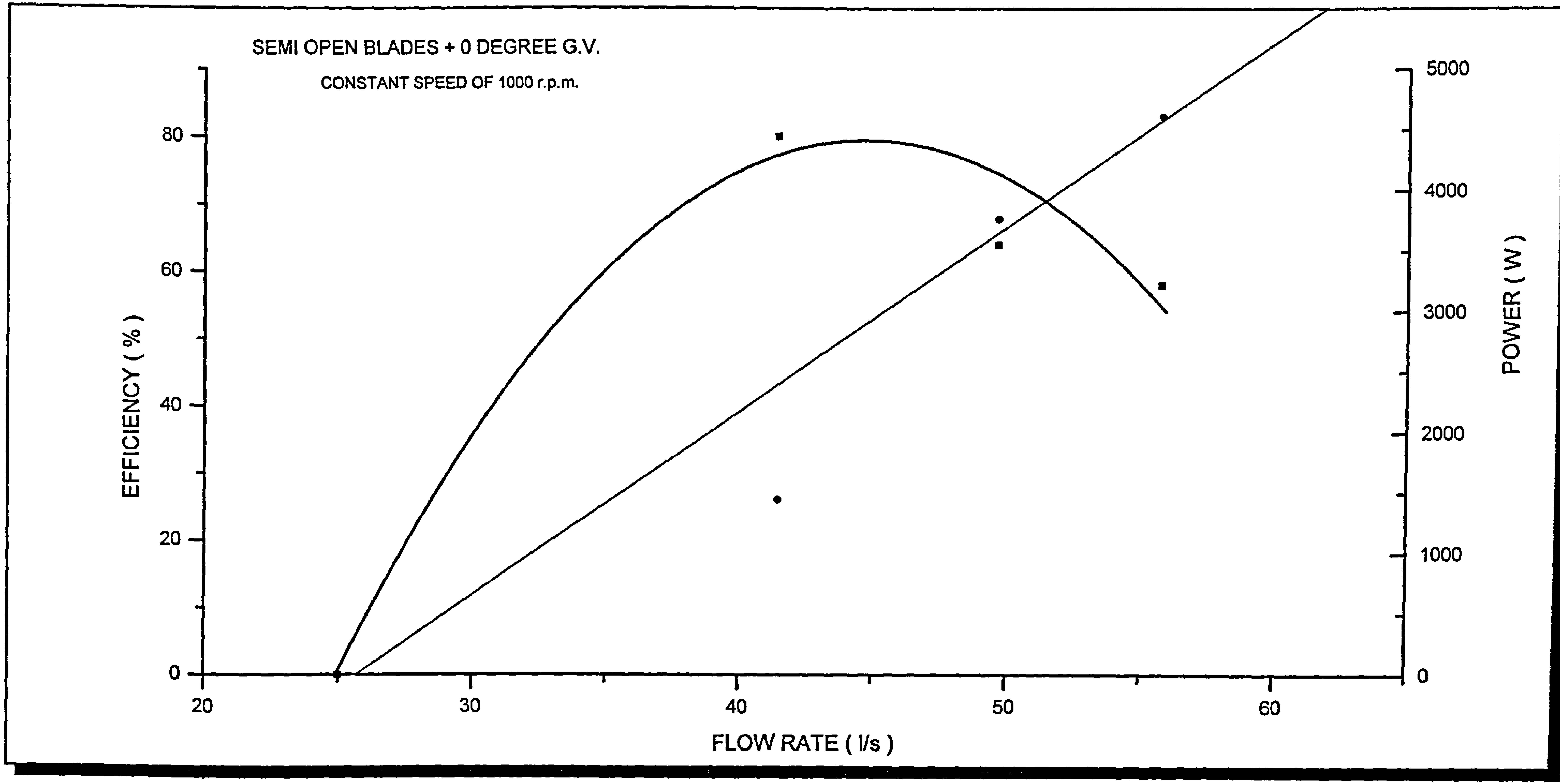

SEMI OPEN BLADES + 0 DEGREE G.V.
CONSTANT SPEED OF 1000 r.p.m.
EFFICIENCY ( % )
0
20
40
60
80
POWER ( W )
0
1000
2000
3000
4000
5000
FLOW RATE ( l/s )
20
30
40
50
60

Figure ( 6.11.5 )

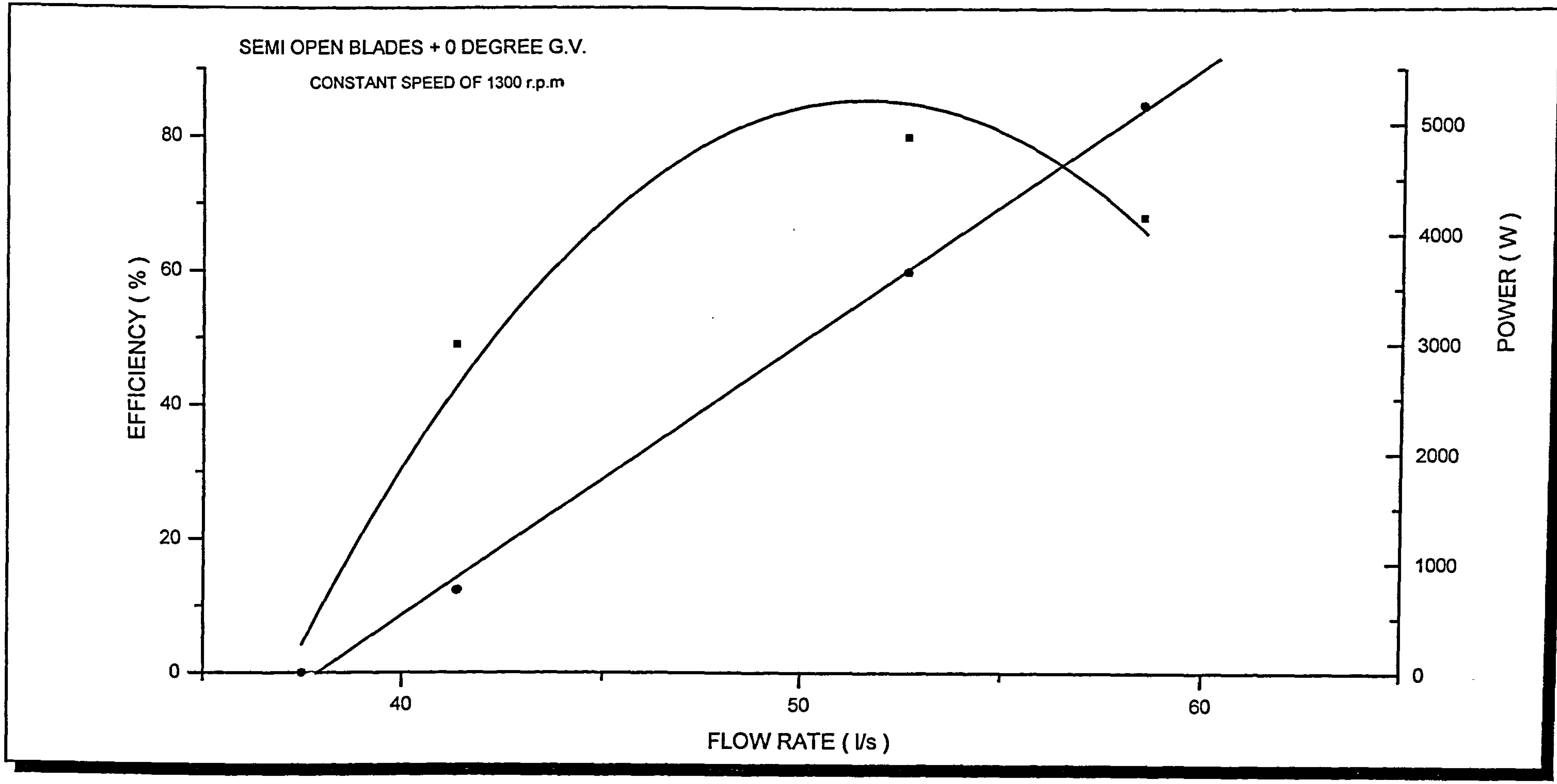

SEMI OPEN BLADES + 0 DEGREE G.V.
CONSTANT SPEED OF 1300 r.p.m
EFFICIENCY ( % )
0
20
40
60
80
FLOW RATE ( l/s )
40
50
60
POWER ( W )
0
1000
2000
3000
4000
5000

Figure ( 6.12.5 )

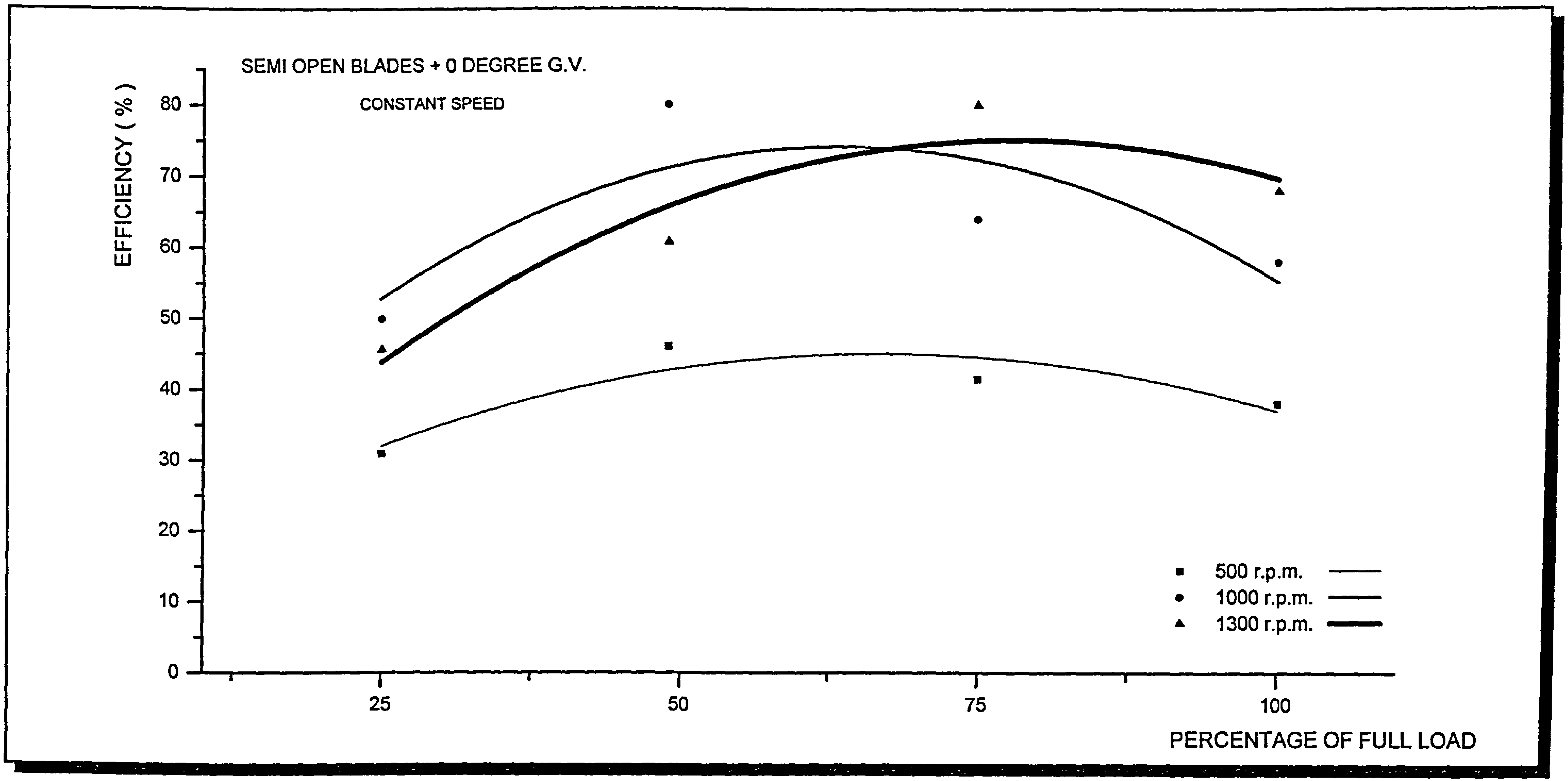

SEMI OPEN BLADES + 0 DEGREE G.V.
CONSTANT SPEED
EFFICIENCY ( % )
0
10
20
30
40
50
60
70
80
25
50
75
100
PERCENTAGE OF FULL LOAD
500 r.p.m.
1000 r.p.m.
1300 r.p.m.

Figure ( 6.13.5 )

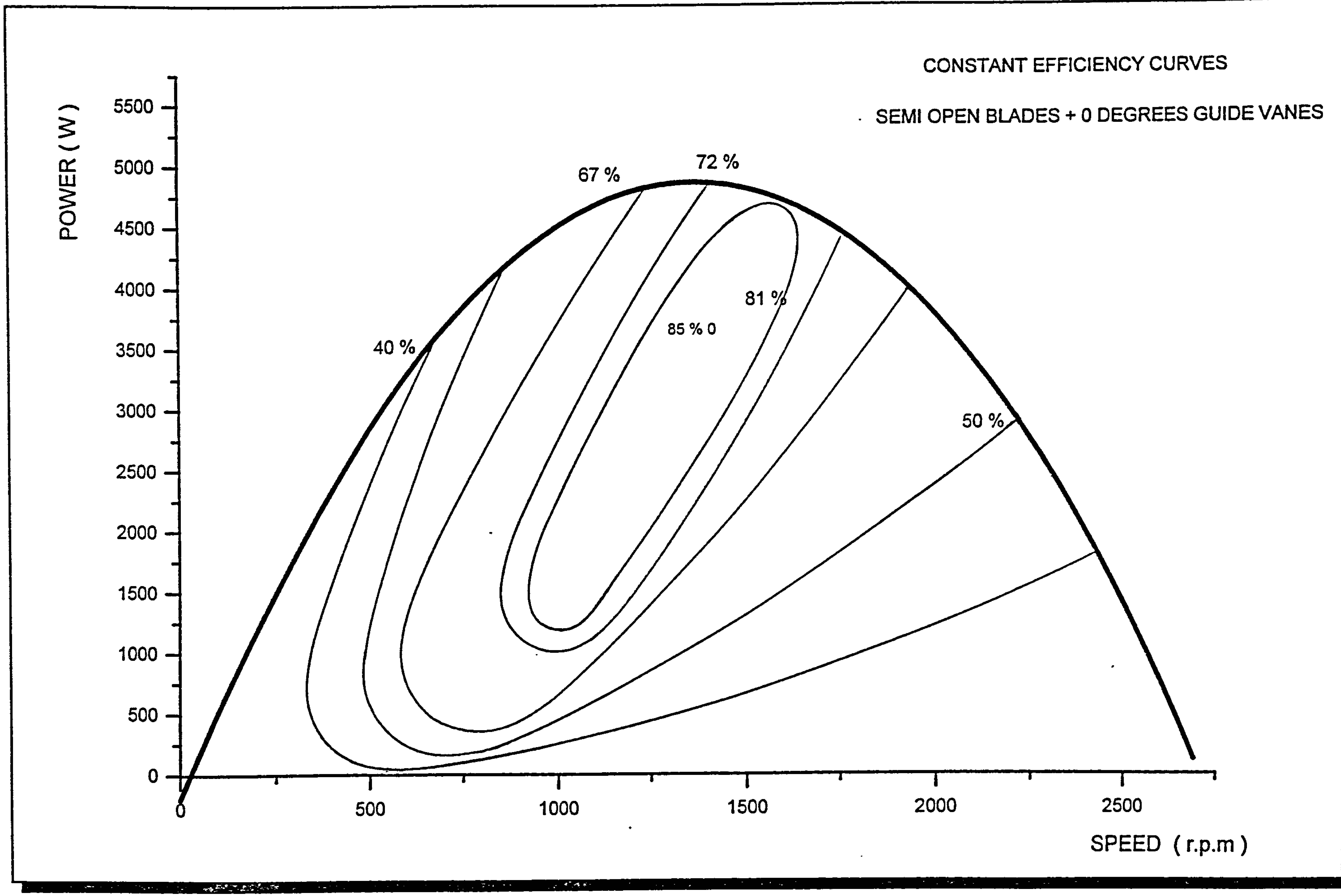

CONSTANT EFFICIENCY CURVES
SEMI OPEN BLADES + 0 DEGREES GUIDE VANES
POWER ( W )
0
500
1000
1500
2000
2500
3000
3500
4000
4500
5000
5500
SPEED ( r.p.m )
0
500
1000
1500
2000
2500
40 %
67 %
72 %
81 %
85 % 0
50 %

Figure ( 6.14.5 )

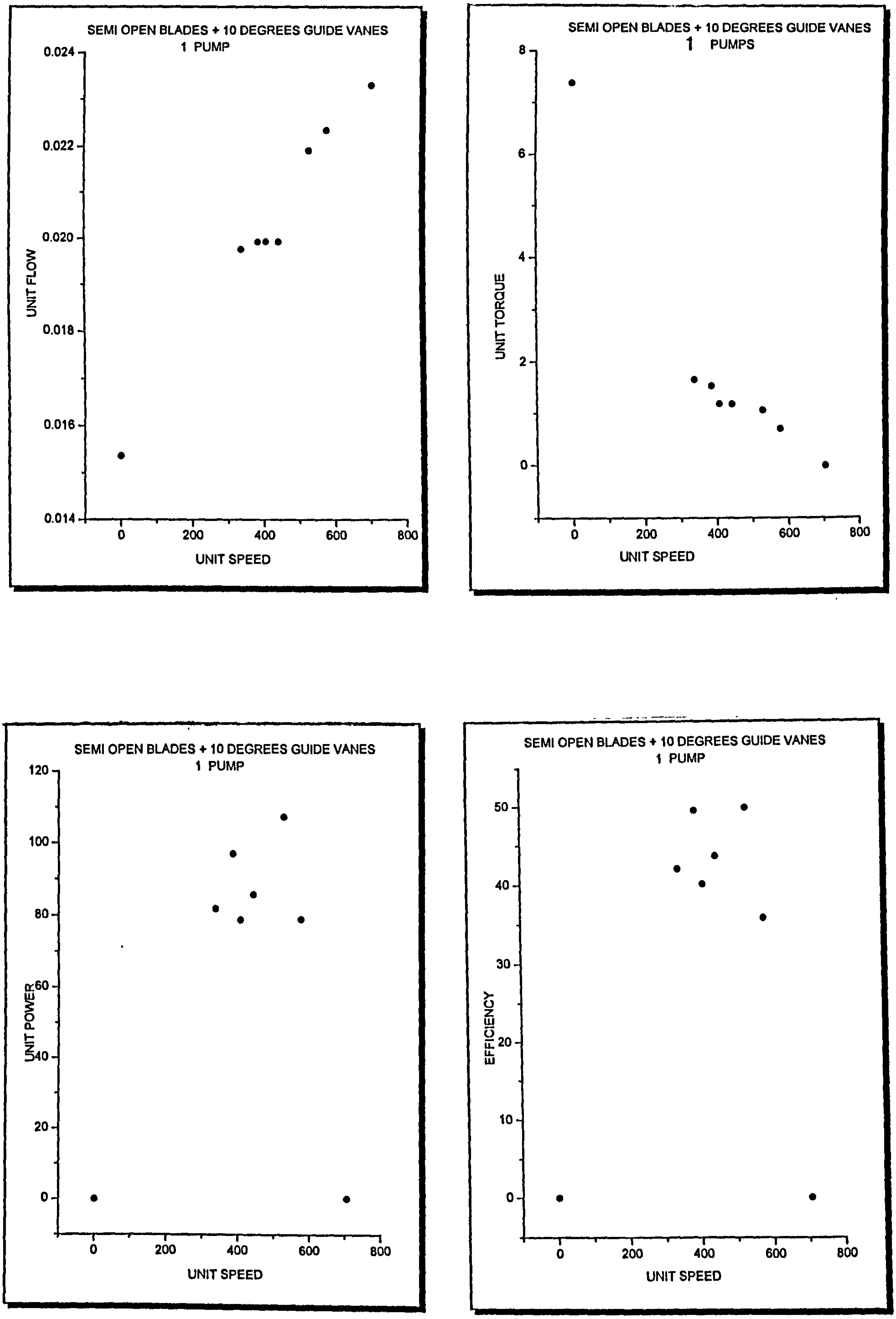

SEMI OPEN BLADES + 10 DEGREES GUIDE VANES
1 PUMP
UNIT FLOW
UNIT SPEED
SEMI OPEN BLADES + 10 DEGREES GUIDE VANES
1 PUMPS
UNIT TORQUE
UNIT SPEED
SEMI OPEN BLADES + 10 DEGREES GUIDE VANES
1 PUMP
UNIT POWER
UNIT SPEED
SEMI OPEN BLADES + 10 DEGREES GUIDE VANES
1 PUMP
EFFICIENCY
UNIT SPEED

Figure ( 6.1.6 )

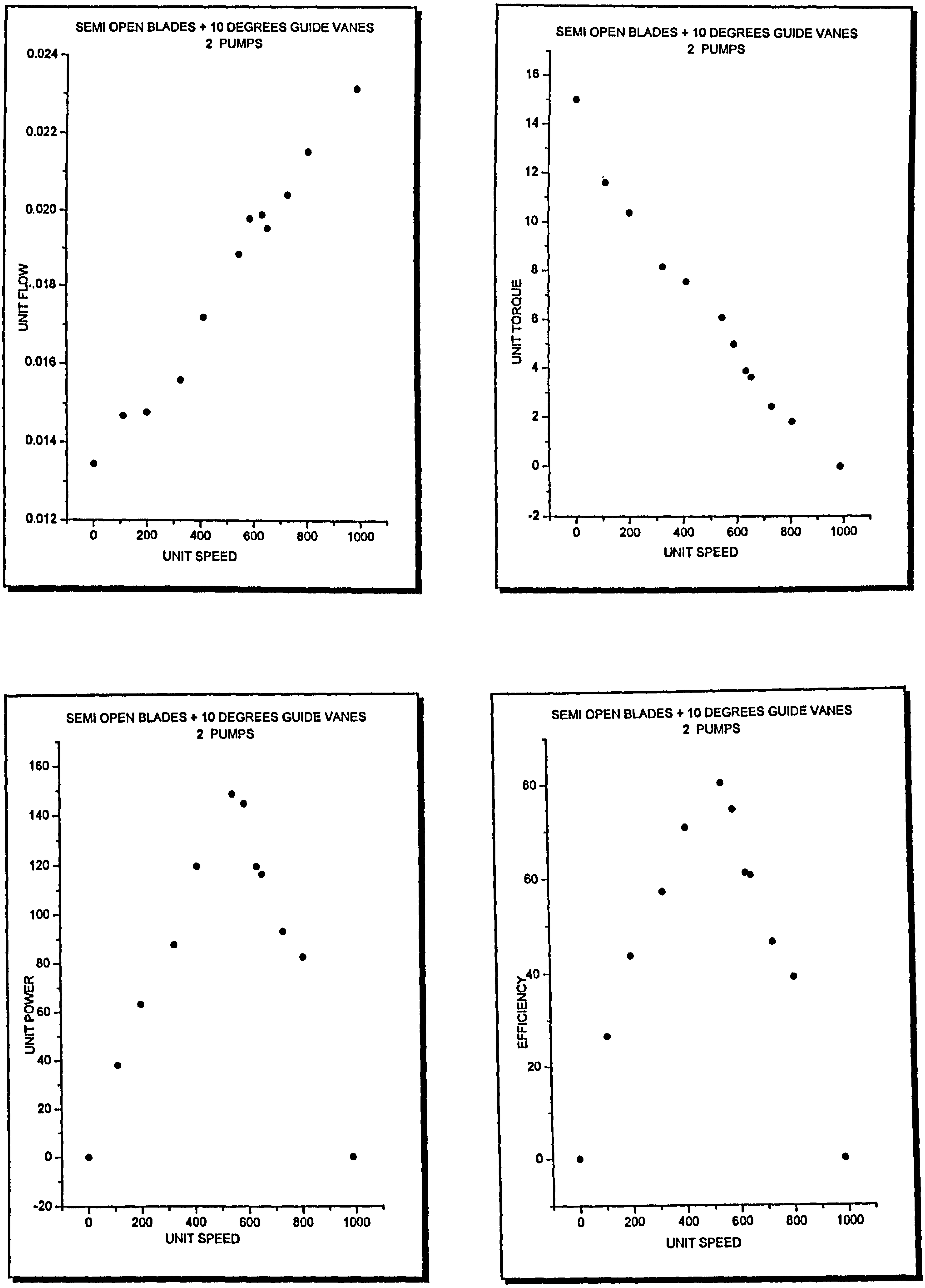

SEMI OPEN BLADES + 10 DEGREES GUIDE VANES
2 PUMPS
UNIT FLOW
0.024
0.022
0.020
0.018
0.016
0.014
0.012
0
200
400
600
800
1000
UNIT SPEED
SEMI OPEN BLADES + 10 DEGREES GUIDE VANES
2 PUMPS
UNIT TORQUE
16
14
12
10
8
6
4
2
0
-2
0
200
400
600
800
1000
UNIT SPEED
SEMI OPEN BLADES + 10 DEGREES GUIDE VANES
2 PUMPS
UNIT POWER
160
140
120
100
80
60
40
20
0
-20
0
200
400
600
800
1000
UNIT SPEED
SEMI OPEN BLADES + 10 DEGREES GUIDE VANES
2 PUMPS
EFFICIENCY
80
60
40
20
0
0
200
400
600
800
1000
UNIT SPEED

Figure ( 6.2.6 )

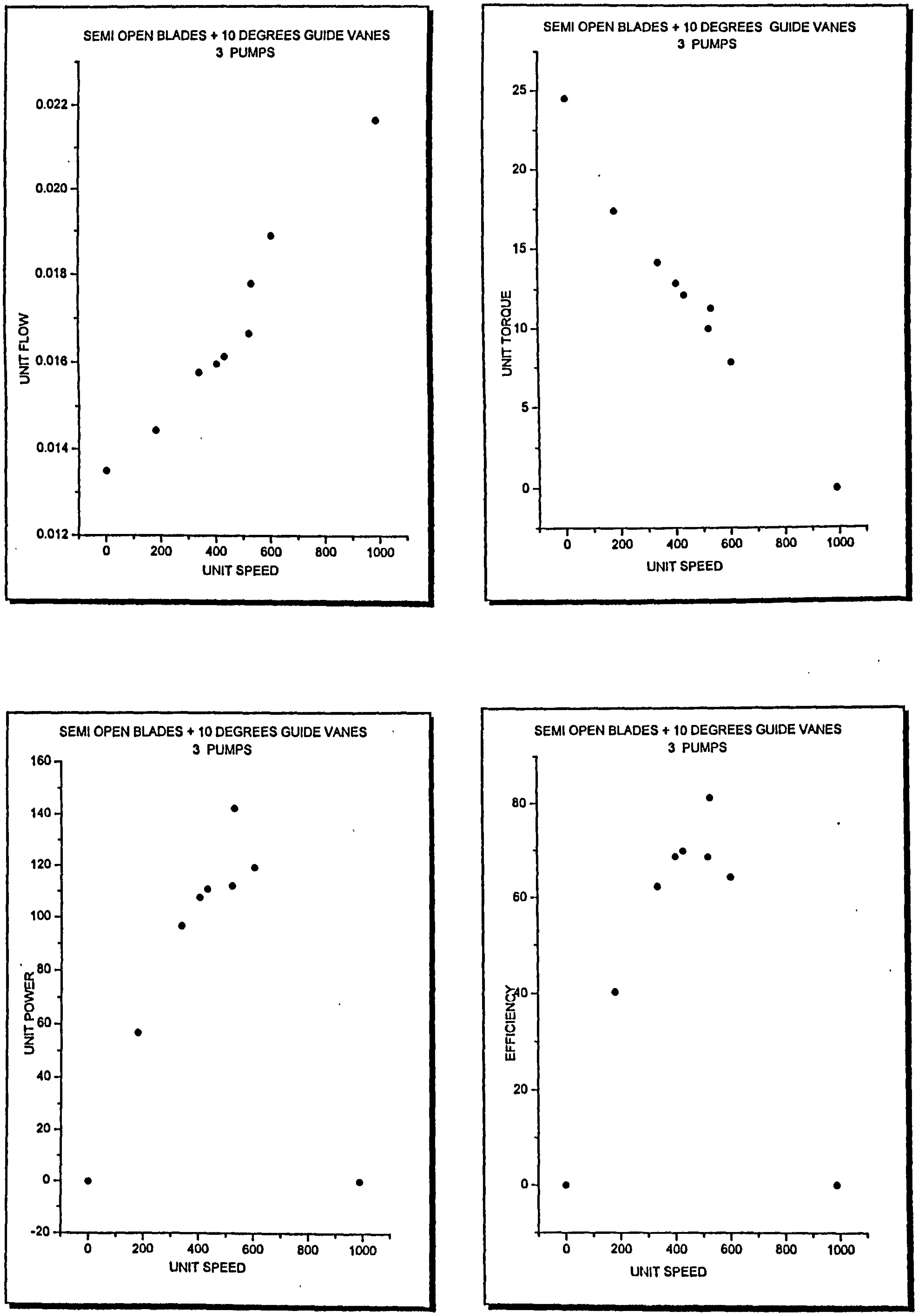

SEMI OPEN BLADES + 10 DEGREES GUIDE VANES
3 PUMPS
UNIT FLOW
UNIT SPEED
UNIT TORQUE
UNIT POWER
EFFICIENCY

Figure ( 6.3.6 )

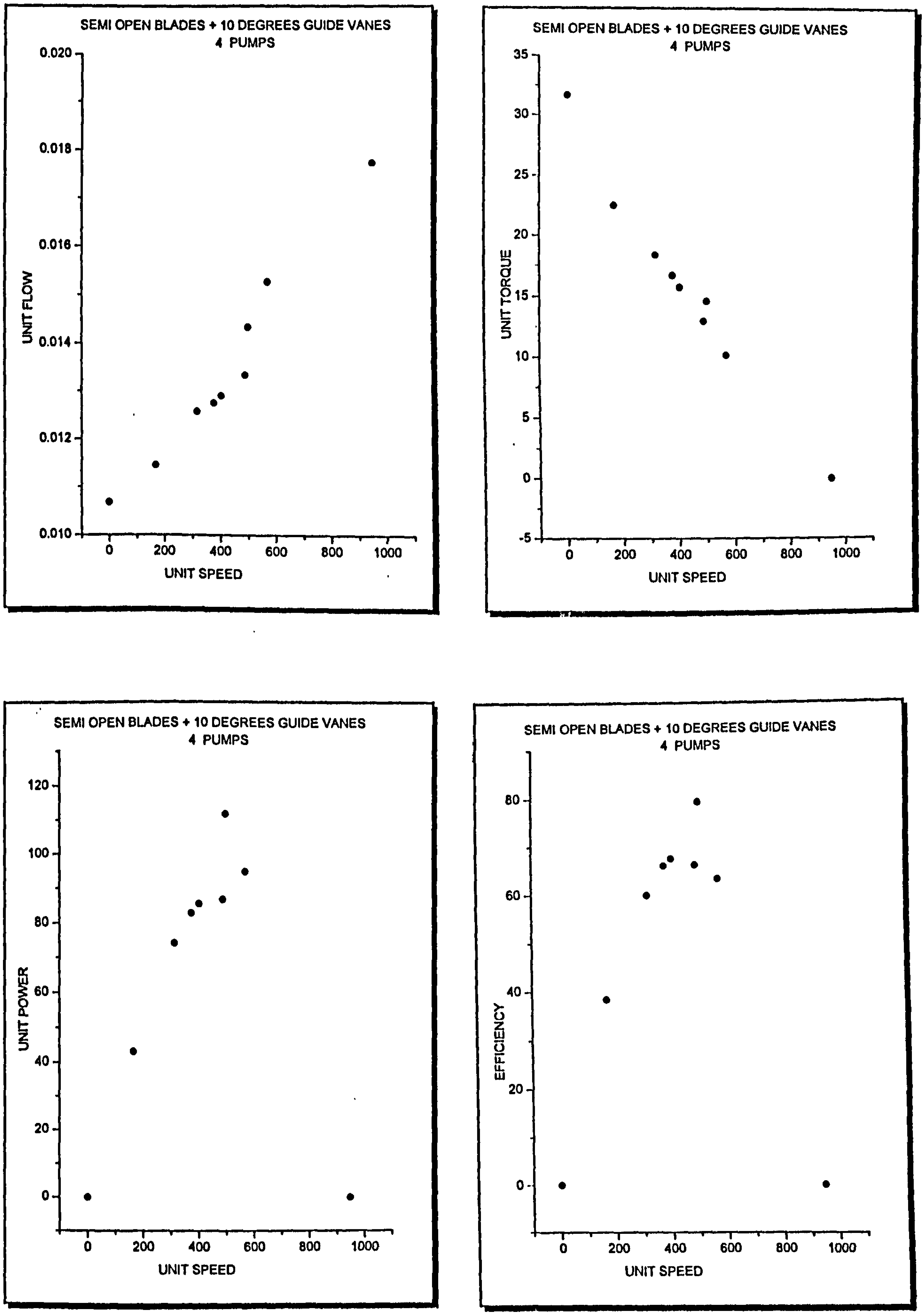

SEMI OPEN BLADES + 10 DEGREES GUIDE VANES
4 PUMPS
UNIT FLOW
UNIT SPEED
0.020
0.018
0.016
0.014
0.012
0.010
0
200
400
600
800
1000
SEMI OPEN BLADES + 10 DEGREES GUIDE VANES
4 PUMPS
UNIT TORQUE
UNIT SPEED
35
30
25
20
15
10
5
0
-5
SEMI OPEN BLADES + 10 DEGREES GUIDE VANES
4 PUMPS
UNIT POWER
UNIT SPEED
120
100
80
60
40
20
0
SEMI OPEN BLADES + 10 DEGREES GUIDE VANES
4 PUMPS
EFFICIENCY
UNIT SPEED
80
60
40
20
0

Figure ( 6.4.6 )

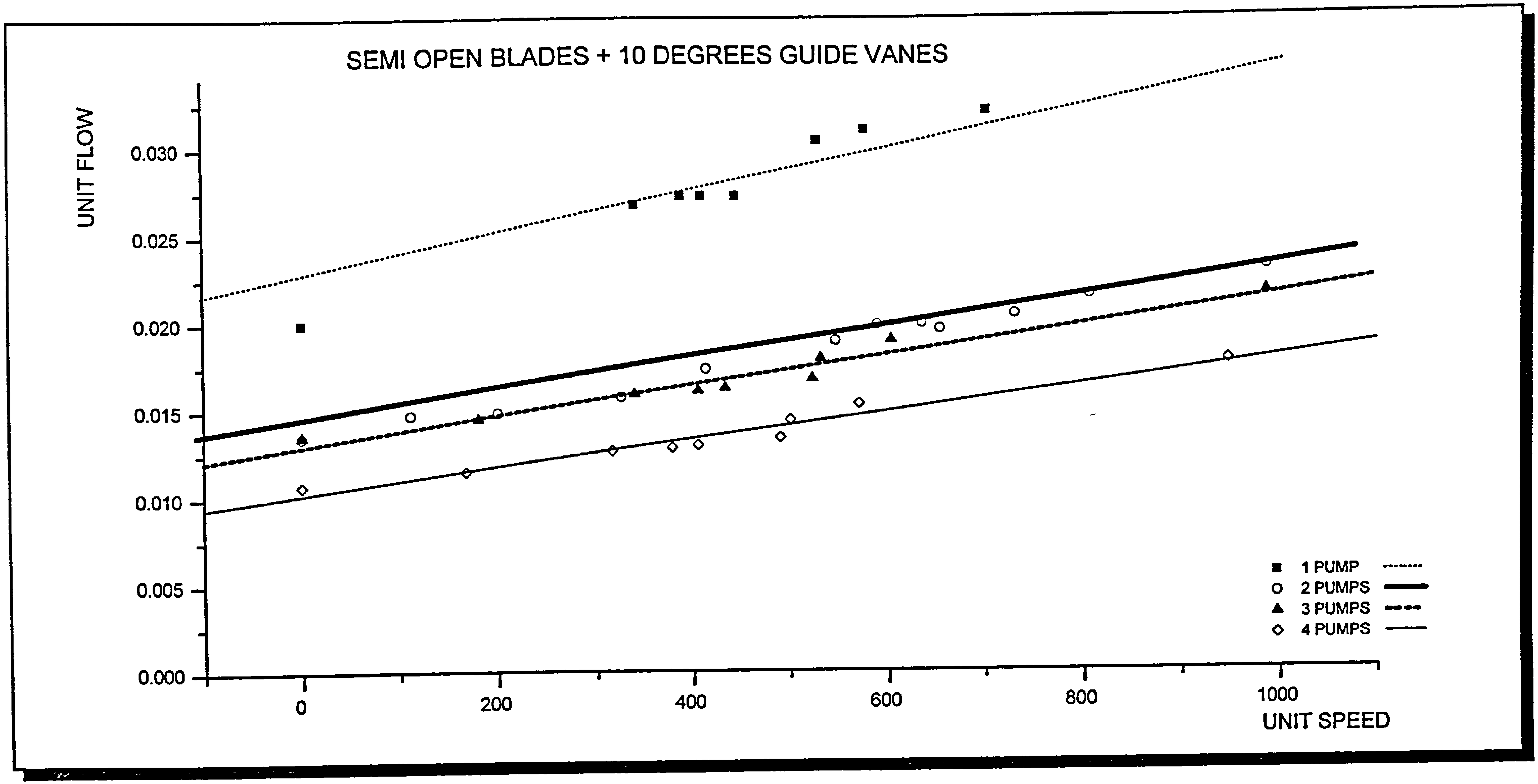

SEMI OPEN BLADES + 10 DEGREES GUIDE VANES
UNIT FLOW
0.030
0.025
0.020
0.015
0.010
0.005
0.000
0
200
400
600
800
1000
UNIT SPEED
1 PUMP
2 PUMPS
3 PUMPS
4 PUMPS

Figure ( 6.5.6 )

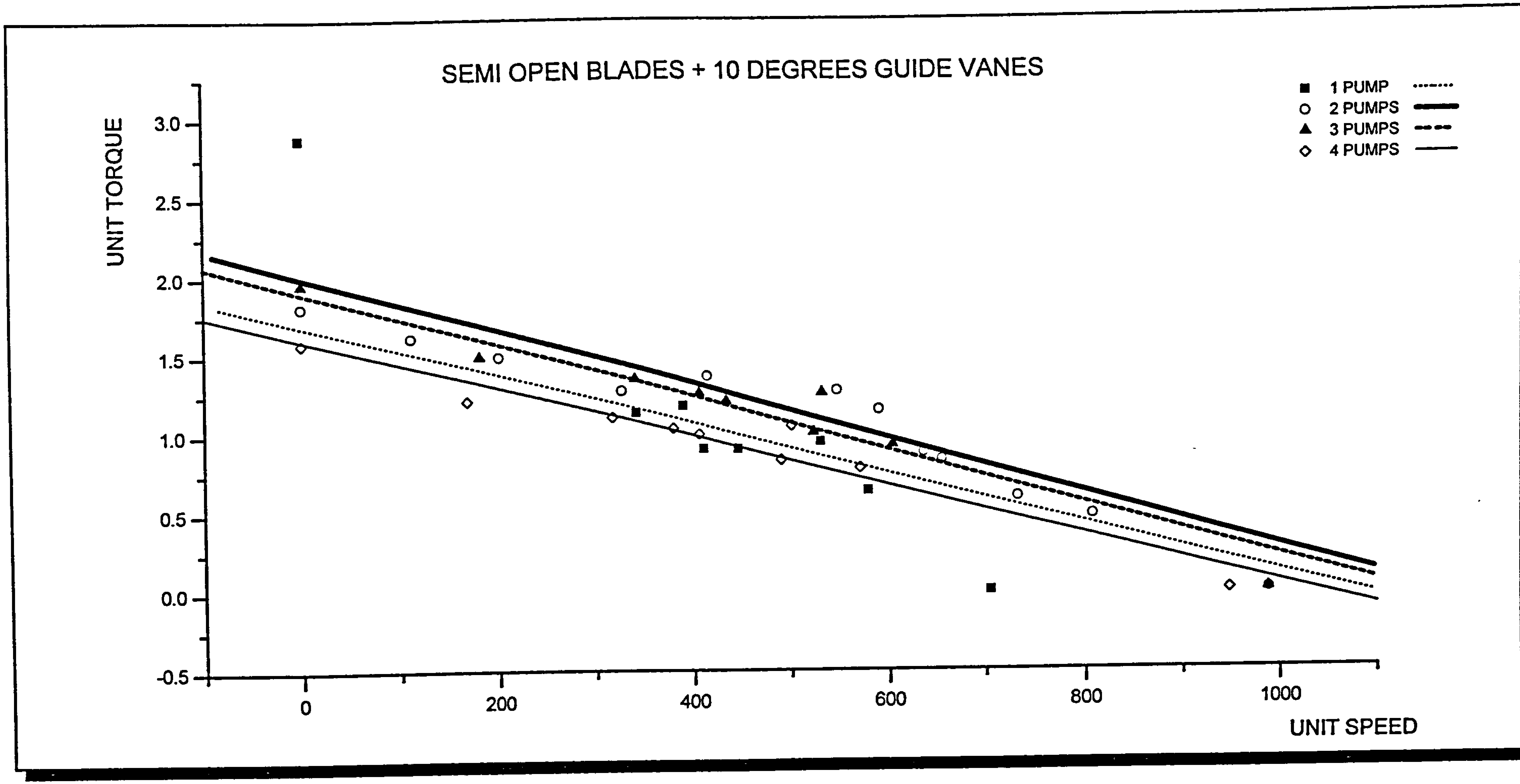
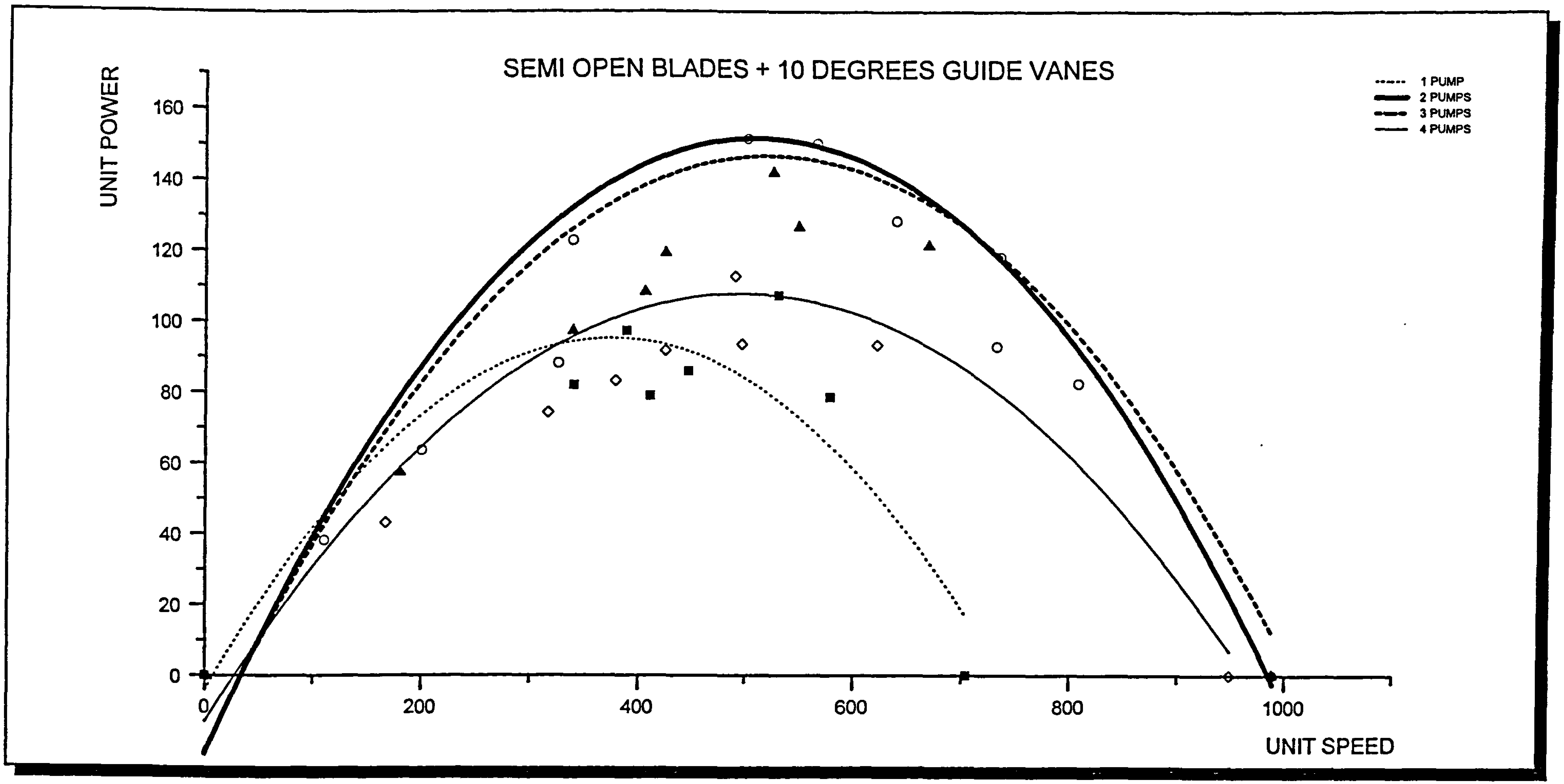
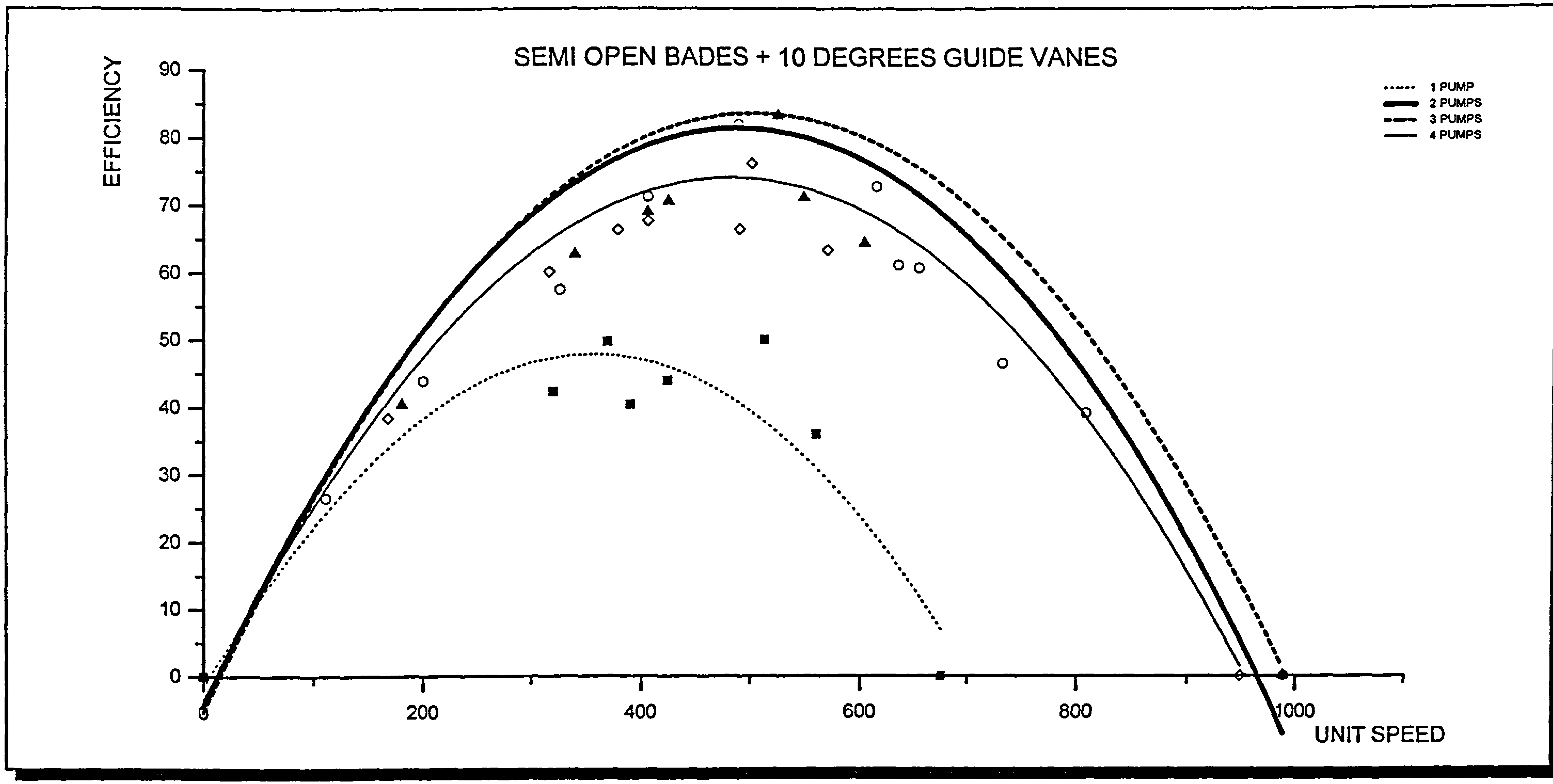

SEMI OPEN BLADES + 10 DEGREES GUIDE VANES
UNIT TORQUE
3.0
2.5
2.0
1.5
1.0
0.5
0.0
-0.5
0
200
400
600
800
1000
UNIT SPEED
1 PUMP
2 PUMPS
3 PUMPS
4 PUMPS

Figure ( 6.6.6 )



SEMI OPEN BLADES + 10 DEGREES GUIDE VANES
UNIT POWER
160
140
120
100
80
60
40
20
0
0
200
400
600
800
1000
UNIT SPEED
1 PUMP
2 PUMPS
3 PUMPS
4 PUMPS


Figure ( 6.7.6 )



SEMI OPEN BADES + 10 DEGREES GUIDE VANES
EFFICIENCY
0
10
20
30
40
50
60
70
80
90
0
200
400
600
800
1000
UNIT SPEED
1 PUMP
2 PUMPS
3 PUMPS
4 PUMPS


Figure ( 6.8.6 )

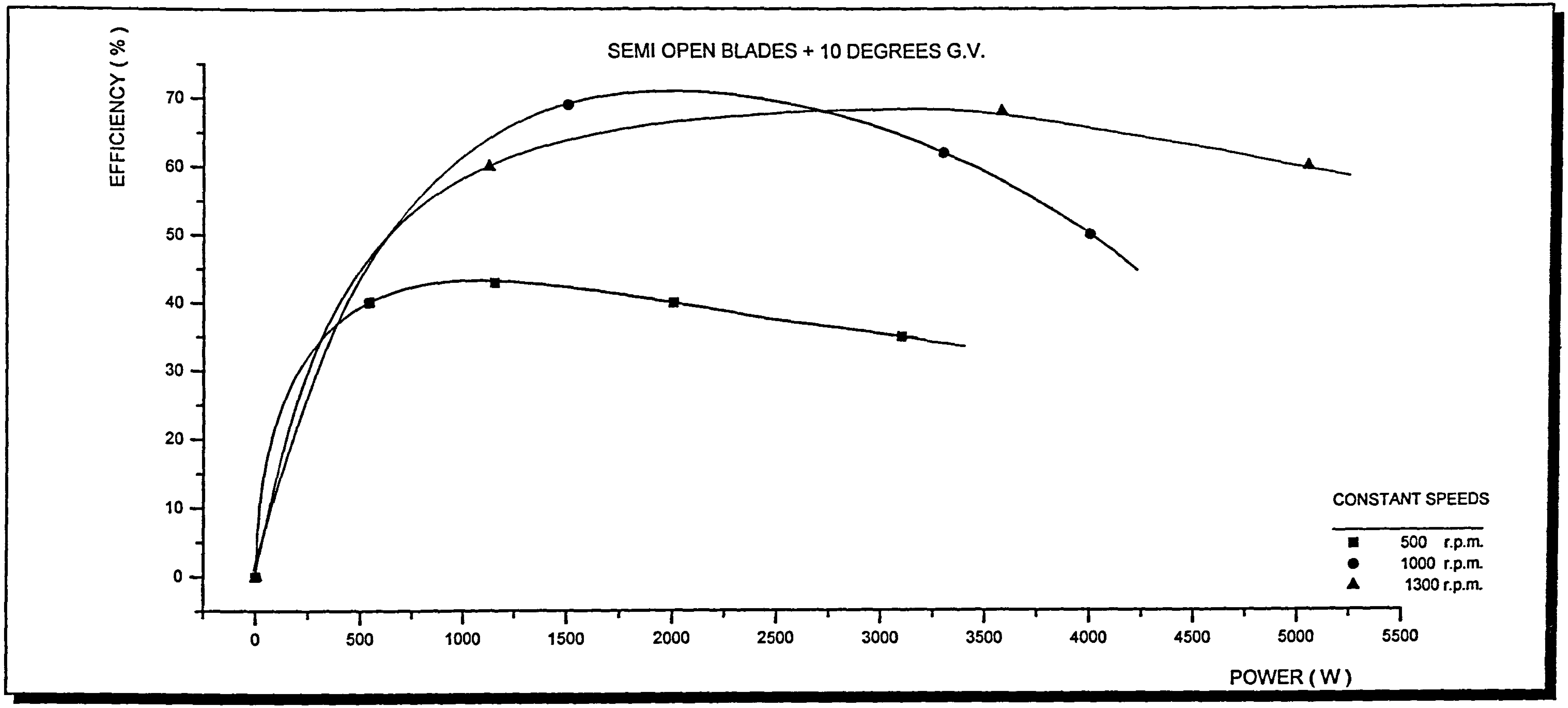

SEMI OPEN BLADES + 10 DEGREES G.V.
EFFICIENCY ( % )
0
10
20
30
40
50
60
70
0
500
1000
1500
2000
2500
3000
3500
4000
4500
5000
5500
POWER ( W )
CONSTANT SPEEDS
500 r.p.m.
1000 r.p.m.
1300 r.p.m.

Figure ( 6.9.6 )

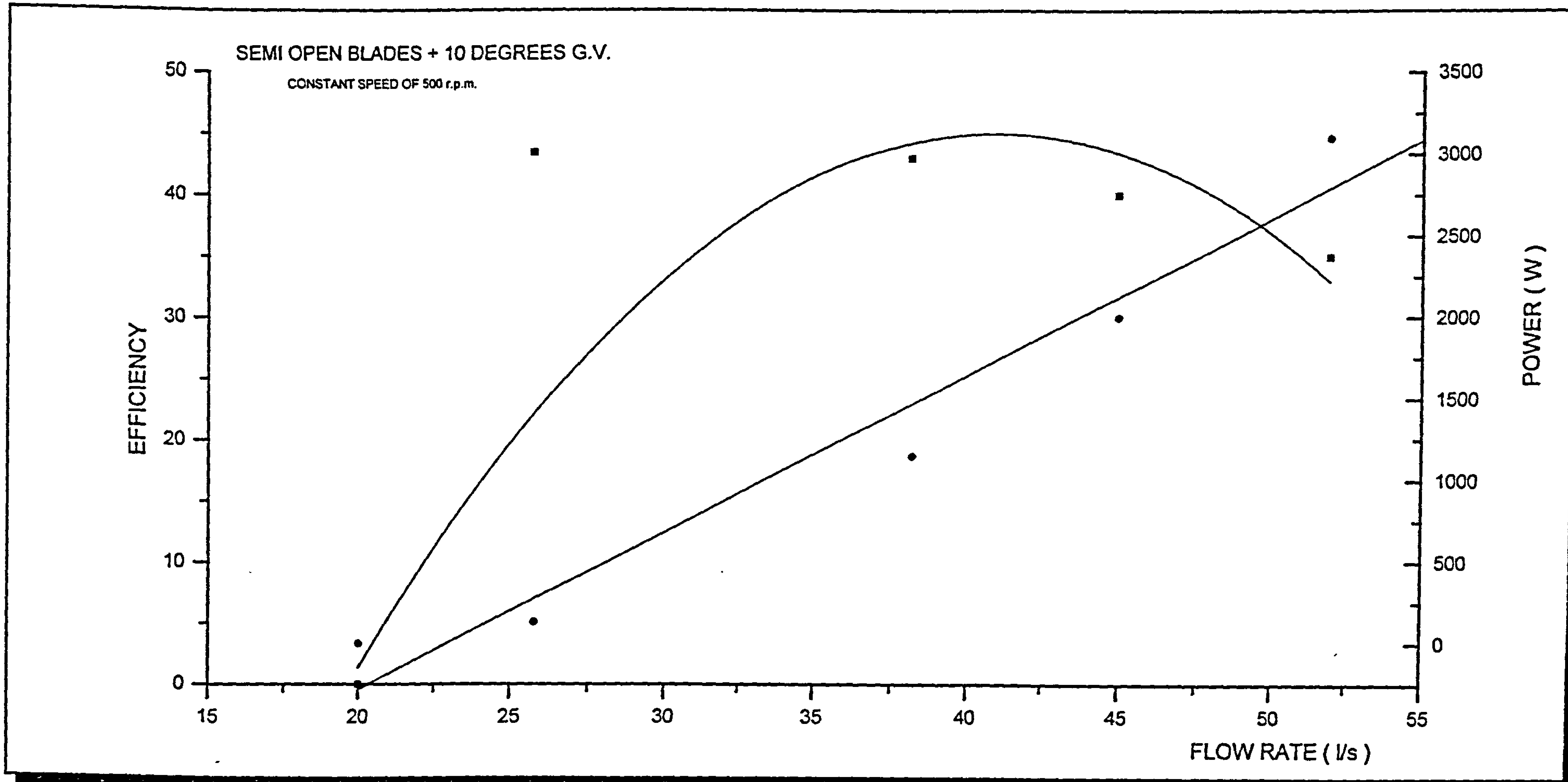

SEMI OPEN BLADES + 10 DEGREES G.V.
CONSTANT SPEED OF 500 r.p.m.
EFFICIENCY
50
40
30
20
10
0
POWER ( W )
3500
3000
2500
2000
1500
1000
500
0
15
20
25
30
35
40
45
50
55
FLOW RATE ( l/s )

Figure ( 6.10.6 )

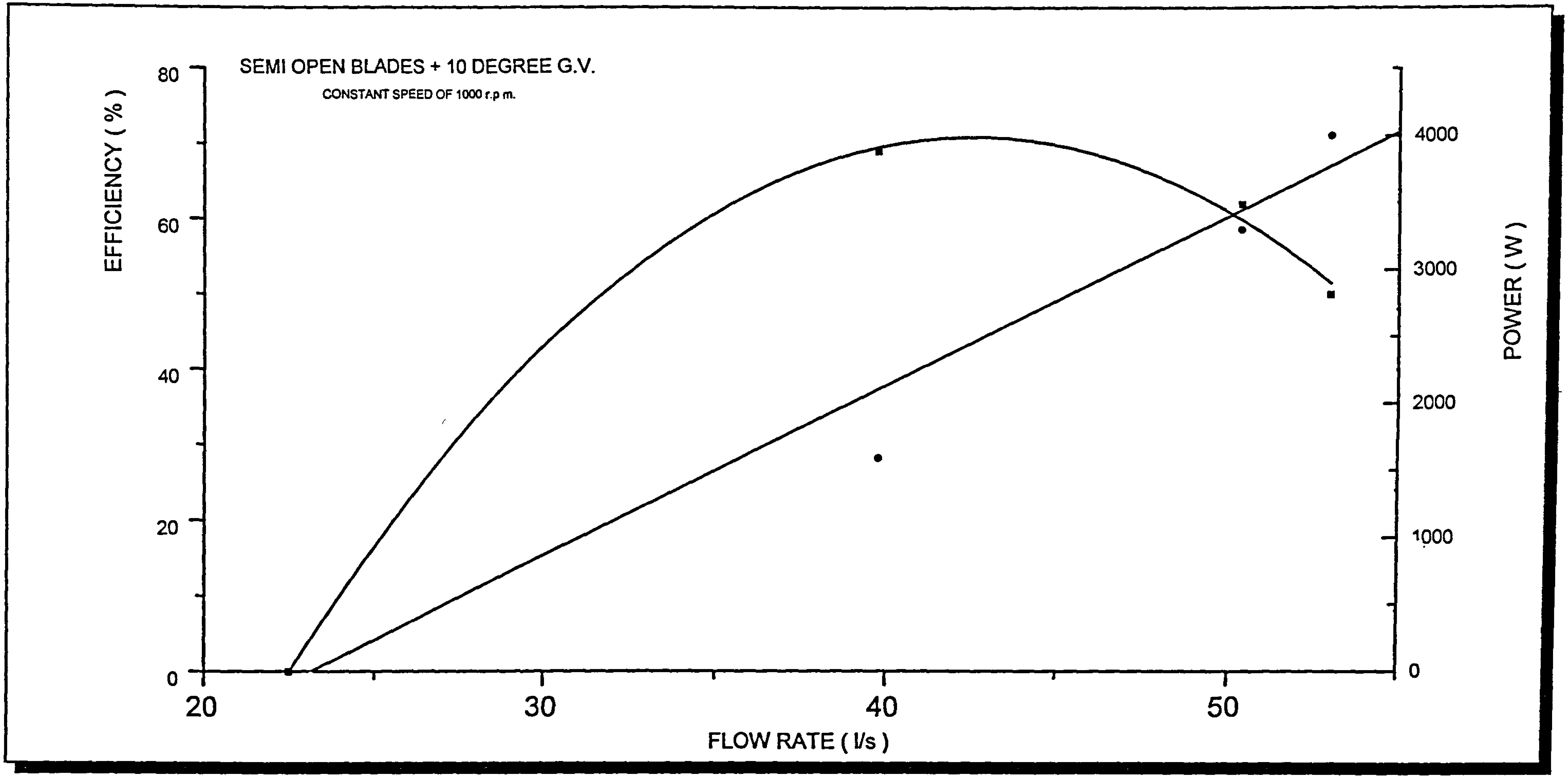

SEMI OPEN BLADES + 10 DEGREE G.V.
CONSTANT SPEED OF 1000 r.p m.
EFFICIENCY ( % )
80
60
40
20
0
20
30
40
50
FLOW RATE ( l/s )
POWER ( W )
4000
3000
2000
1000
0

Figure ( 6.11.6 )

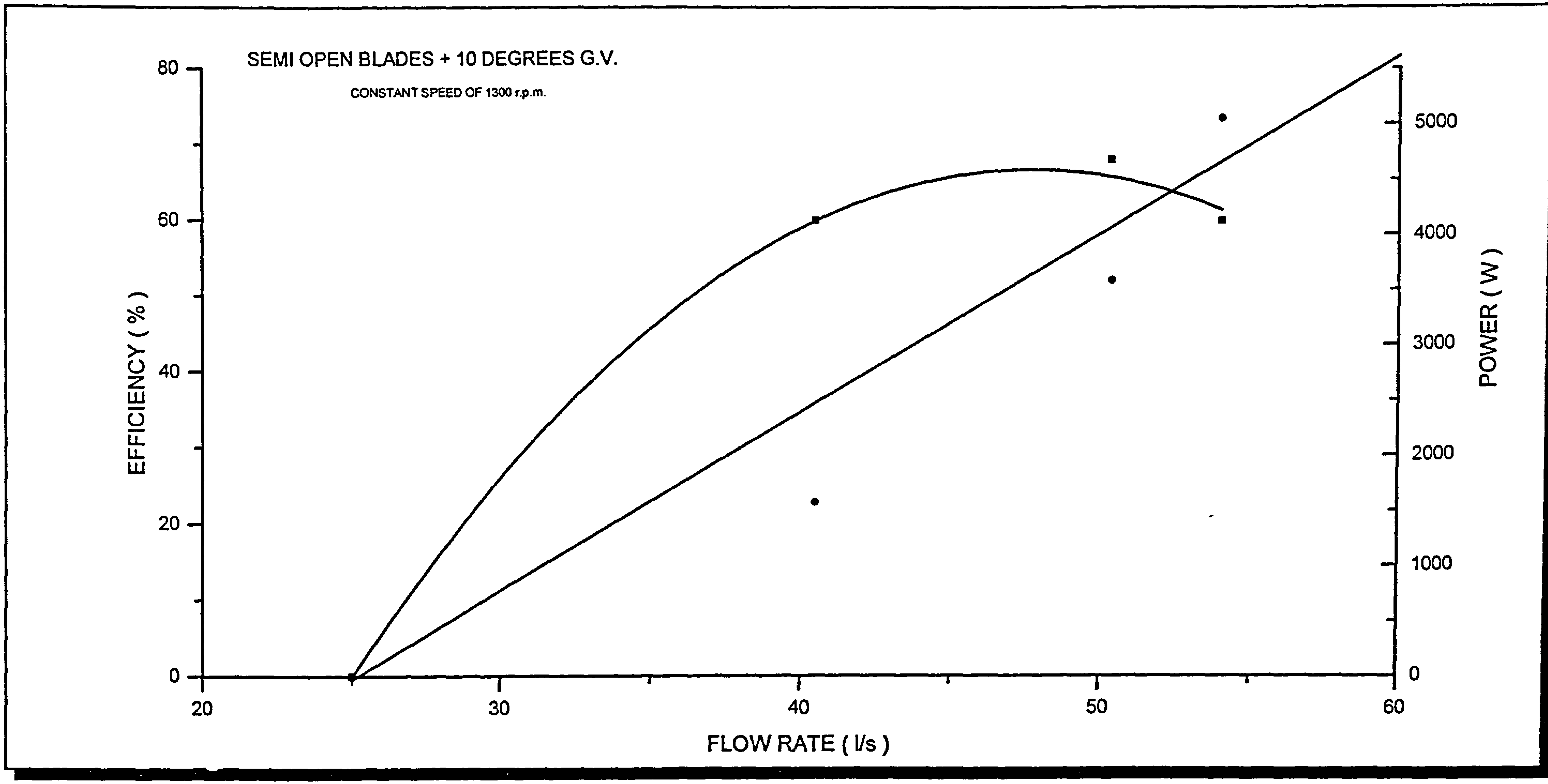

SEMI OPEN BLADES + 10 DEGREES G.V.
CONSTANT SPEED OF 1300 r.p.m.
EFFICIENCY ( % )
0
20
40
60
80
FLOW RATE ( l/s )
20
30
40
50
60
POWER ( W )
0
1000
2000
3000
4000
5000

Figure ( 6.12.6 )

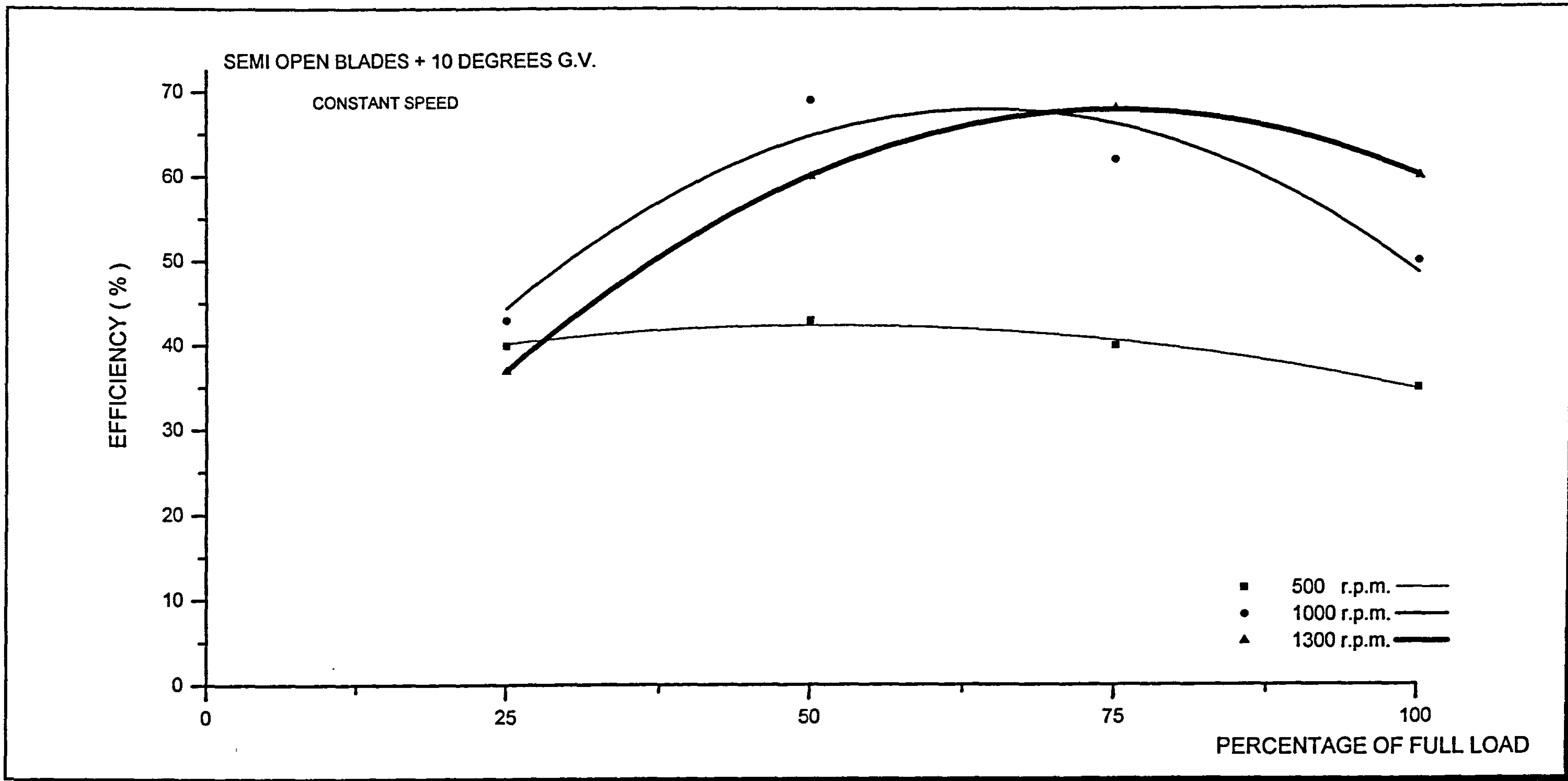

SEMI OPEN BLADES + 10 DEGREES G.V.
CONSTANT SPEED
EFFICIENCY ( % )
0
10
20
30
40
50
60
70
0
25
50
75
100
PERCENTAGE OF FULL LOAD
500 r.p.m.
1000 r.p.m.
1300 r.p.m.

Figure ( 6.13.6 )

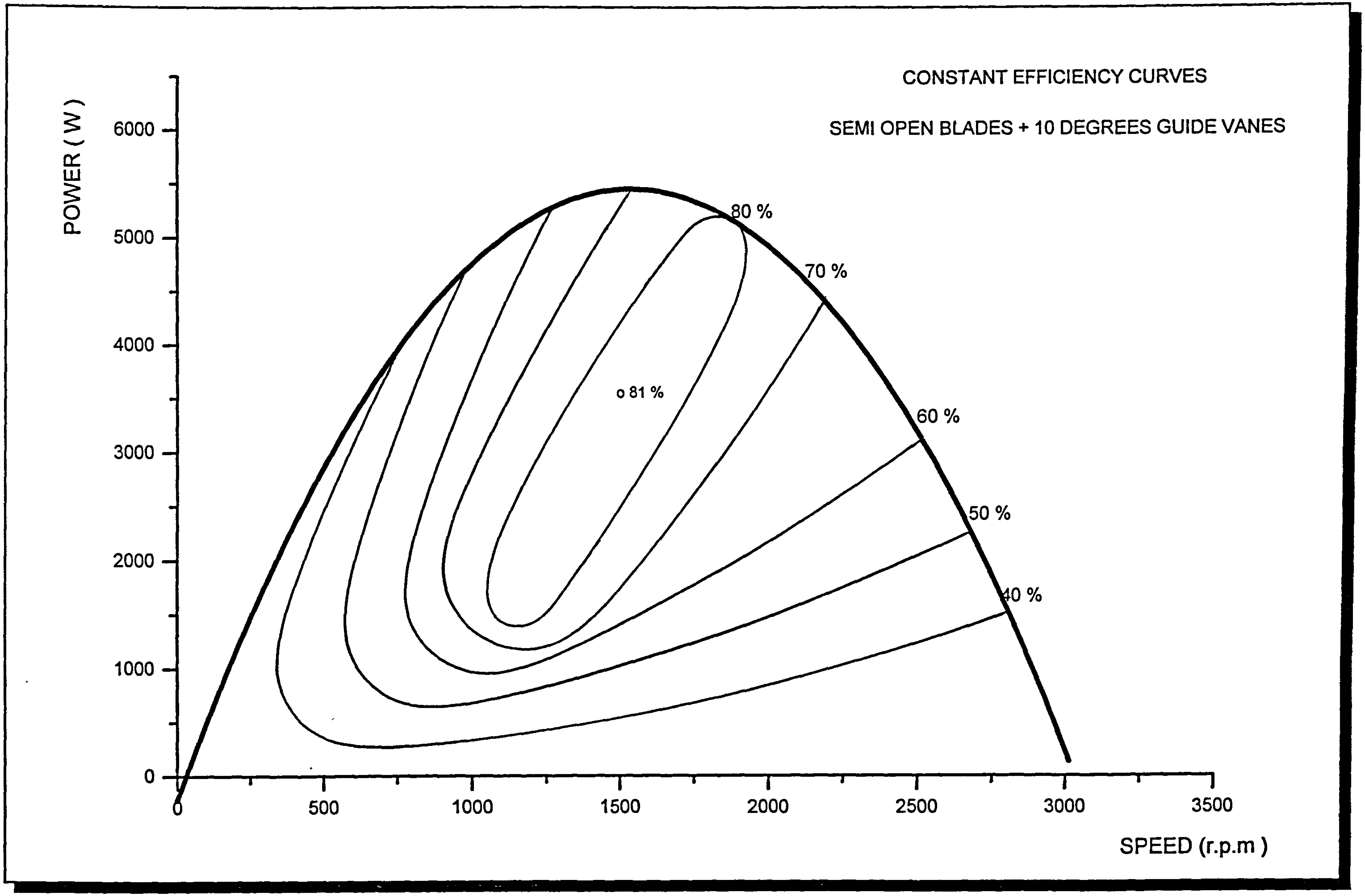

CONSTANT EFFICIENCY CURVES
SEMI OPEN BLADES + 10 DEGREES GUIDE VANES
POWER ( W )
6000
5000
4000
3000
2000
1000
0
0
500
1000
1500
2000
2500
3000
3500
SPEED (r.p.m )
80 %
70 %
60 %
50 %
40 %
o 81 %

Figure ( 6.14.6 )

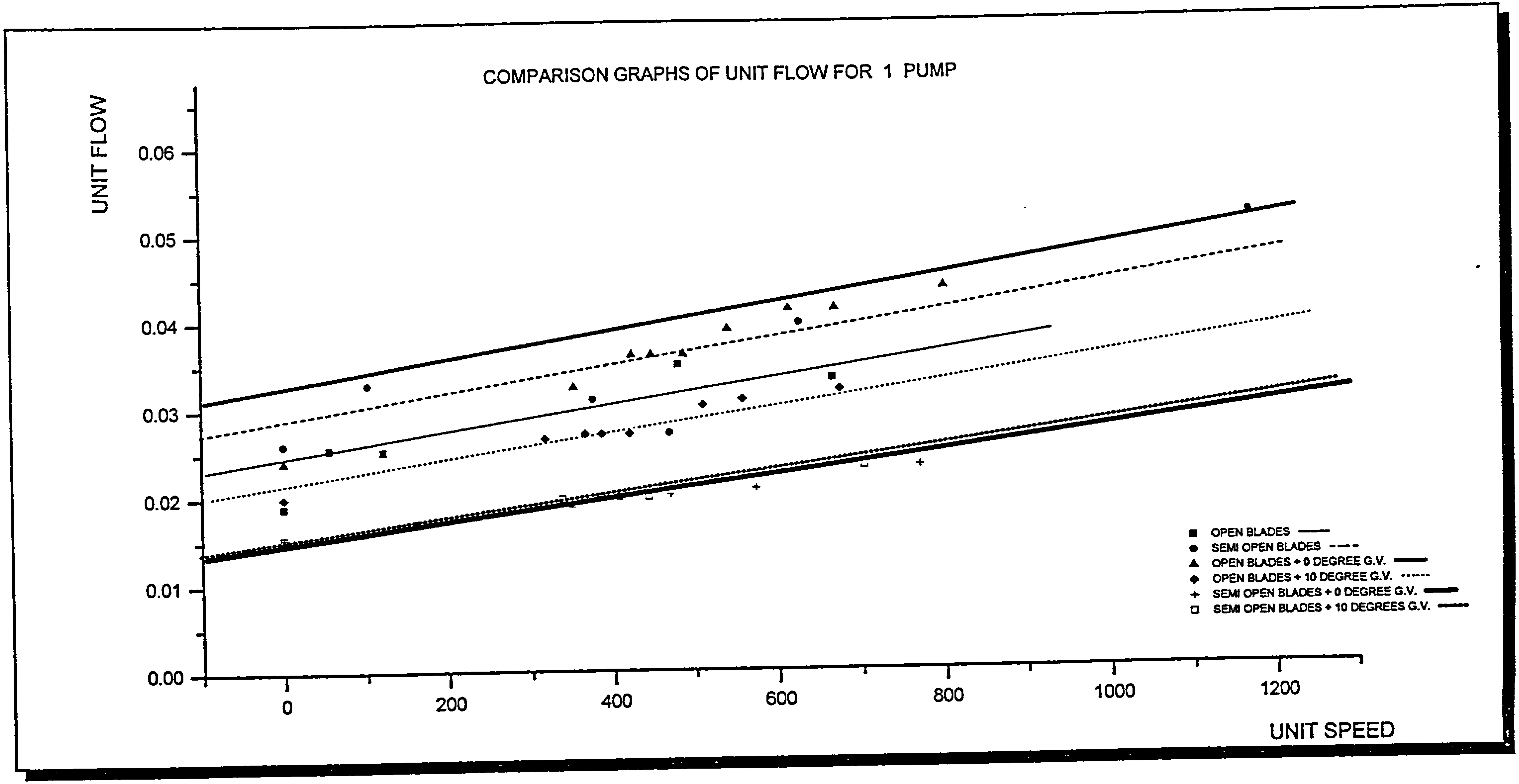

COMPARISON GRAPHS OF UNIT FLOW FOR 1 PUMP
UNIT FLOW
0.06
0.05
0.04
0.03
0.02
0.01
0.00
0
200
400
600
800
1000
1200
UNIT SPEED
OPEN BLADES
SEMI OPEN BLADES
OPEN BLADES + 0 DEGREE G.V.
OPEN BLADES + 10 DEGREE G.V.
SEMI OPEN BLADES + 0 DEGREE G.V.
SEMI OPEN BLADES + 10 DEGREES G.V.

Figure ( 6.15 )

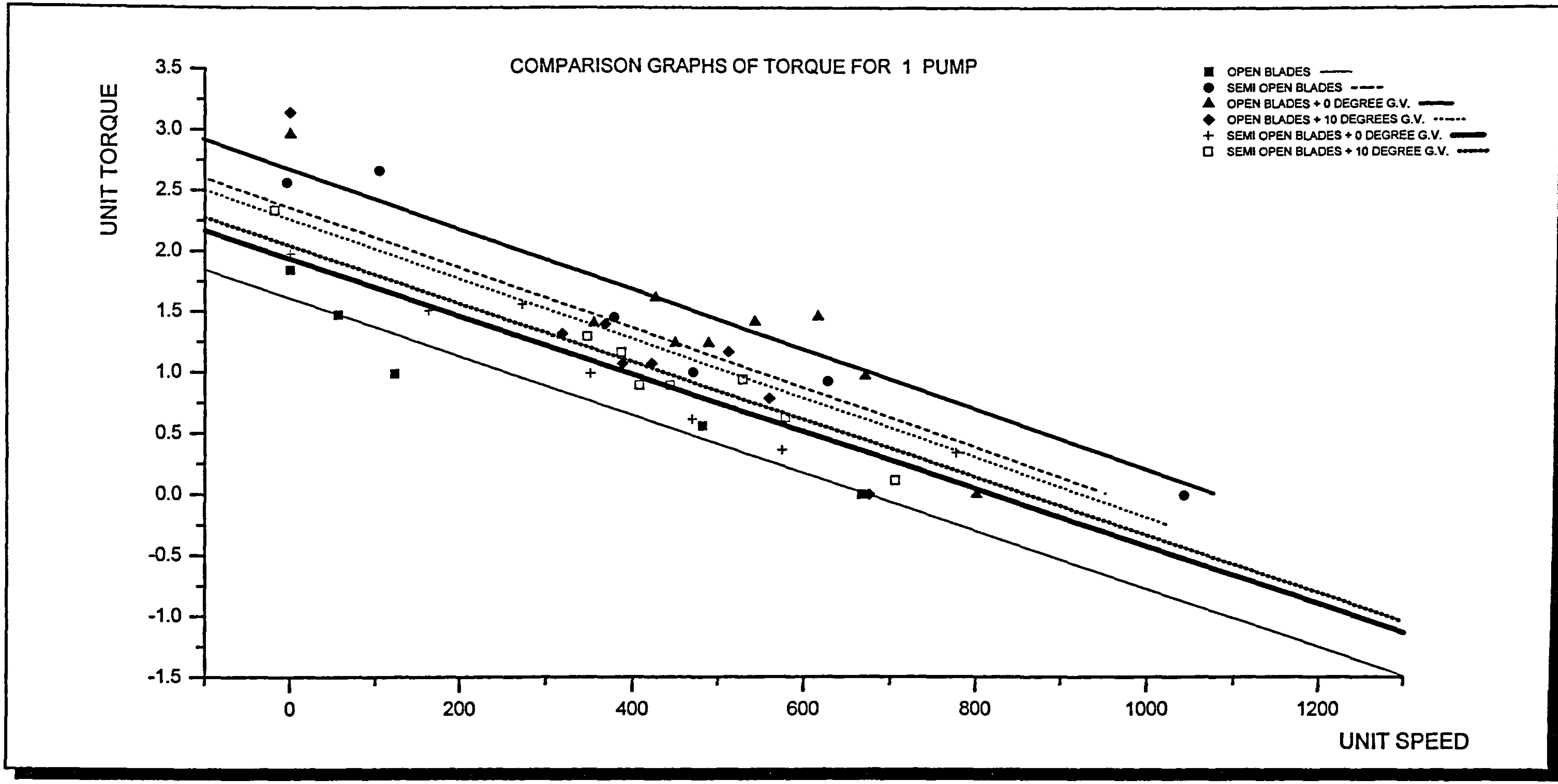

COMPARISON GRAPHS OF TORQUE FOR 1 PUMP
UNIT TORQUE
UNIT SPEED
3.5
3.0
2.5
2.0
1.5
1.0
0.5
0.0
-0.5
-1.0
-1.5
0
200
400
600
800
1000
1200
OPEN BLADES
SEMI OPEN BLADES
OPEN BLADES + 0 DEGREE G.V.
OPEN BLADES + 10 DEGREES G.V.
SEMI OPEN BLADES + 0 DEGREE G.V.
SEMI OPEN BLADES + 10 DEGREE G.V.

Figure ( 6.16 )

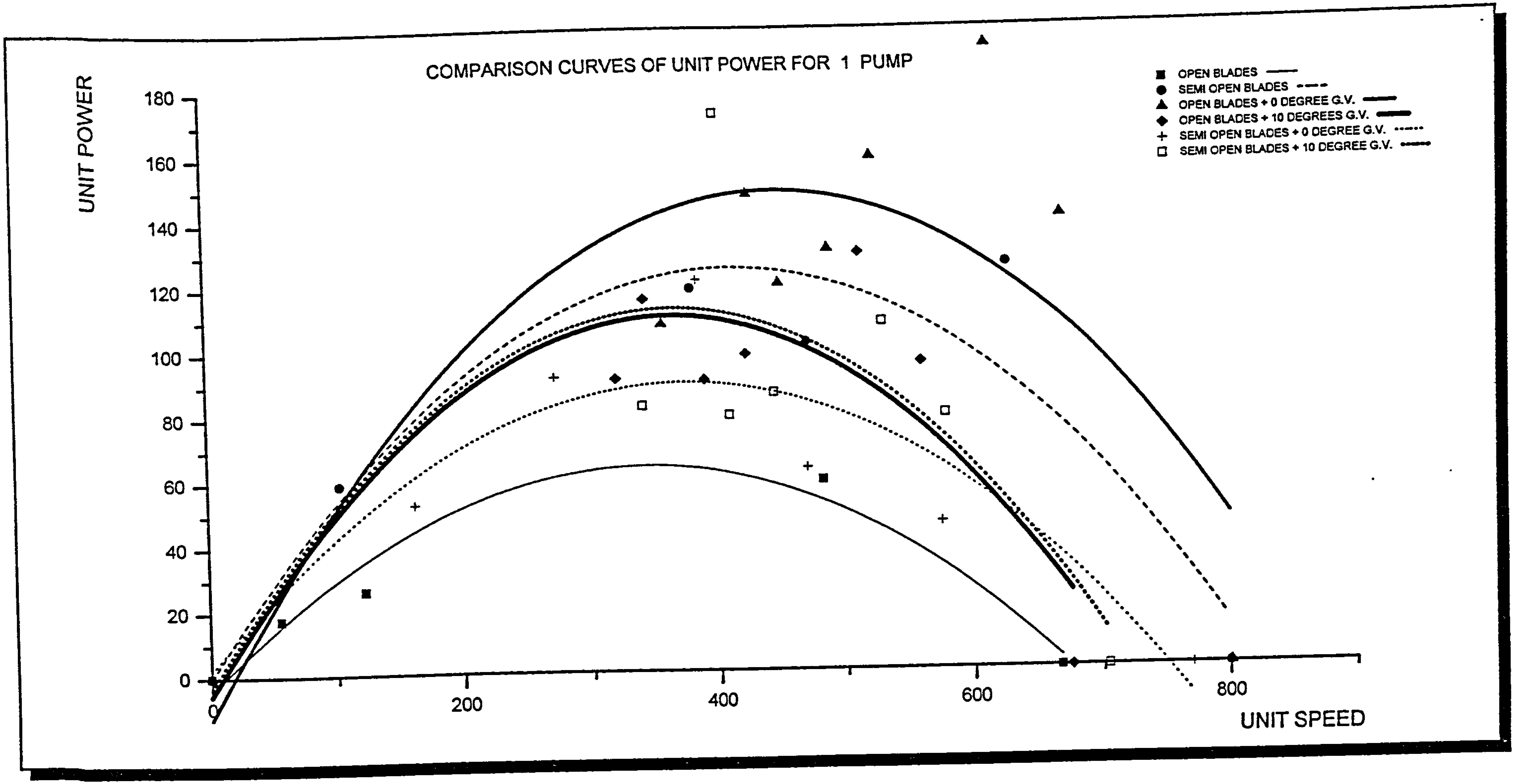

COMPARISON CURVES OF UNIT POWER FOR 1 PUMP
UNIT POWER
180
160
140
120
100
80
60
40
20
0
0
200
400
600
800
UNIT SPEED
OPEN BLADES
SEMI OPEN BLADES
OPEN BLADES + 0 DEGREE G.V.
OPEN BLADES + 10 DEGREES G.V.
SEMI OPEN BLADES + 0 DEGREE G.V.
SEMI OPEN BLADES + 10 DEGREE G.V.

Figure ( 6.17 )

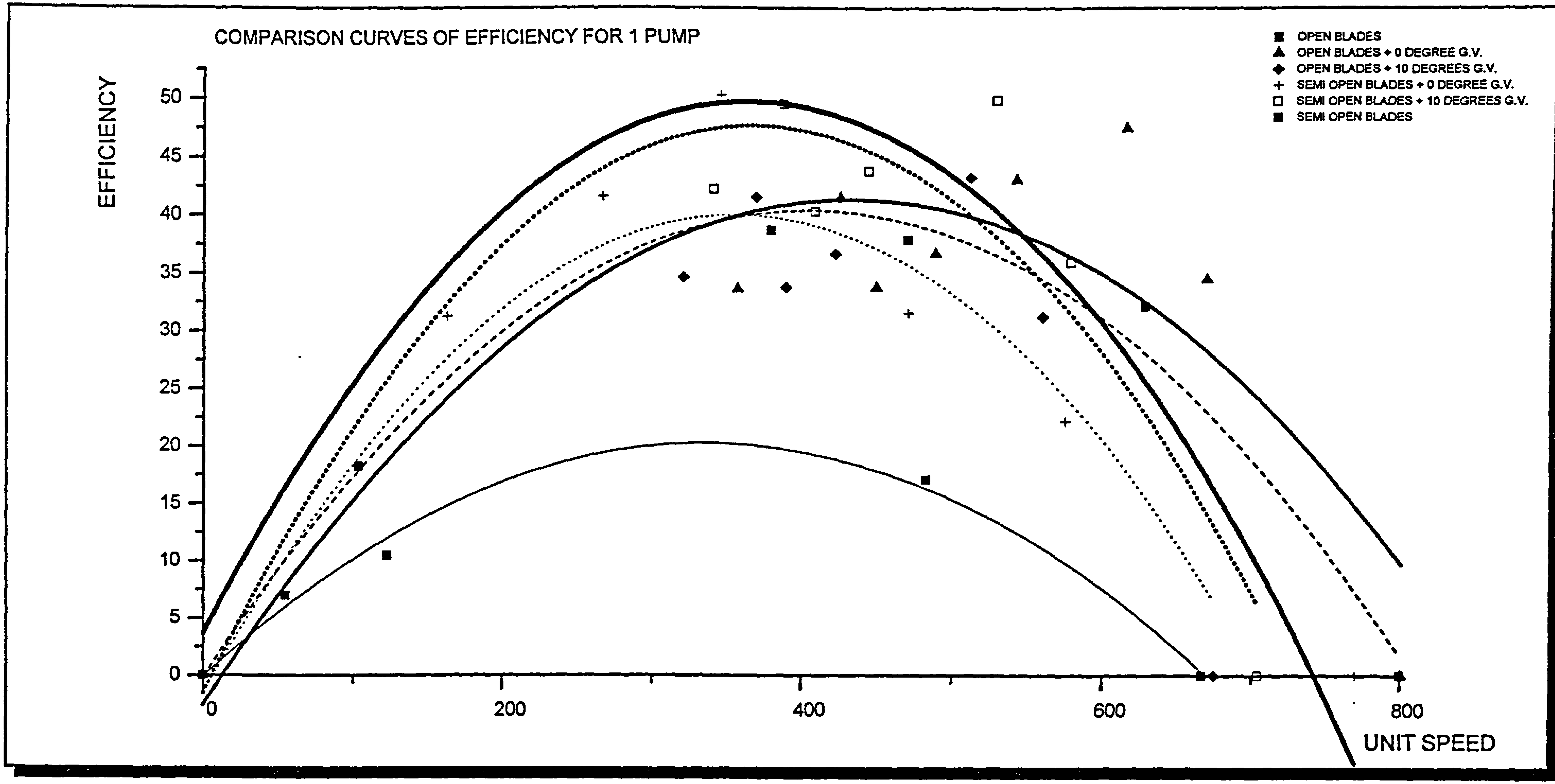

COMPARISON CURVES OF EFFICIENCY FOR 1 PUMP
EFFICIENCY
50
45
40
35
30
25
20
15
10
5
0
0
200
400
600
800
UNIT SPEED
OPEN BLADES
OPEN BLADES + 0 DEGREE G.V.
OPEN BLADES + 10 DEGREES G.V.
SEMI OPEN BLADES + 0 DEGREE G.V.
SEMI OPEN BLADES + 10 DEGREES G.V.
SEMI OPEN BLADES

Figure ( 6.18 )

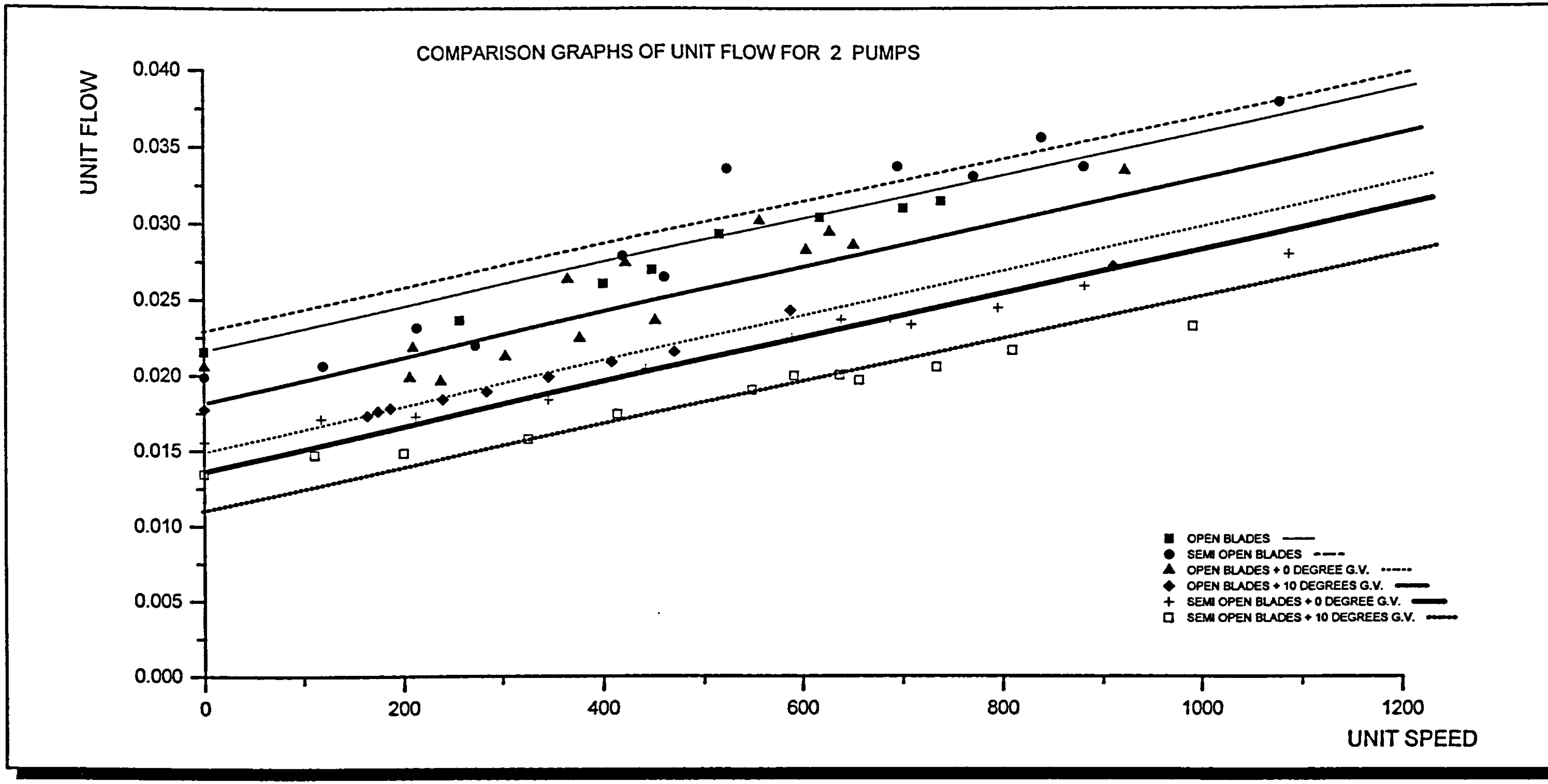

COMPARISON GRAPHS OF UNIT FLOW FOR 2 PUMPS
UNIT FLOW
0.040
0.035
0.030
0.025
0.020
0.015
0.010
0.005
0.000
0
200
400
600
800
1000
1200
UNIT SPEED
OPEN BLADES
SEMI OPEN BLADES
OPEN BLADES + 0 DEGREE G.V.
OPEN BLADES + 10 DEGREES G.V.
SEMI OPEN BLADES + 0 DEGREE G.V.
SEMI OPEN BLADES + 10 DEGREES G.V.

Figure ( 6.19 )

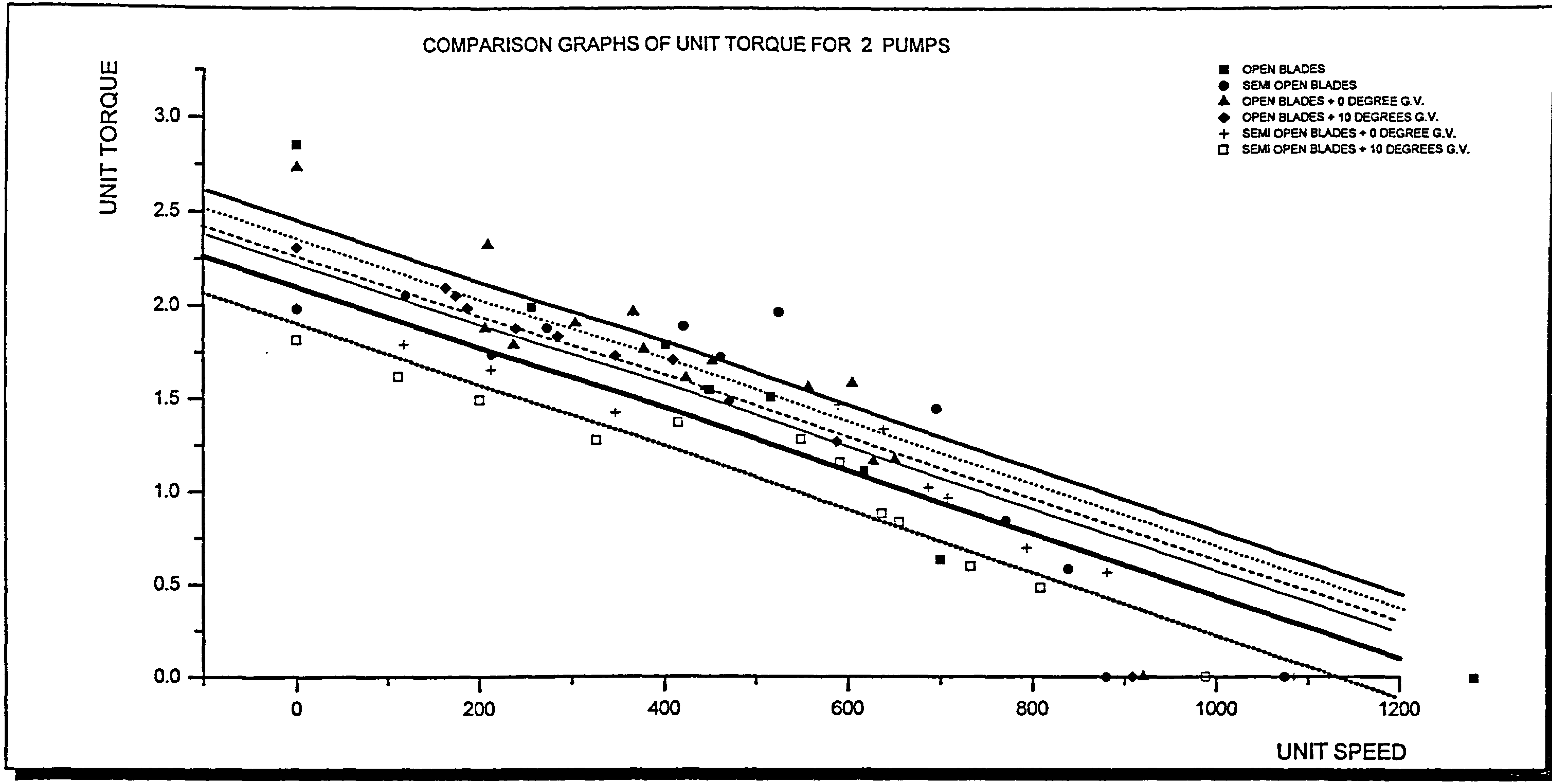

COMPARISON GRAPHS OF UNIT TORQUE FOR 2 PUMPS
UNIT TORQUE
3.0
2.5
2.0
1.5
1.0
0.5
0.0
0
200
400
600
800
1000
1200
UNIT SPEED
OPEN BLADES
SEMI OPEN BLADES
OPEN BLADES + 0 DEGREE G.V.
OPEN BLADES + 10 DEGREES G.V.
SEMI OPEN BLADES + 0 DEGREE G.V.
SEMI OPEN BLADES + 10 DEGREES G.V.

Figure ( 6.20 )

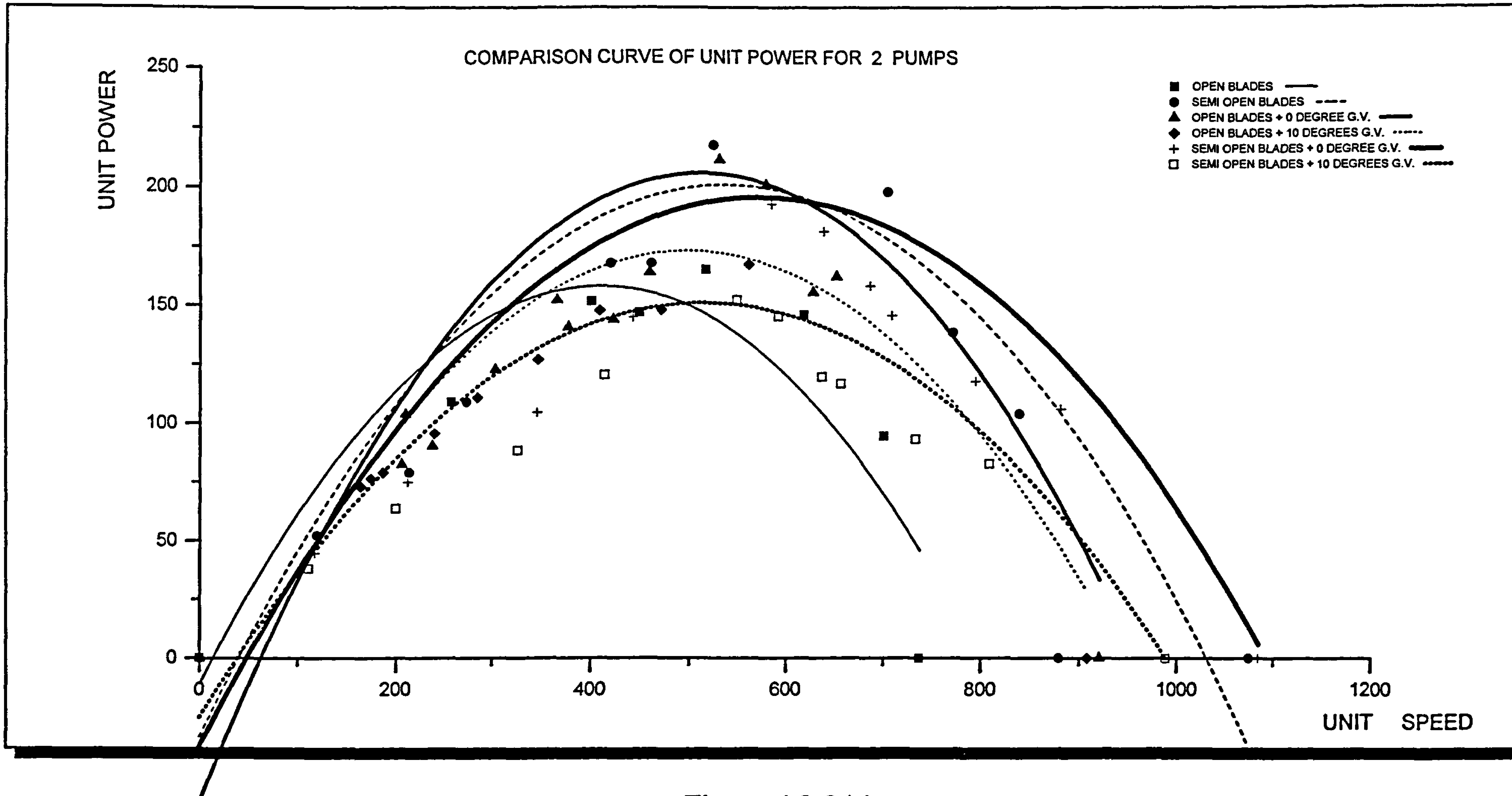

COMPARISON CURVE OF UNIT POWER FOR 2 PUMPS
UNIT POWER
250
200
150
100
50
0
0
200
400
600
800
1000
1200
UNIT SPEED
OPEN BLADES
SEMI OPEN BLADES
OPEN BLADES + 0 DEGREE G.V.
OPEN BLADES + 10 DEGREES G.V.
SEMI OPEN BLADES + 0 DEGREE G.V.
SEMI OPEN BLADES + 10 DEGREES G.V.

Figure ( 6.21 )

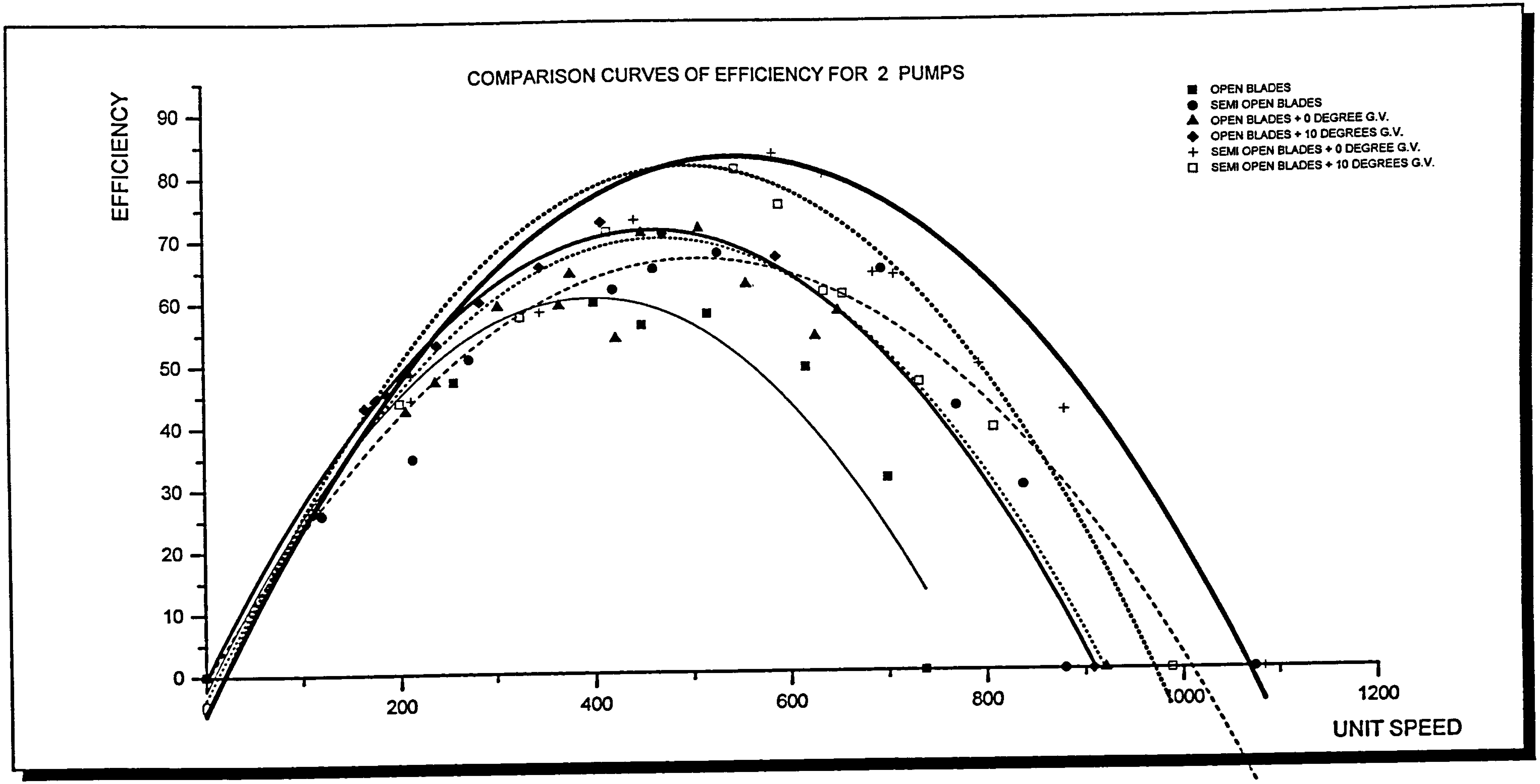
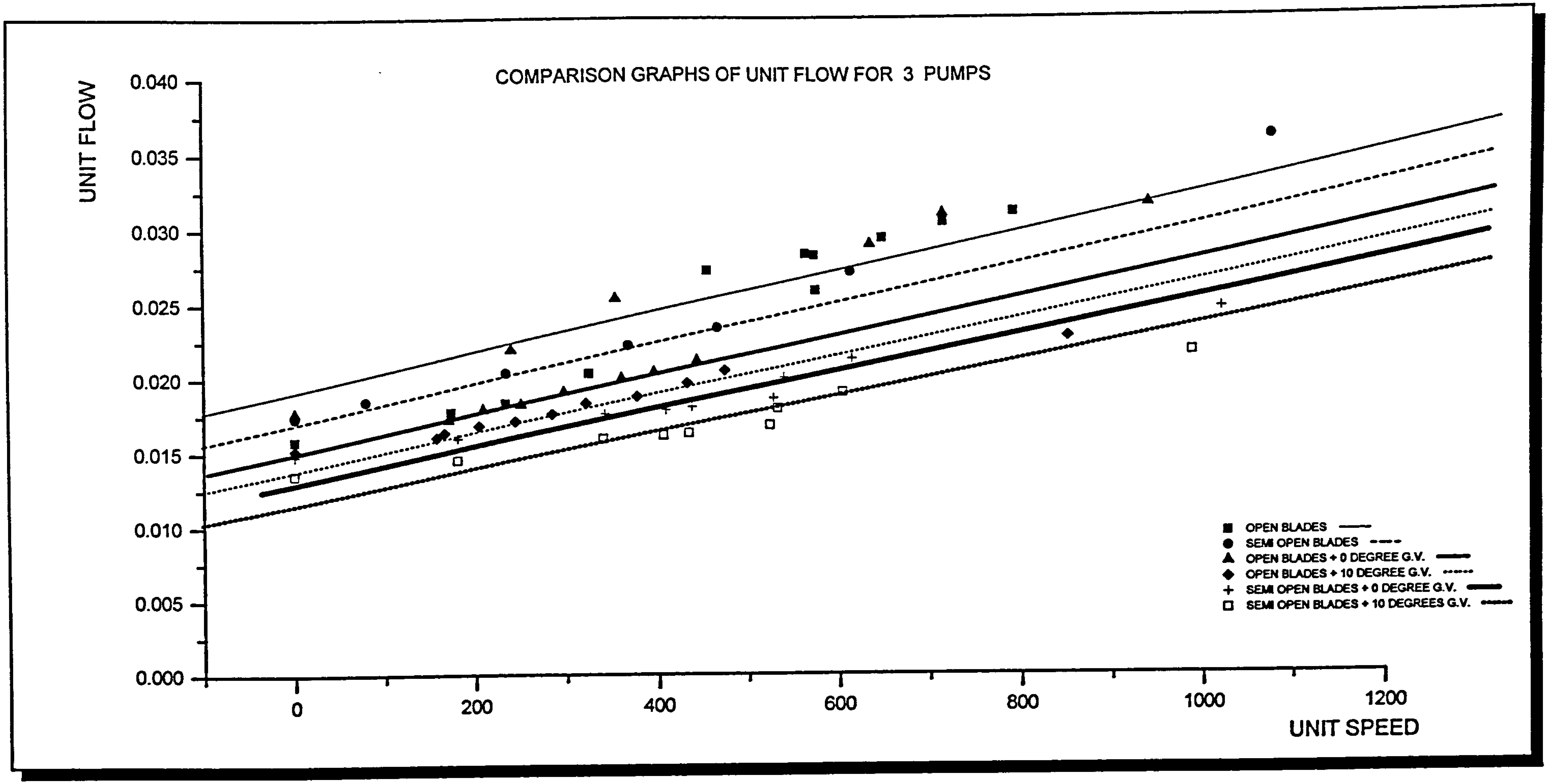

COMPARISON CURVES OF EFFICIENCY FOR 2 PUMPS
EFFICIENCY
0
10
20
30
40
50
60
70
80
90
200
400
600
800
1000
1200
UNIT SPEED
OPEN BLADES
SEMI OPEN BLADES
OPEN BLADES + 0 DEGREE G.V.
OPEN BLADES + 10 DEGREES G.V.
SEMI OPEN BLADES + 0 DEGREE G.V.
SEMI OPEN BLADES + 10 DEGREES G.V.

Figure ( 6.22 )



COMPARISON GRAPHS OF UNIT FLOW FOR 3 PUMPS
UNIT FLOW
0.040
0.035
0.030
0.025
0.020
0.015
0.010
0.005
0.000
0
200
400
600
800
1000
1200
UNIT SPEED
OPEN BLADES
SEMI OPEN BLADES
OPEN BLADES + 0 DEGREE G.V.
OPEN BLADES + 10 DEGREE G.V.
SEMI OPEN BLADES + 0 DEGREE G.V.
SEMI OPEN BLADES + 10 DEGREES G.V.


Figure ( 6.23 )

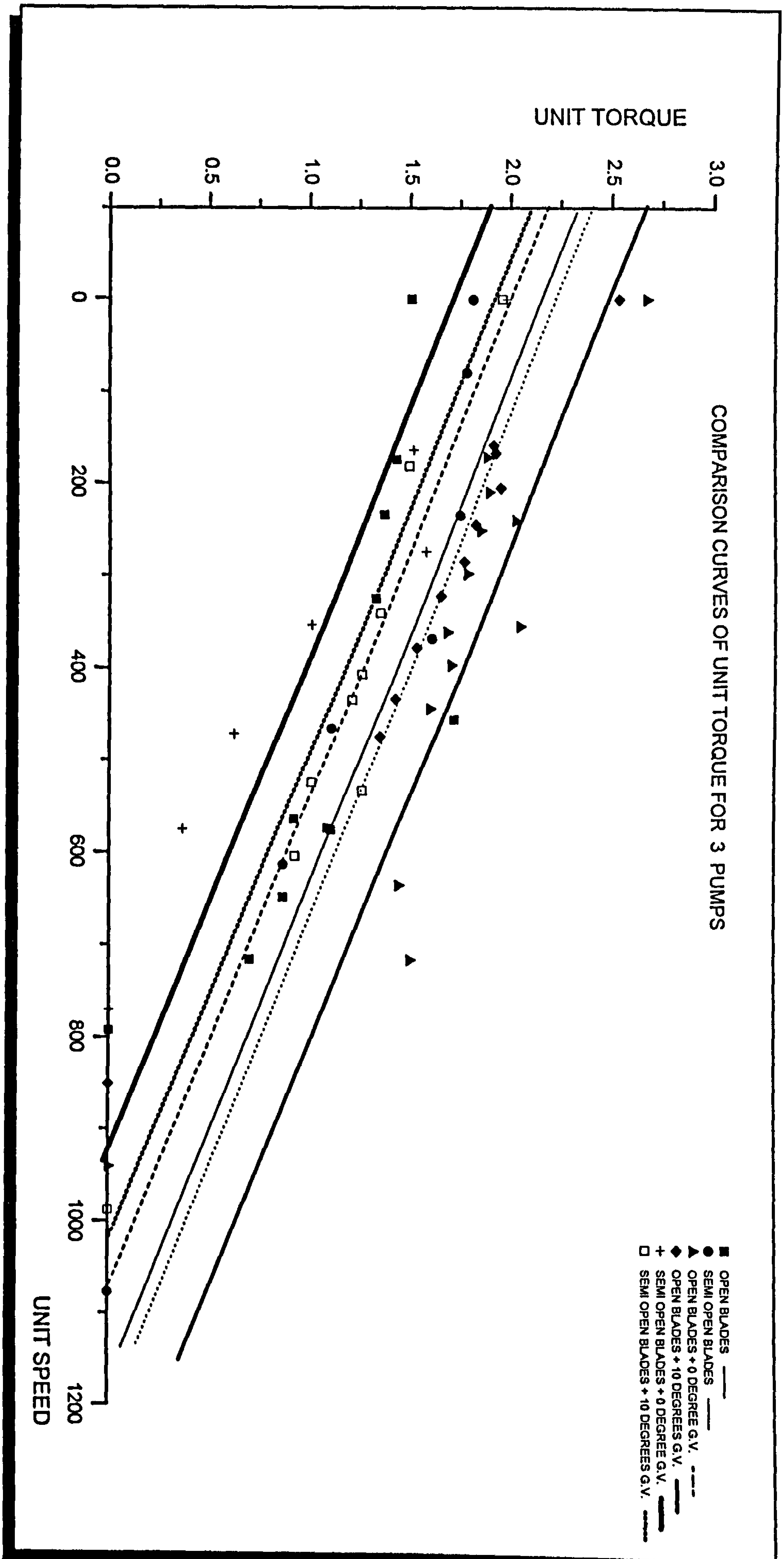

COMPARISON CURVES OF UNIT TORQUE FOR 3 PUMPS
UNIT TORQUE
3.0
2.5
2.0
1.5
1.0
0.5
0.0
0
200
400
600
800
1000
1200
UNIT SPEED
OPEN BLADES
SEMI OPEN BLADES
OPEN BLADES + 0 DEGREE G.V.
OPEN BLADES + 10 DEGREES G.V.
SEMI OPEN BLADES + 0 DEGREE G.V.
SEMI OPEN BLADES + 10 DEGREES G.V.

Figure ( 6.24 )

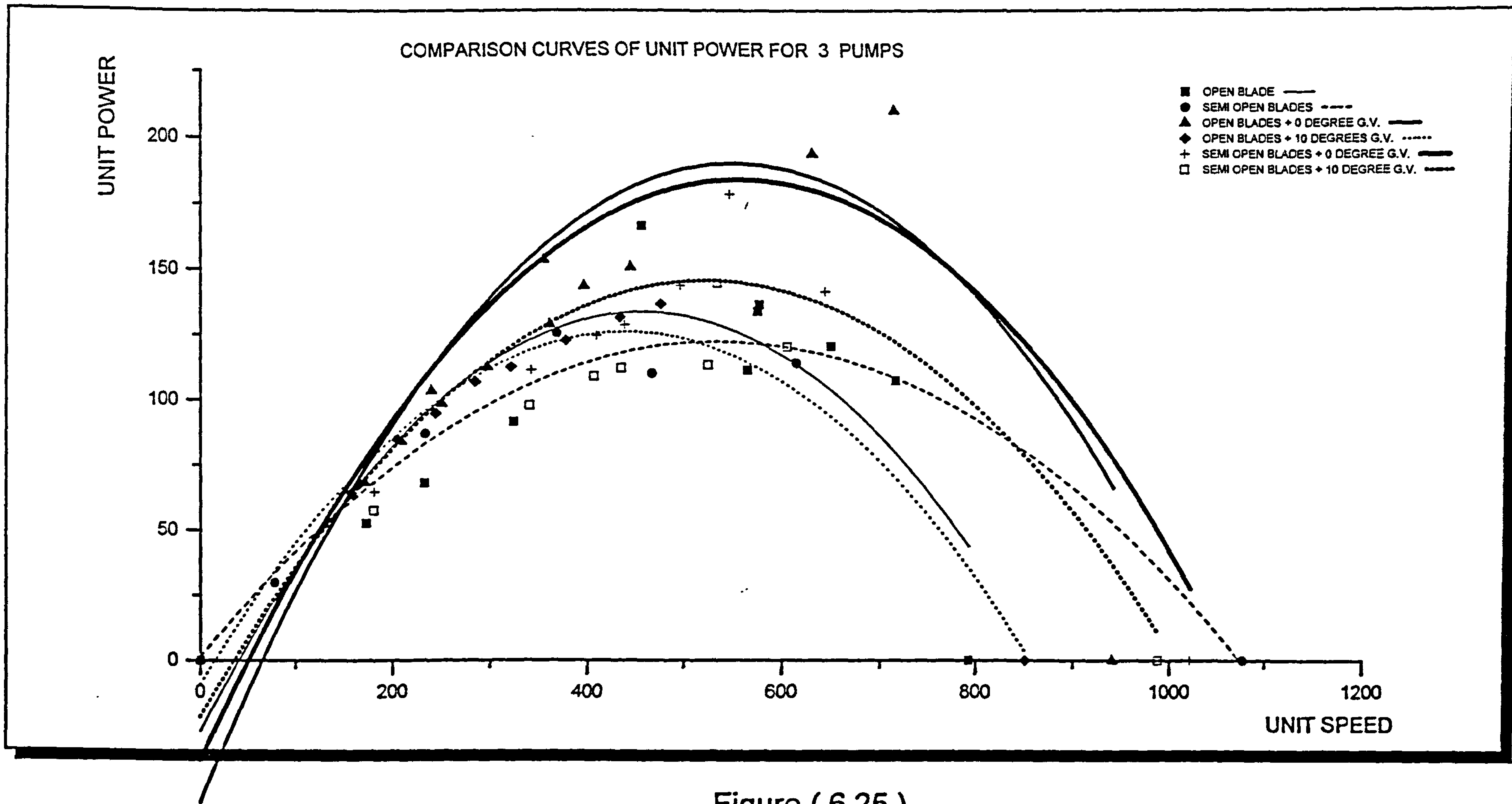

COMPARISON CURVES OF UNIT POWER FOR 3 PUMPS
UNIT POWER
0
50
100
150
200
0
200
400
600
800
1000
1200
UNIT SPEED
OPEN BLADE
SEMI OPEN BLADES
OPEN BLADES + 0 DEGREE G.V.
OPEN BLADES + 10 DEGREES G.V.
SEMI OPEN BLADES + 0 DEGREE G.V.
SEMI OPEN BLADES + 10 DEGREE G.V.

Figure ( 6.25 )

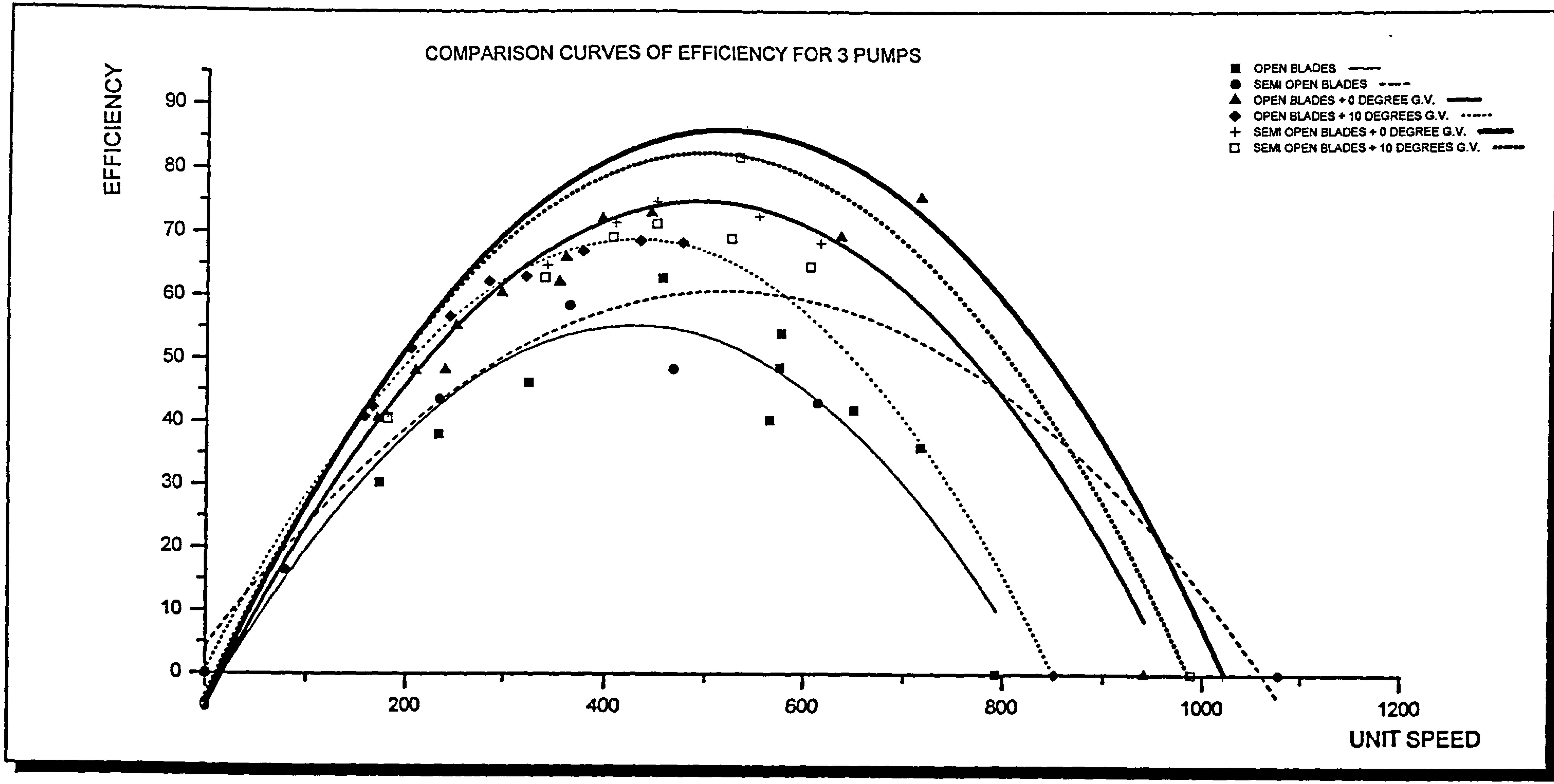
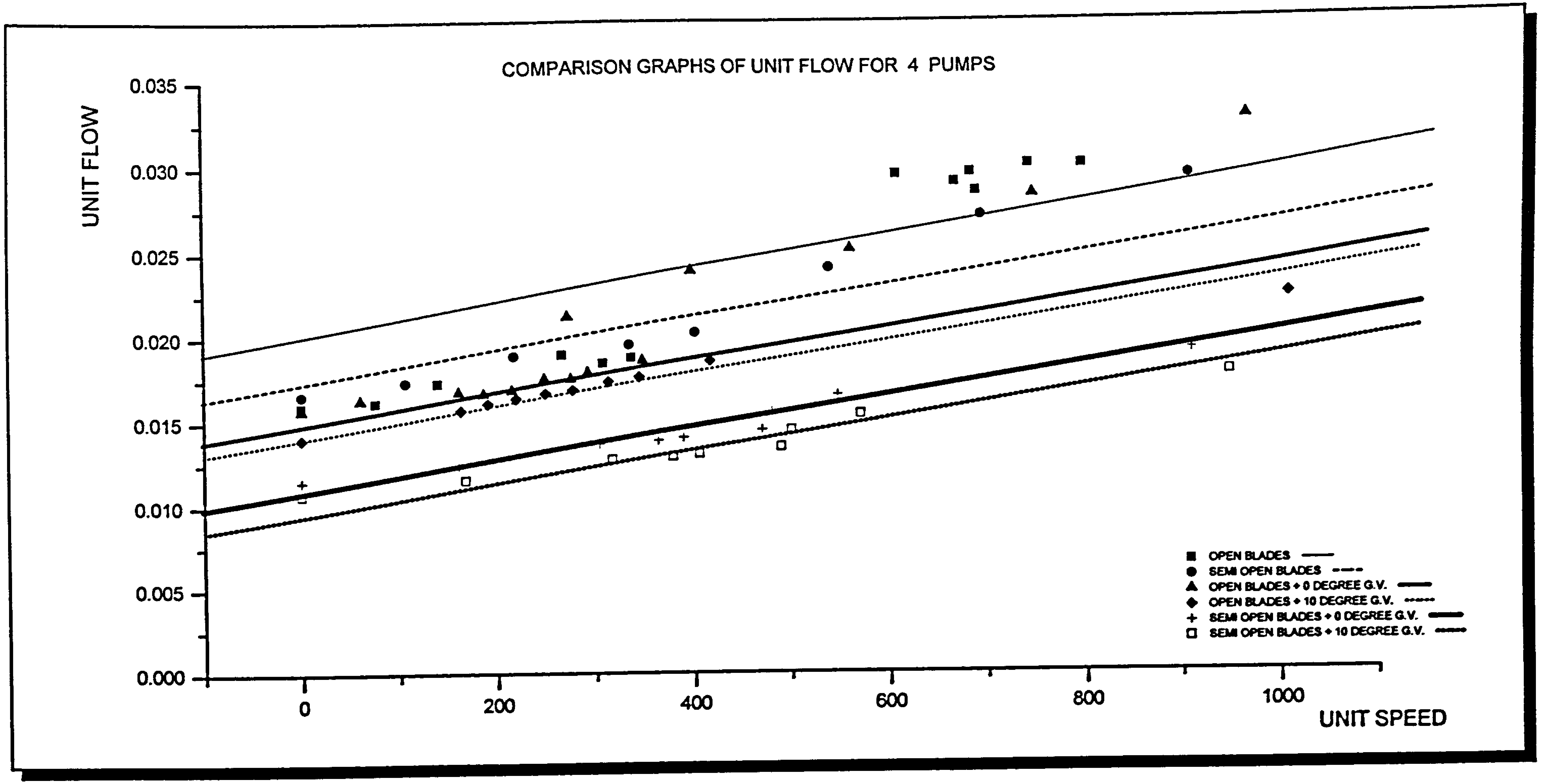

COMPARISON CURVES OF EFFICIENCY FOR 3 PUMPS
EFFICIENCY
0
10
20
30
40
50
60
70
80
90
200
400
600
800
1000
1200
UNIT SPEED
OPEN BLADES
SEMI OPEN BLADES
OPEN BLADES + 0 DEGREE G.V.
OPEN BLADES + 10 DEGREES G.V.
SEMI OPEN BLADES + 0 DEGREE G.V.
SEMI OPEN BLADES + 10 DEGREES G.V.

Figure ( 6.26 )



COMPARISON GRAPHS OF UNIT FLOW FOR 4 PUMPS
UNIT FLOW
0.035
0.030
0.025
0.020
0.015
0.010
0.005
0.000
0
200
400
600
800
1000
UNIT SPEED
OPEN BLADES
SEMI OPEN BLADES
OPEN BLADES + 0 DEGREE G.V.
OPEN BLADES + 10 DEGREE G.V.
SEMI OPEN BLADES + 0 DEGREE G.V.
SEMI OPEN BLADES + 10 DEGREE G.V.


Figure ( 6.27 )

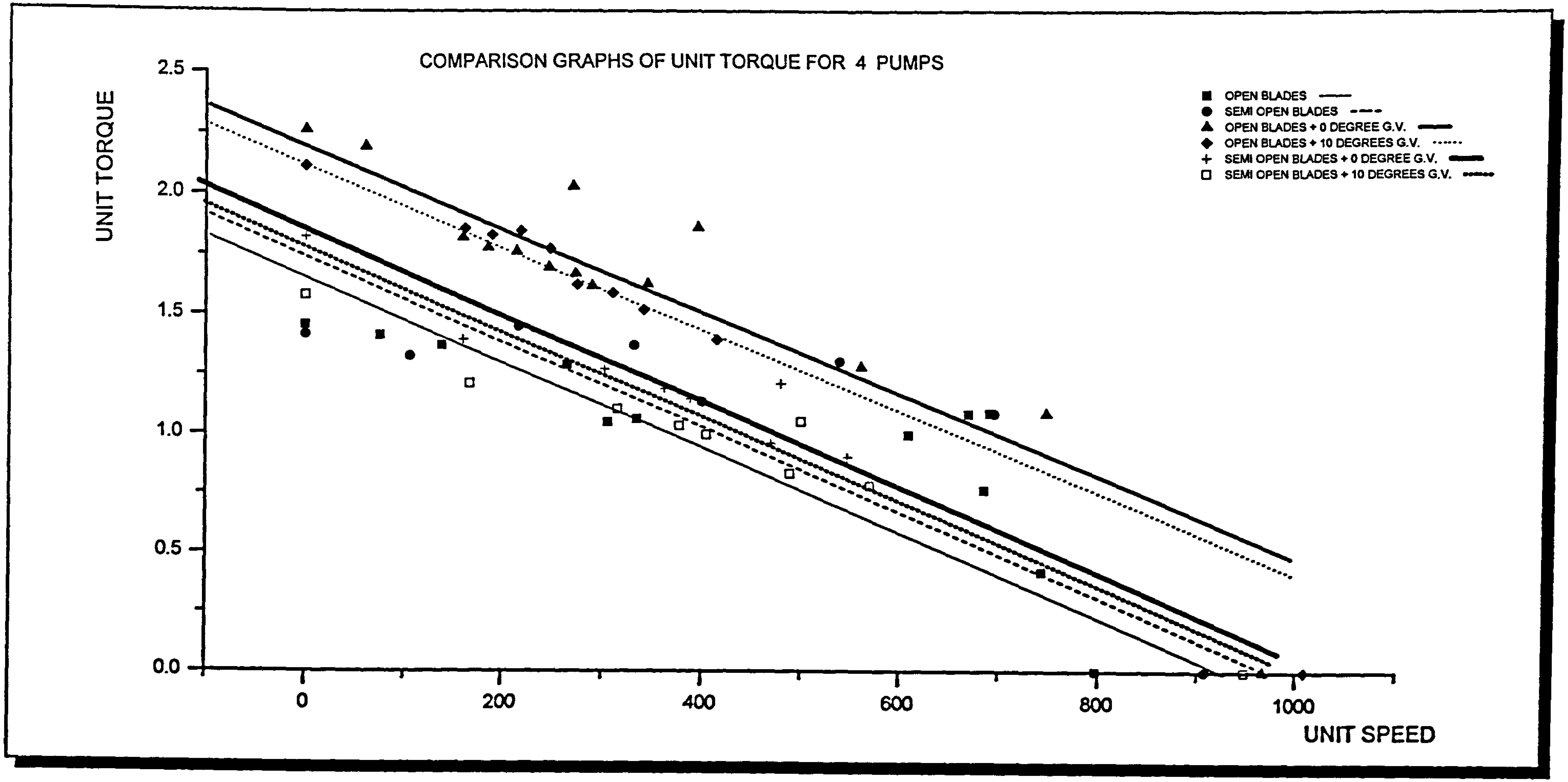

COMPARISON GRAPHS OF UNIT TORQUE FOR 4 PUMPS
UNIT TORQUE
UNIT SPEED
2.5
2.0
1.5
1.0
0.5
0.0
0
200
400
600
800
1000
OPEN BLADES
SEMI OPEN BLADES
OPEN BLADES + 0 DEGREE G.V.
OPEN BLADES + 10 DEGREES G.V.
SEMI OPEN BLADES + 0 DEGREE G.V.
SEMI OPEN BLADES + 10 DEGREES G.V.

Figure ( 6.28 )

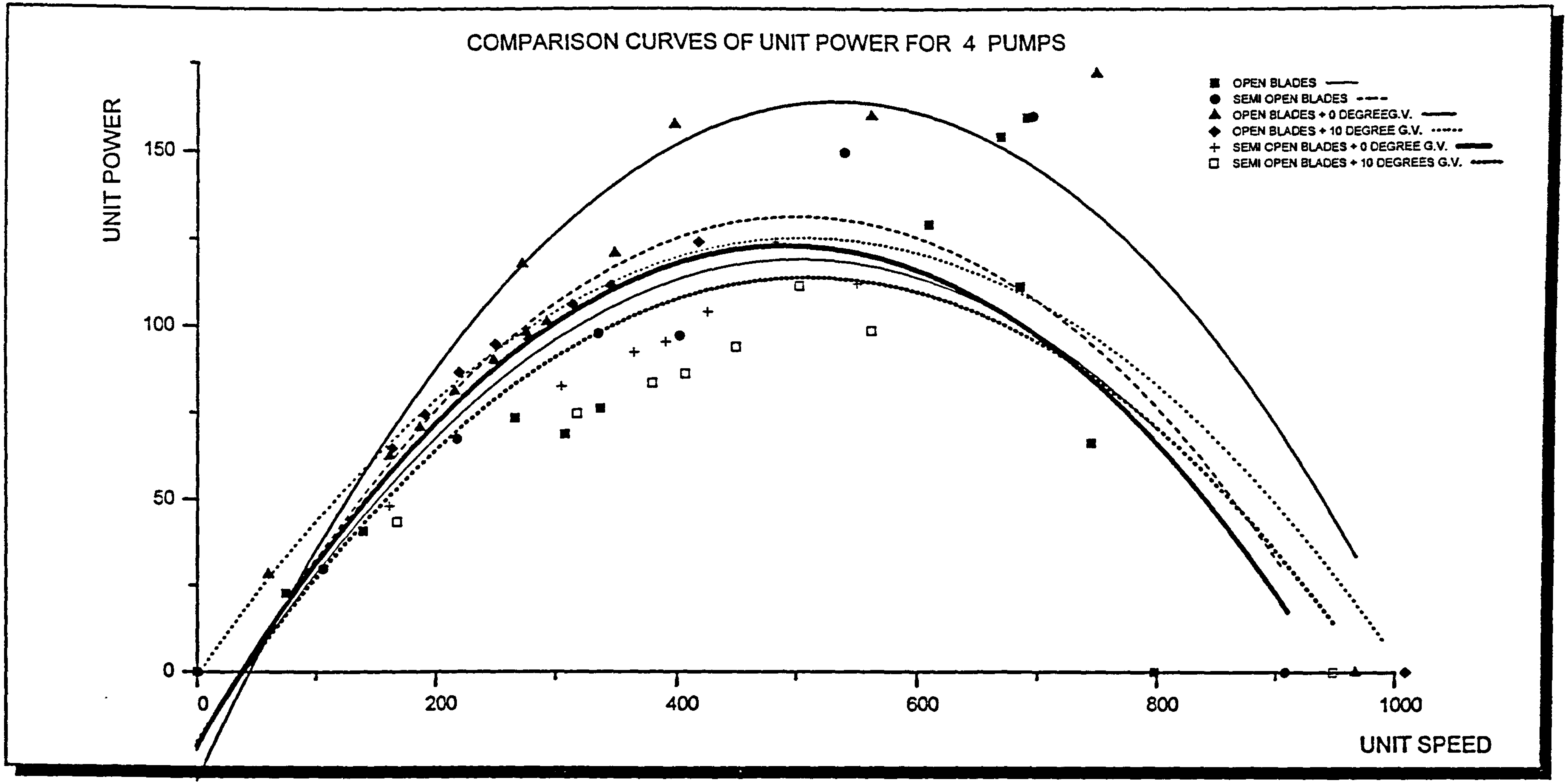

COMPARISON CURVES OF UNIT POWER FOR 4 PUMPS
UNIT POWER
UNIT SPEED
0
50
100
150
0
200
400
600
800
1000
OPEN BLADES
SEMI OPEN BLADES
OPEN BLADES + 0 DEGREEG.V.
OPEN BLADES + 10 DEGREE G.V.
SEMI OPEN BLADES + 0 DEGREE G.V.
SEMI OPEN BLADES + 10 DEGREES G.V.

Figure ( 6.29 )

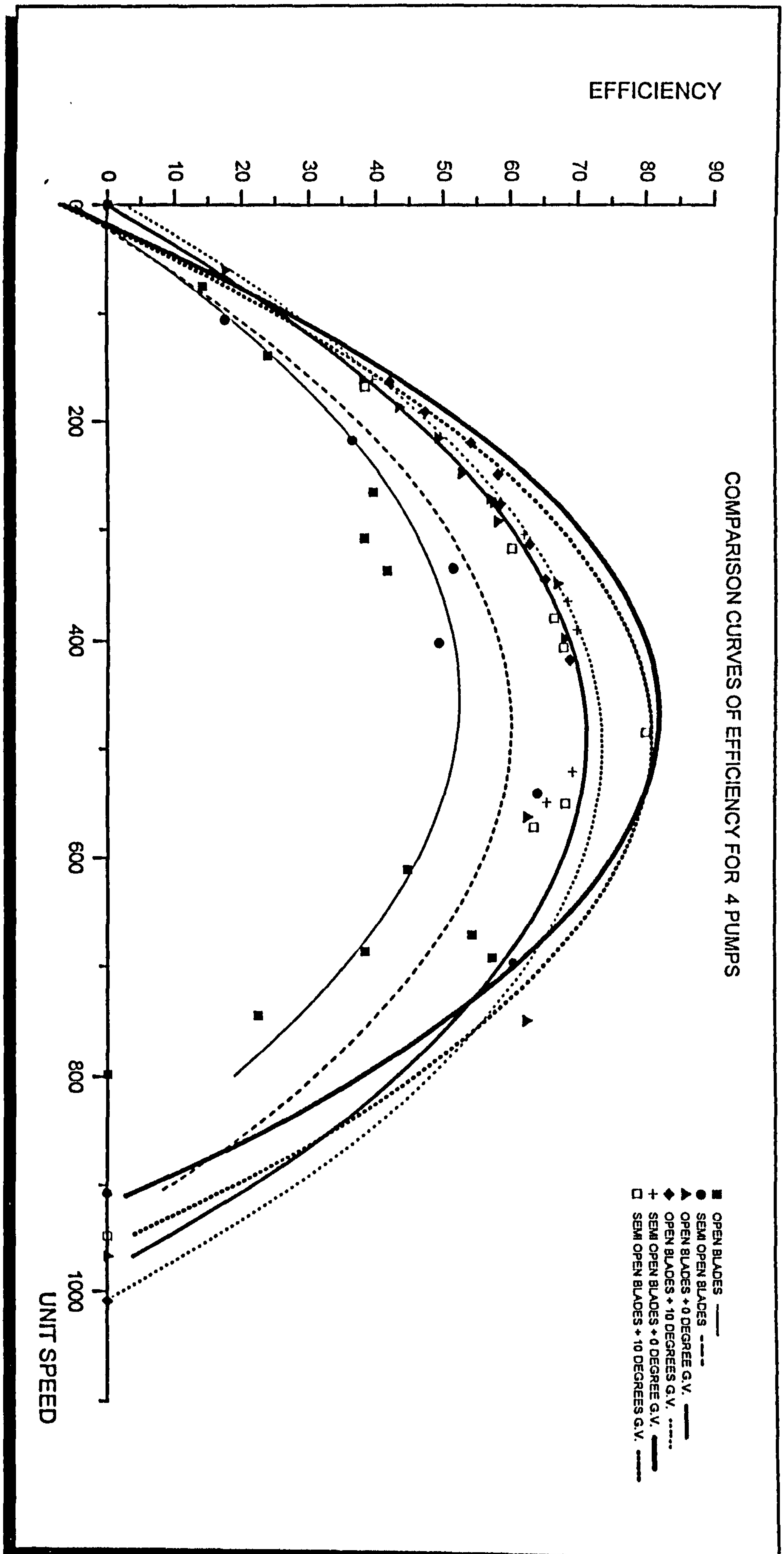

COMPARISON CURVES OF EFFICIENCY FOR 4 PUMPS
EFFICIENCY
0
10
20
30
40
50
60
70
80
90
0
200
400
600
800
1000
UNIT SPEED
OPEN BLADES
SEMI OPEN BLADES
OPEN BLADES + 0 DEGREE G.V.
OPEN BLADES + 10 DEGREES G.V.
SEMI OPEN BLADES + 0 DEGREE G.V.
SEMI OPEN BLADES + 10 DEGREES G.V.

Figure ( 6.30 )

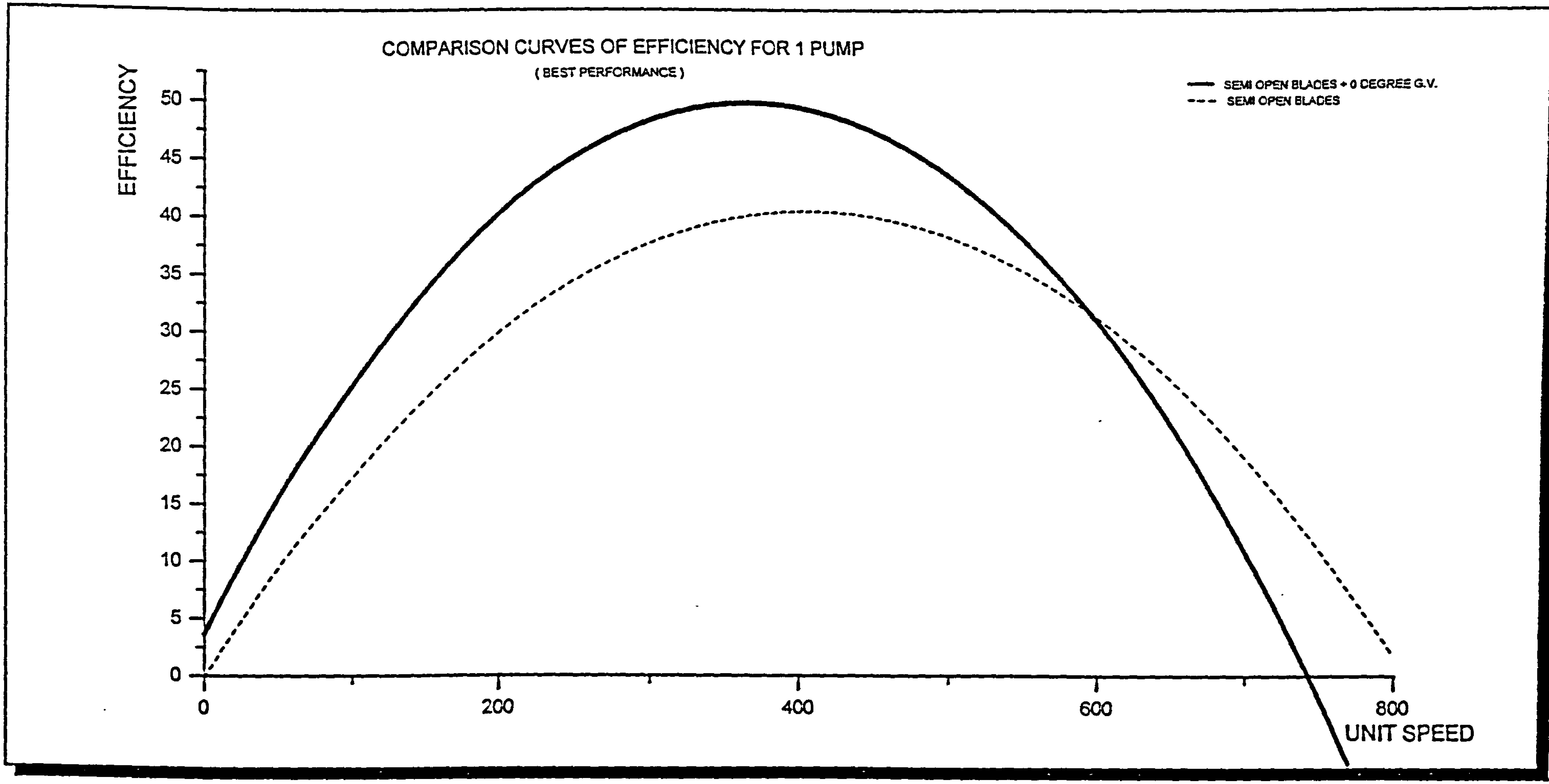

COMPARISON CURVES OF EFFICIENCY FOR 1 PUMP
( BEST PERFORMANCE )
SEMI OPEN BLADES + 0 DEGREE G.V.
SEMI OPEN BLADES
EFFICIENCY
50
45
40
35
30
25
20
15
10
5
0
0
200
400
600
800
UNIT SPEED

Figure ( 6.3 1)

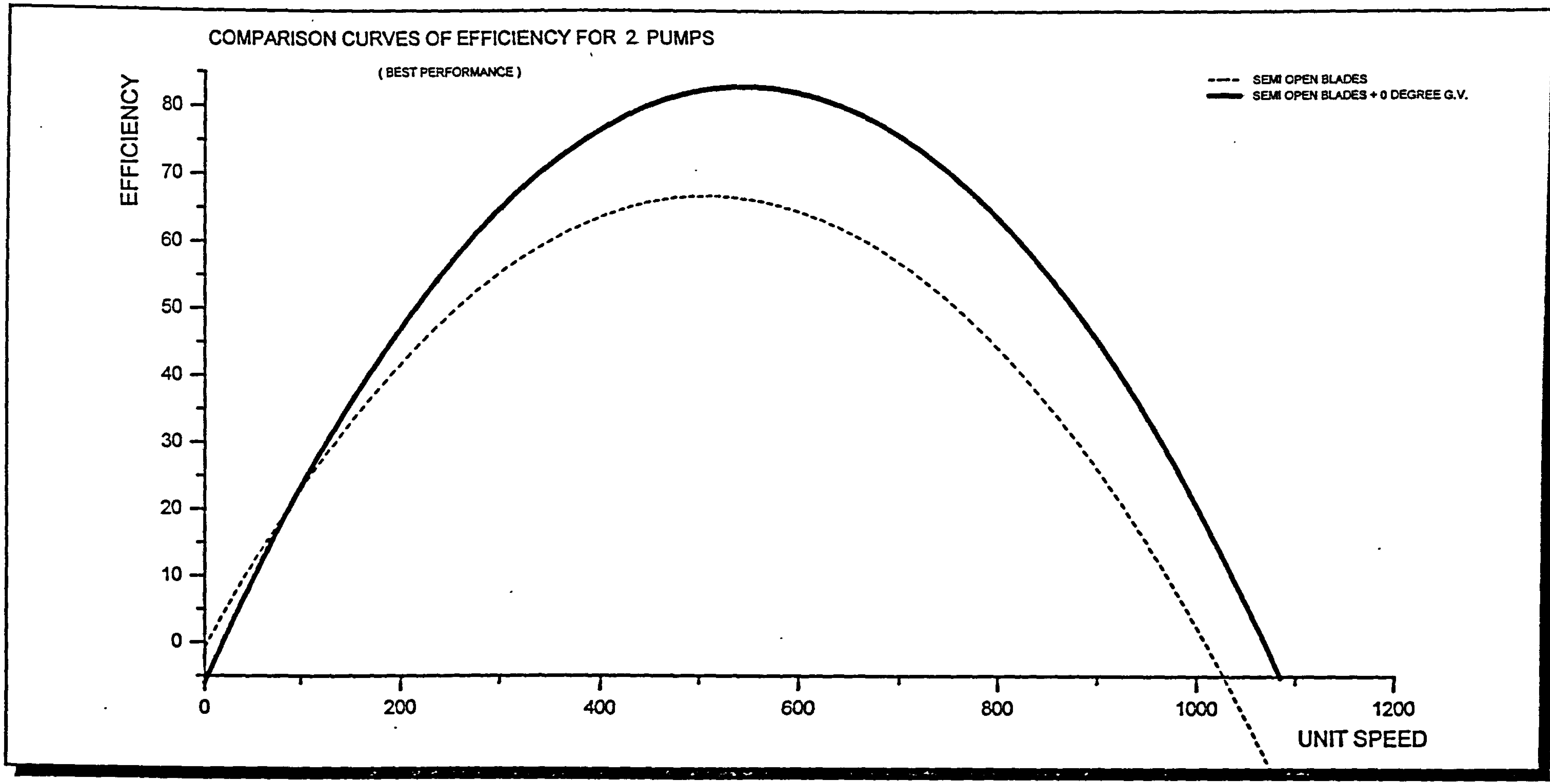

COMPARISON CURVES OF EFFICIENCY FOR 2 PUMPS
( BEST PERFORMANCE )
SEMI OPEN BLADES
SEMI OPEN BLADES + 0 DEGREE G.V.
EFFICIENCY
80
70
60
50
40
30
20
10
0
0
200
400
600
800
1000
1200
UNIT SPEED

Figure ( 6.32 )

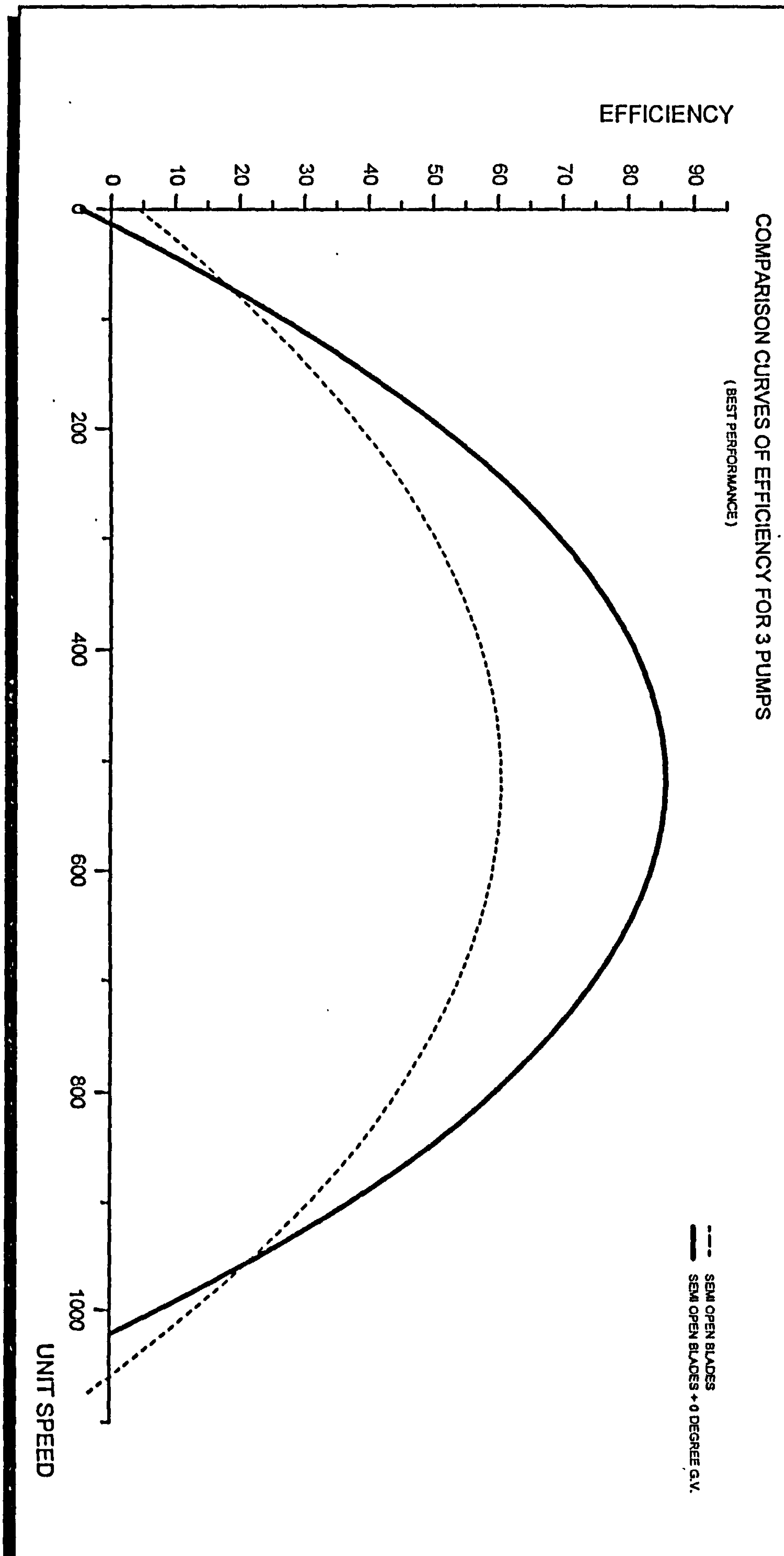

COMPARISON CURVES OF EFFICIENCY FOR 3 PUMPS
( BEST PERFORMANCE )
EFFICIENCY
0
10
20
30
40
50
60
70
80
90
UNIT SPEED
0
200
400
600
800
1000
SEMI OPEN BLADES
SEMI OPEN BLADES + 0 DEGREE G.V.

Figure ( 6.33 )

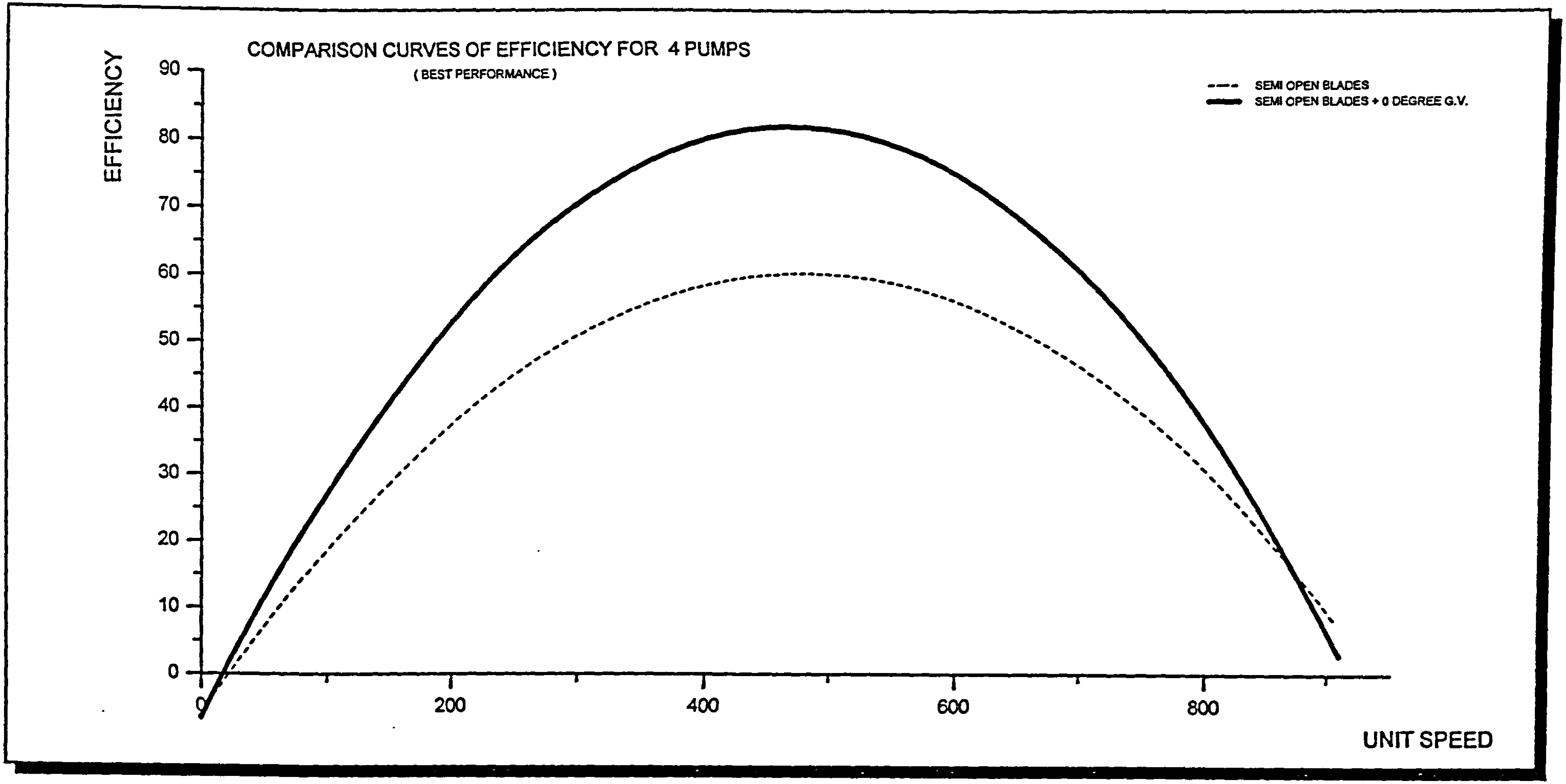

COMPARISON CURVES OF EFFICIENCY FOR 4 PUMPS
( BEST PERFORMANCE )
EFFICIENCY
90
80
70
60
50
40
30
20
10
0
0
200
400
600
800
UNIT SPEED
SEMI OPEN BLADES
SEMI OPEN BLADES + 0 DEGREE G.V.

Figure ( 6.34 )

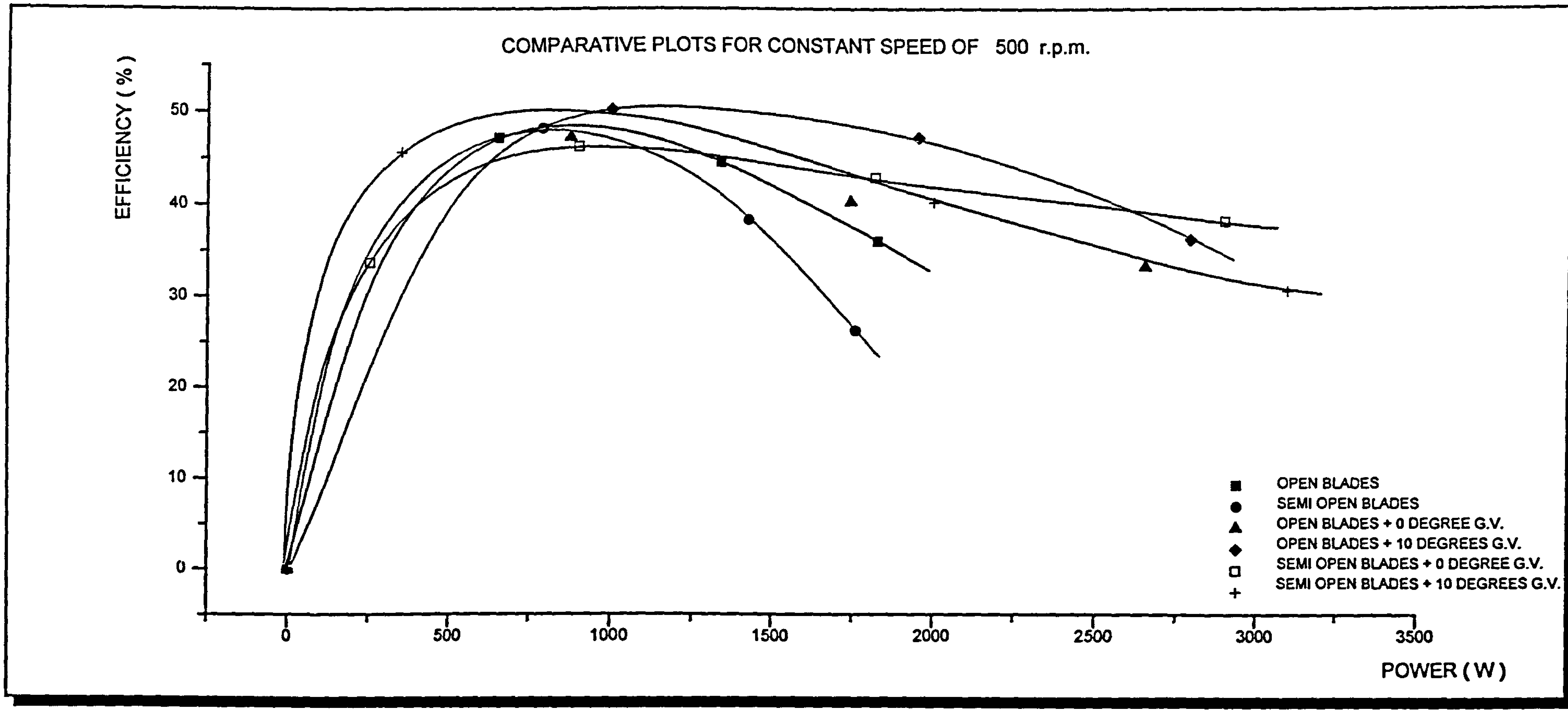

COMPARATIVE PLOTS FOR CONSTANT SPEED OF 500 r.p.m.
EFFICIENCY ( % )
0
10
20
30
40
50
0
500
1000
1500
2000
2500
3000
3500
POWER ( W )
OPEN BLADES
SEMI OPEN BLADES
OPEN BLADES + 0 DEGREE G.V.
OPEN BLADES + 10 DEGREES G.V.
SEMI OPEN BLADES + 0 DEGREE G.V.
SEMI OPEN BLADES + 10 DEGREES G.V.

Figure ( 6.35 )

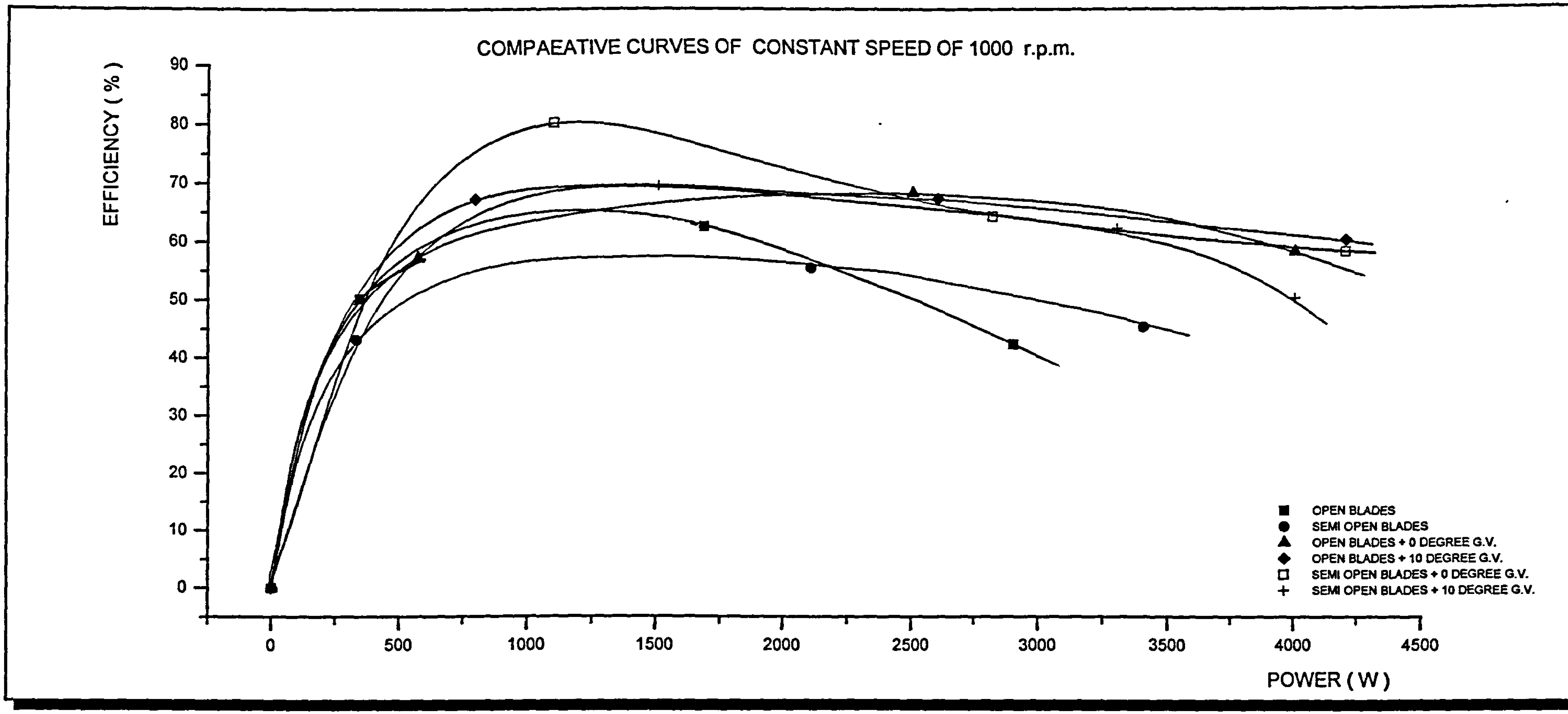

COMPAEATIVE CURVES OF CONSTANT SPEED OF 1000 r.p.m.
EFFICIENCY ( % )
0
10
20
30
40
50
60
70
80
90
0
500
1000
1500
2000
2500
3000
3500
4000
4500
POWER ( W )
OPEN BLADES
SEMI OPEN BLADES
OPEN BLADES + 0 DEGREE G.V.
OPEN BLADES + 10 DEGREE G.V.
SEMI OPEN BLADES + 0 DEGREE G.V.
SEMI OPEN BLADES + 10 DEGREE G.V.

Figure ( 6.36 )

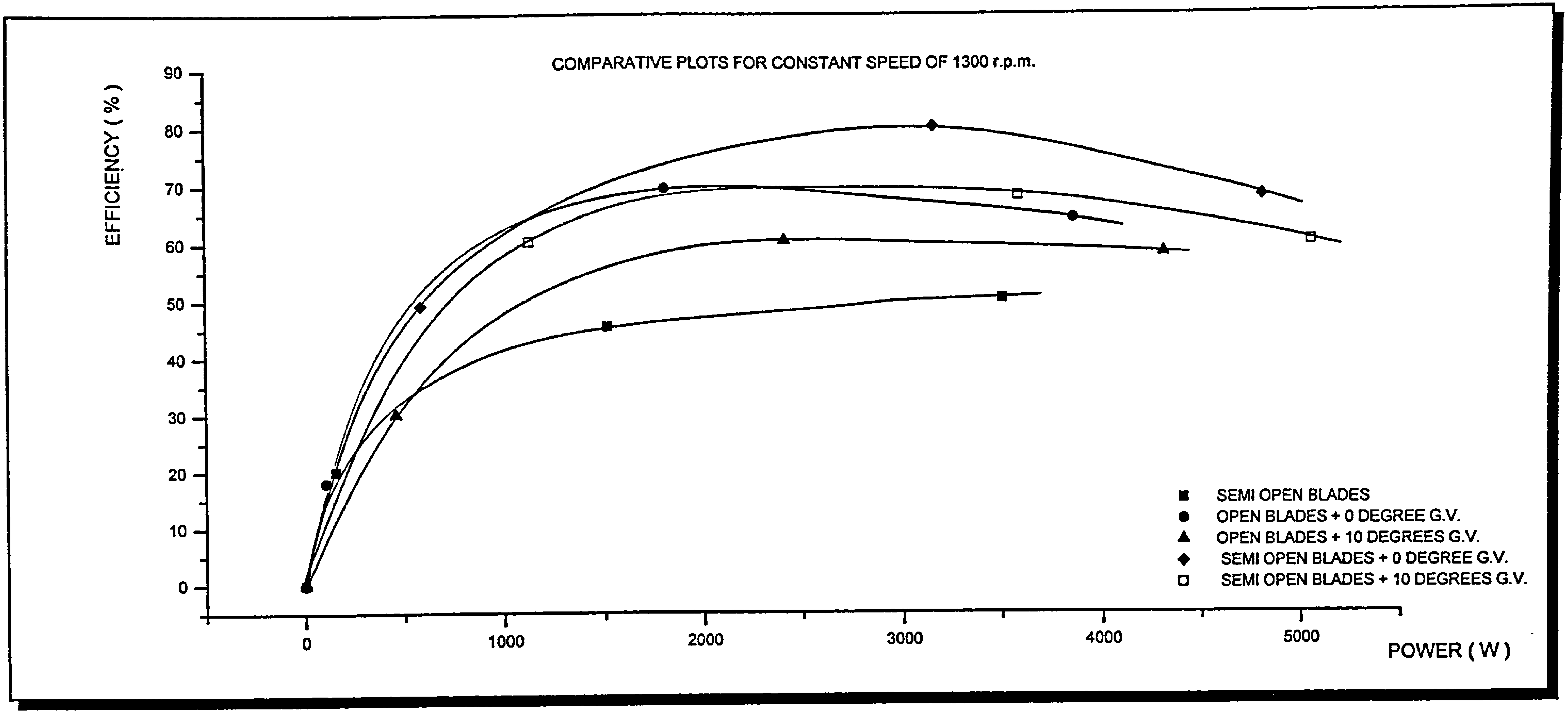
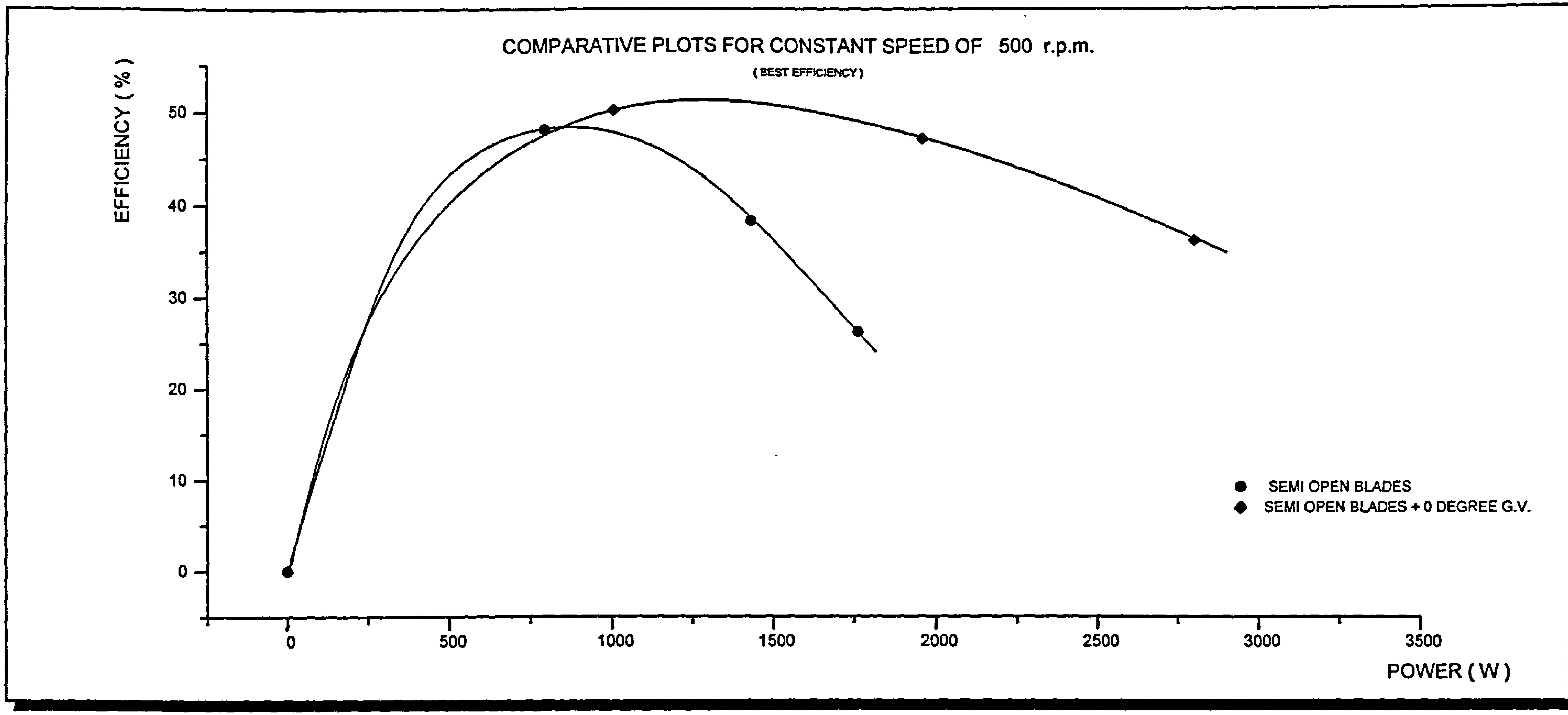

COMPARATIVE PLOTS FOR CONSTANT SPEED OF 1300 r.p.m.
EFFICIENCY ( % )
90
80
70
60
50
40
30
20
10
0
0
1000
2000
3000
4000
5000
POWER ( W )
SEMI OPEN BLADES
OPEN BLADES + 0 DEGREE G.V.
OPEN BLADES + 10 DEGREES G.V.
SEMI OPEN BLADES + 0 DEGREE G.V.
SEMI OPEN BLADES + 10 DEGREES G.V.

Figure ( 6.37 )



COMPARATIVE PLOTS FOR CONSTANT SPEED OF 500 r.p.m.
( BEST EFFICIENCY )
EFFICIENCY ( % )
0
10
20
30
40
50
0
500
1000
1500
2000
2500
3000
3500
POWER ( W )
SEMI OPEN BLADES
SEMI OPEN BLADES + 0 DEGREE G.V.


Figure ( 6.38 )

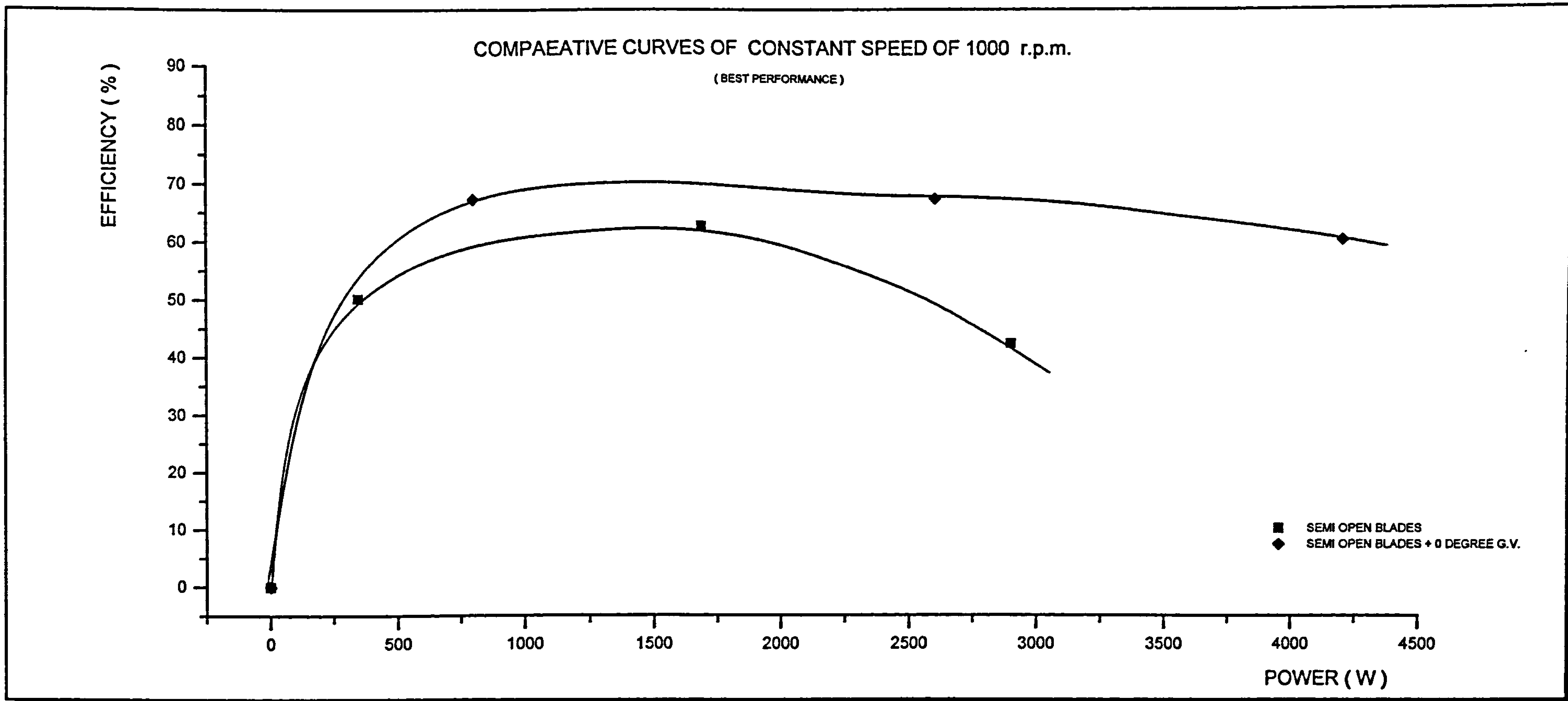

COMPAEATIVE CURVES OF CONSTANT SPEED OF 1000 r.p.m.
( BEST PERFORMANCE )
EFFICIENCY ( % )
90
80
70
60
50
40
30
20
10
0
0
500
1000
1500
2000
2500
3000
3500
4000
4500
POWER ( W )
SEMI OPEN BLADES
SEMI OPEN BLADES + 0 DEGREE G.V.

Figure ( 6.39 )

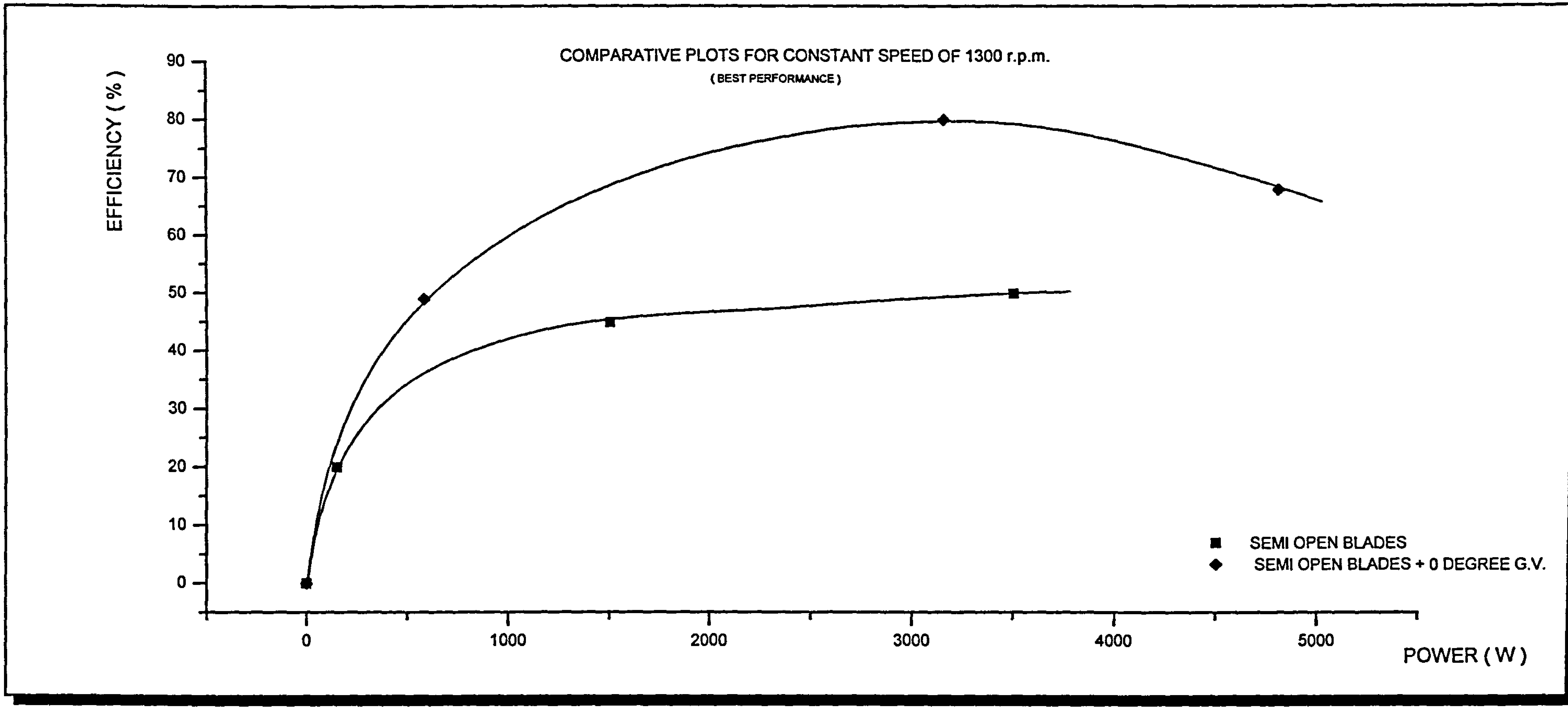

COMPARATIVE PLOTS FOR CONSTANT SPEED OF 1300 r.p.m.
( BEST PERFORMANCE )
EFFICIENCY ( % )
90
80
70
60
50
40
30
20
10
0
0
1000
2000
3000
4000
5000
POWER ( W )
SEMI OPEN BLADES
SEMI OPEN BLADES + 0 DEGREE G.V.

Figure ( 6.40 )

COMPARISON CURVES FOR CONSTANT SPEED OF 500 r.p.m.
EFFICIENCY ( % )
50
40
30
20
10
0
0
20
40
60
80
100
PERCENTAGE OF LOAD
OPEN BLADES
SEMI OPEN BLADES
OPEN BLADES + 0 DEGREE G.V.
OPEN BLADES + 10 DEGREES G.V.
SEMI OPEN BLADES + 0 DEGREE G.V.
SEMI OPEN BLADES + 10 DEGREES G.V.


Figure ( 6.41 )

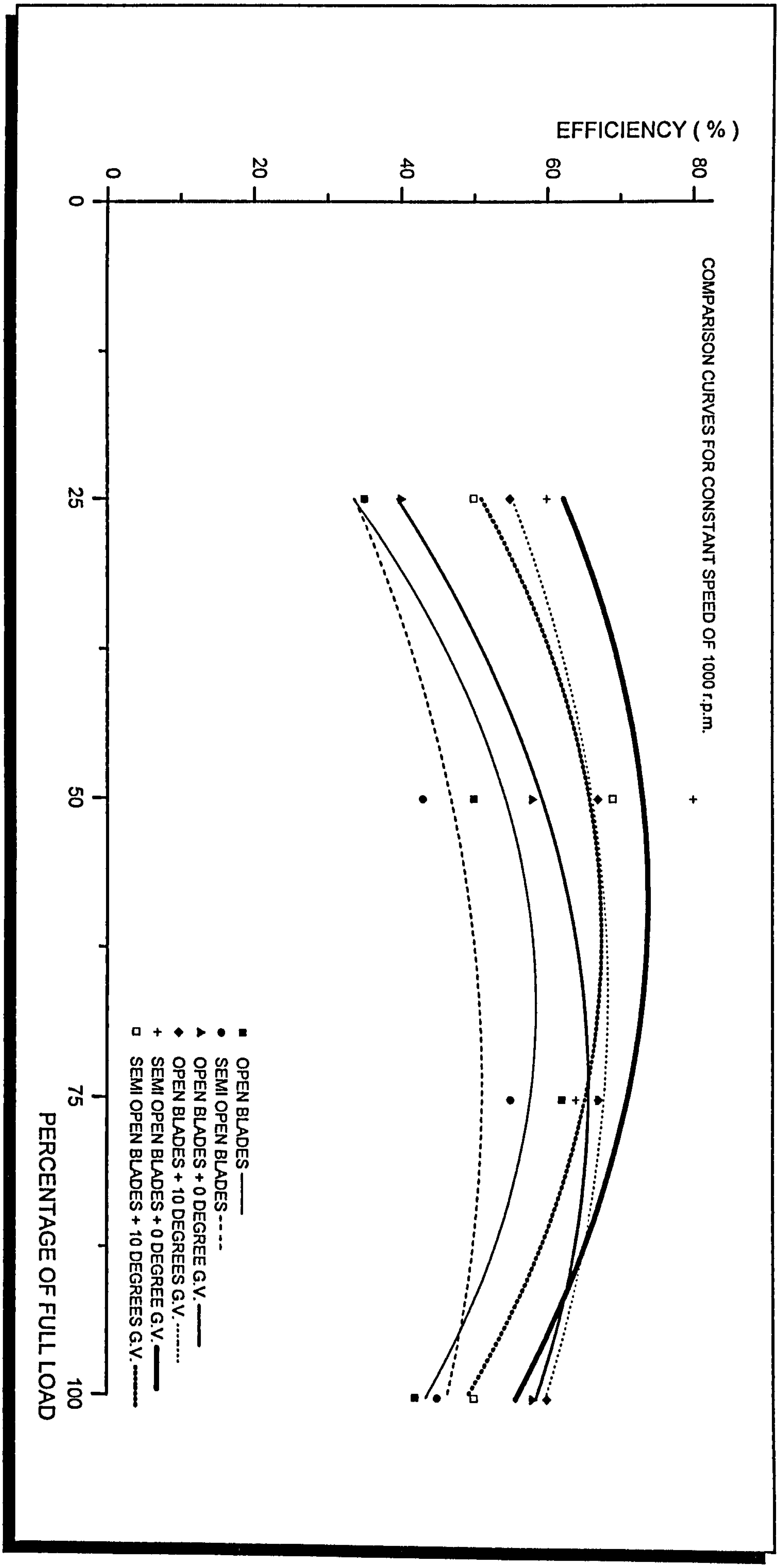

EFFICIENCY ( % )
80
60
40
20
0
0
25
50
75
100
PERCENTAGE OF FULL LOAD
COMPARISON CURVES FOR CONSTANT SPEED OF 1000 r.p.m.
OPEN BLADES
SEMI OPEN BLADES
OPEN BLADES + 0 DEGREE G.V.
OPEN BLADES + 10 DEGREES G.V.
SEMI OPEN BLADES + 0 DEGREE G.V.
SEMI OPEN BLADES + 10 DEGREES G.V.

Figure ( 6.42 )

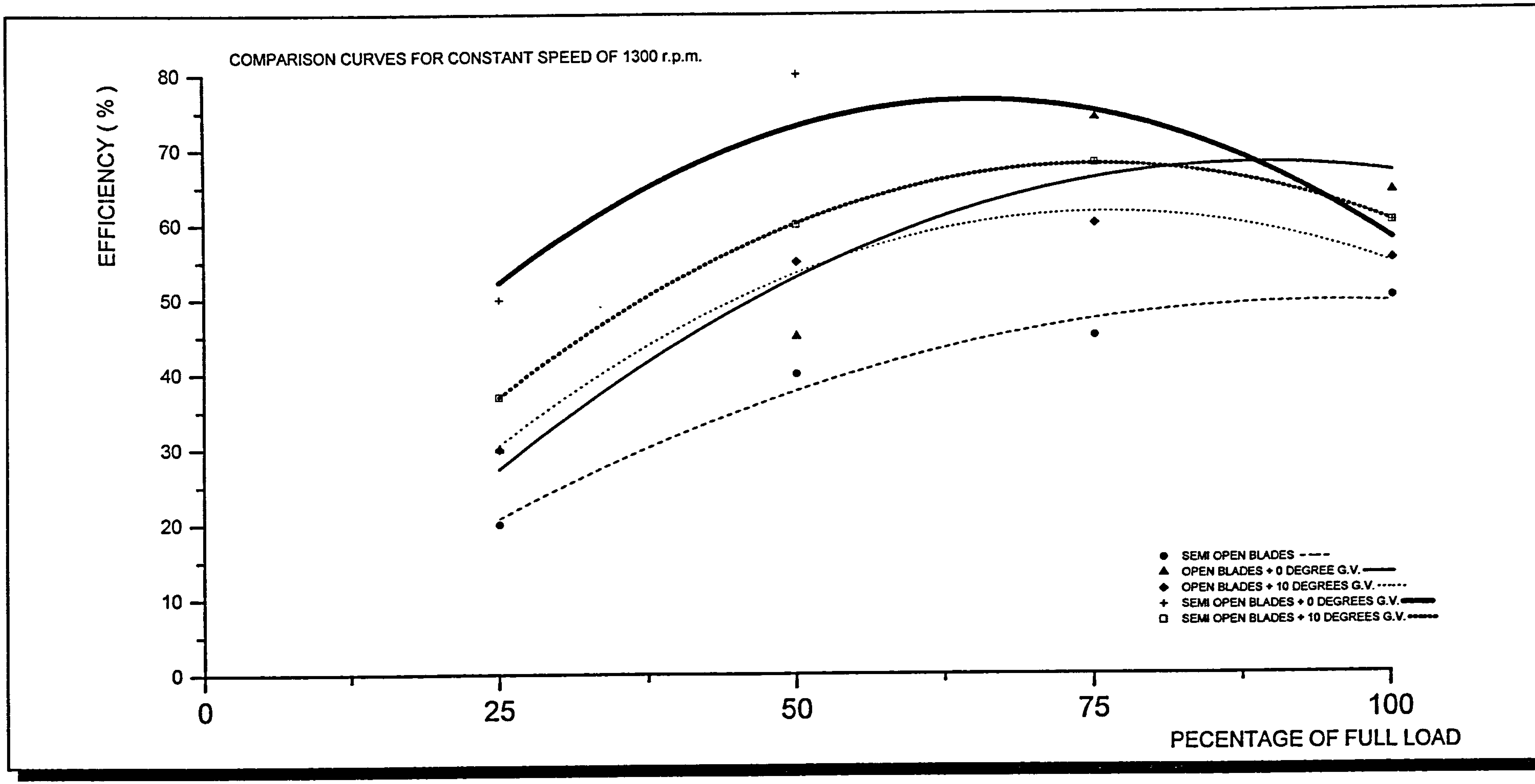

COMPARISON CURVES FOR CONSTANT SPEED OF 1300 r.p.m.
EFFICIENCY ( % )
80
70
60
50
40
30
20
10
0
0
25
50
75
100
PECENTAGE OF FULL LOAD
SEMI OPEN BLADES
OPEN BLADES + 0 DEGREE G.V.
OPEN BLADES + 10 DEGREES G.V.
SEMI OPEN BLADES + 0 DEGREES G.V.
SEMI OPEN BLADES + 10 DEGREES G.V.

Figure ( 6.43 )

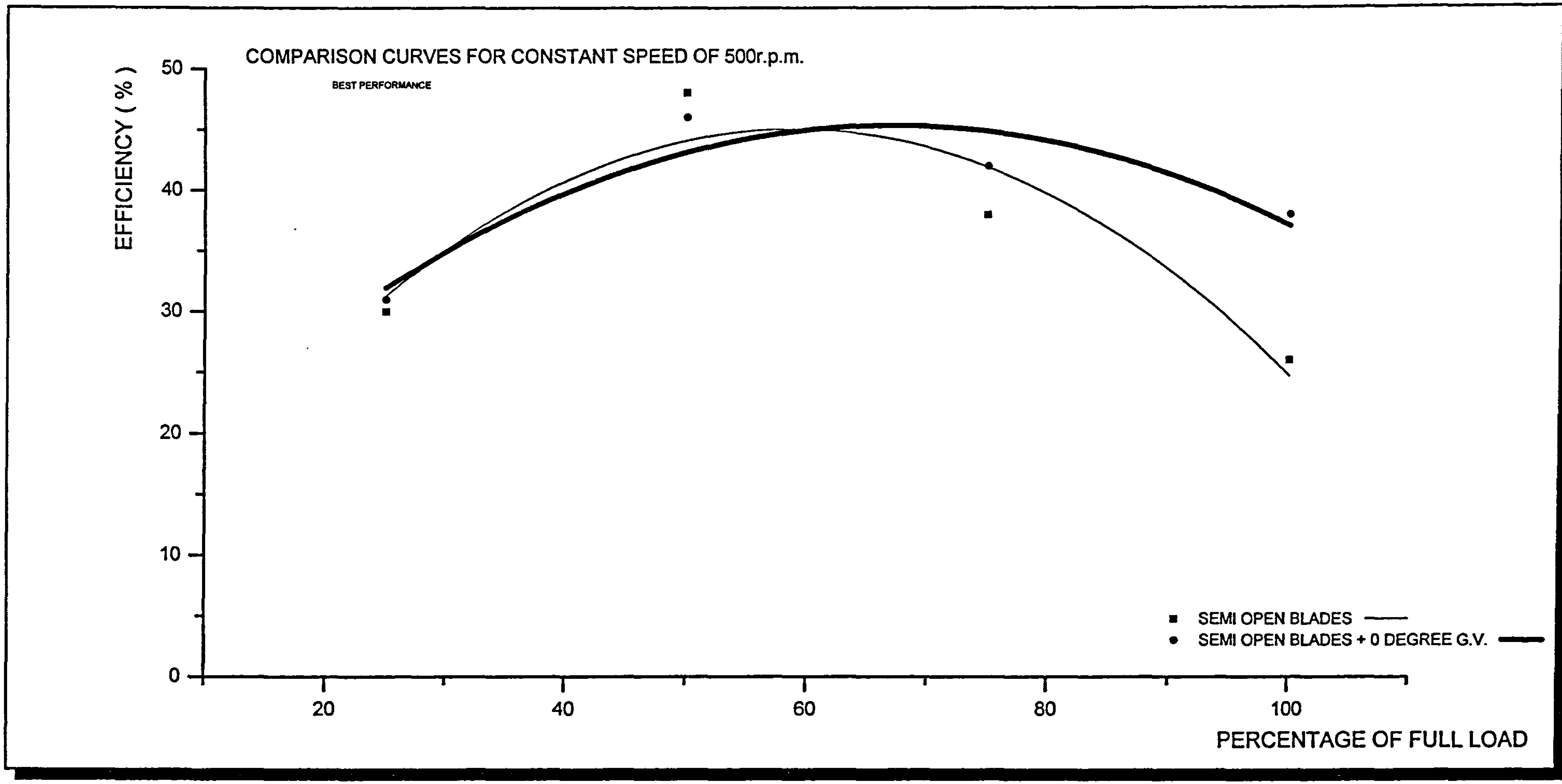

COMPARISON CURVES FOR CONSTANT SPEED OF 500r.p.m.
BEST PERFORMANCE
EFFICIENCY ( % )
50
40
30
20
10
0
20
40
60
80
100
PERCENTAGE OF FULL LOAD
SEMI OPEN BLADES
SEMI OPEN BLADES + 0 DEGREE G.V.

Figure ( 6.44 )

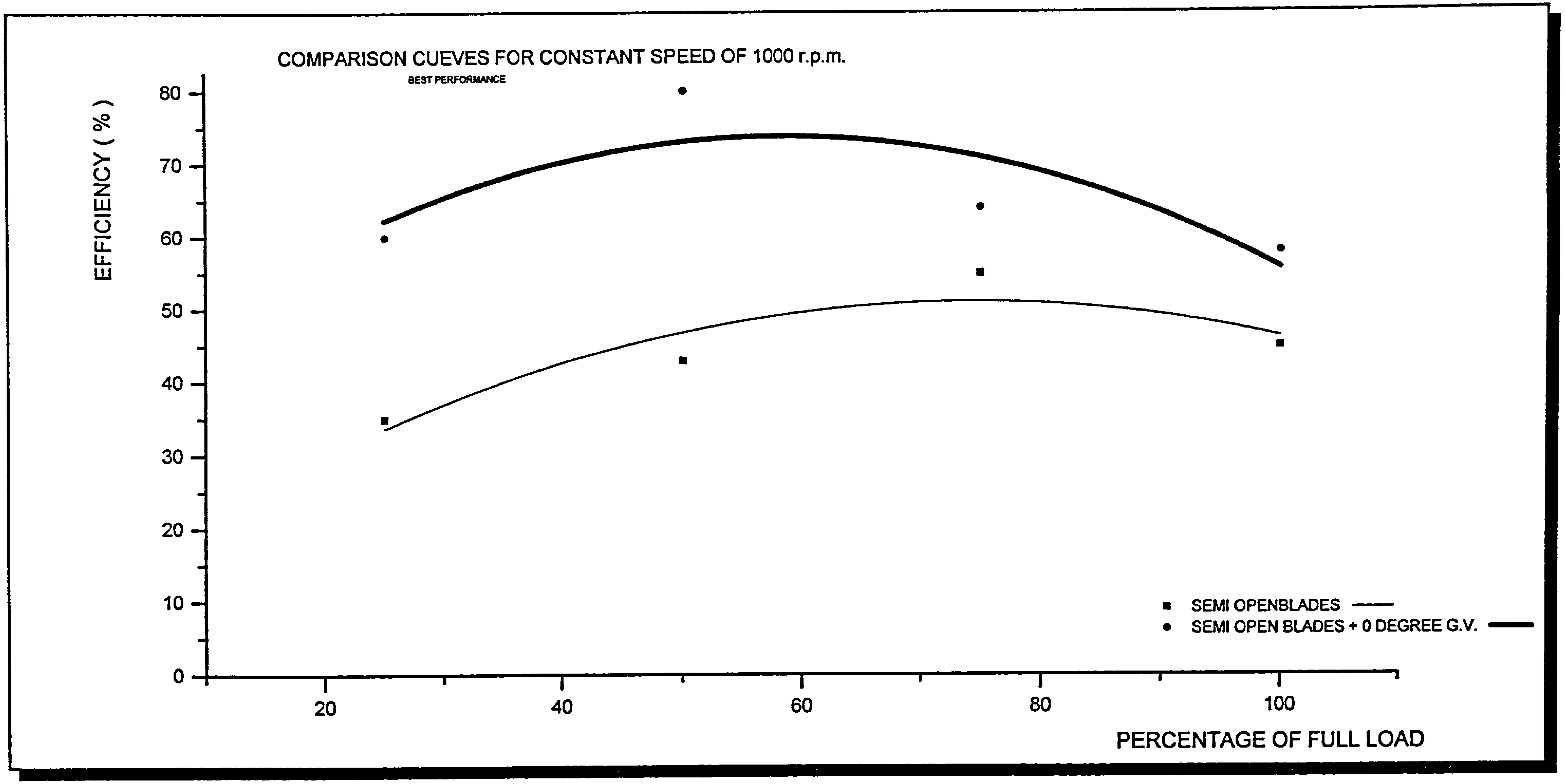
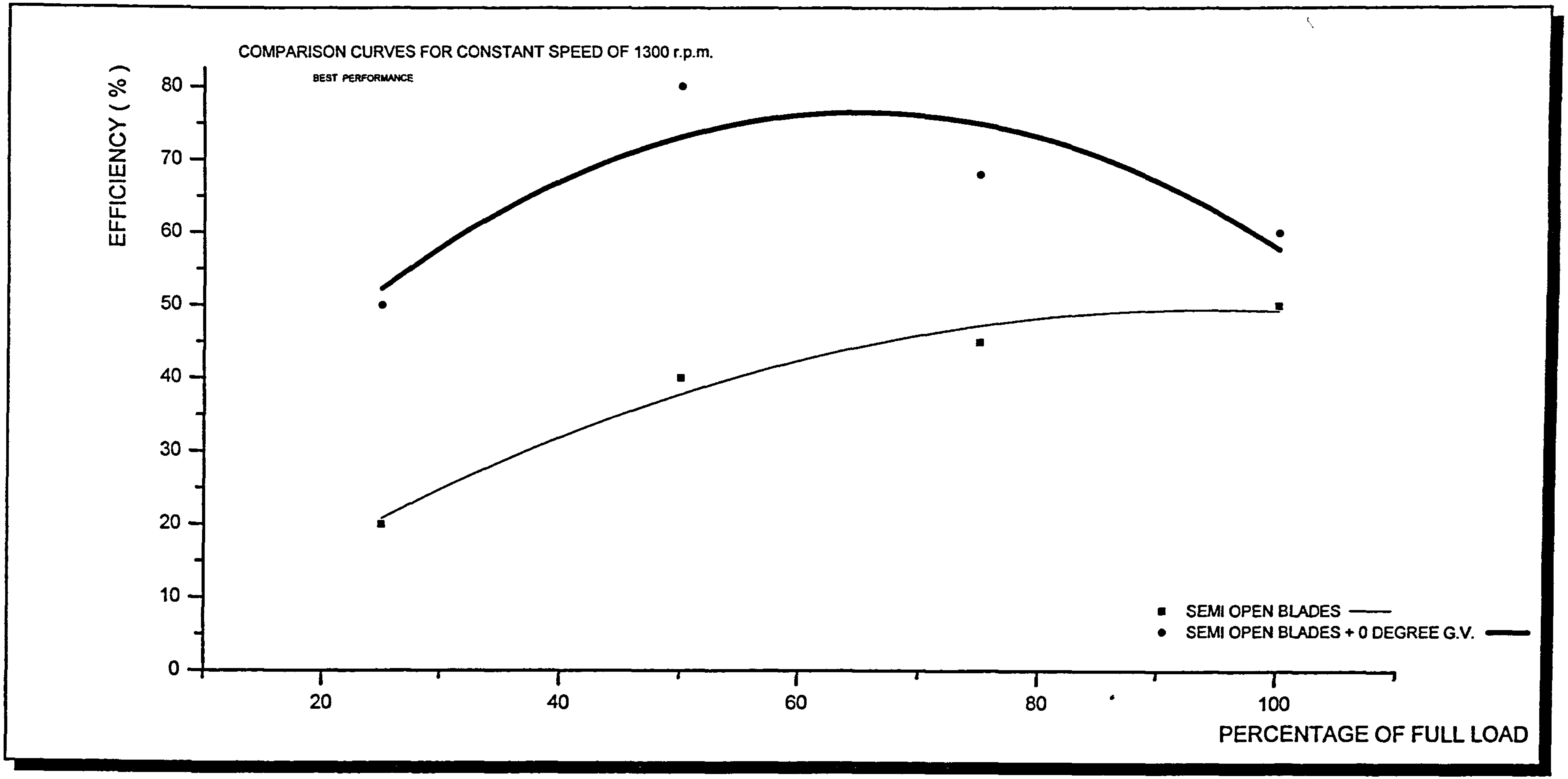

COMPARISON CUEVES FOR CONSTANT SPEED OF 1000 r.p.m.
BEST PERFORMANCE
EFFICIENCY ( % )
0
10
20
30
40
50
60
70
80
20
40
60
80
100
PERCENTAGE OF FULL LOAD
SEMI OPENBLADES
SEMI OPEN BLADES + 0 DEGREE G.V.

Figure ( 6.45 )



COMPARISON CURVES FOR CONSTANT SPEED OF 1300 r.p.m.
BEST PERFORMANCE
EFFICIENCY ( % )
0
10
20
30
40
50
60
70
80
20
40
60
80
100
PERCENTAGE OF FULL LOAD
SEMI OPEN BLADES
SEMI OPEN BLADES + 0 DEGREE G.V.


Figure ( 6.46 )

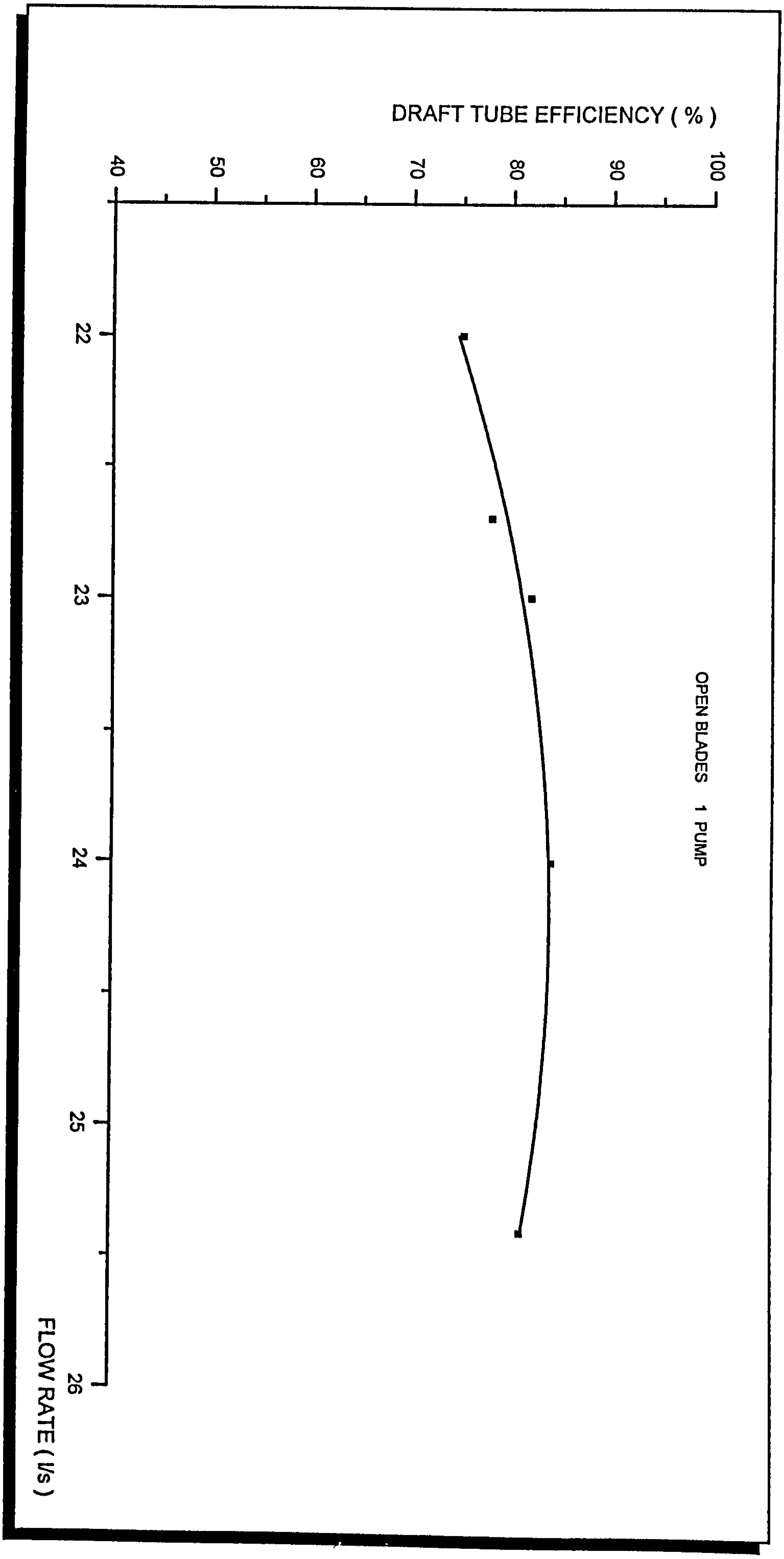

DRAFT TUBE EFFICIENCY ( % )
100
90
80
70
60
50
40
OPEN BLADES 1 PUMP
22
23
24
25
26
FLOW RATE ( l/s )

Figure ( 6.47 )

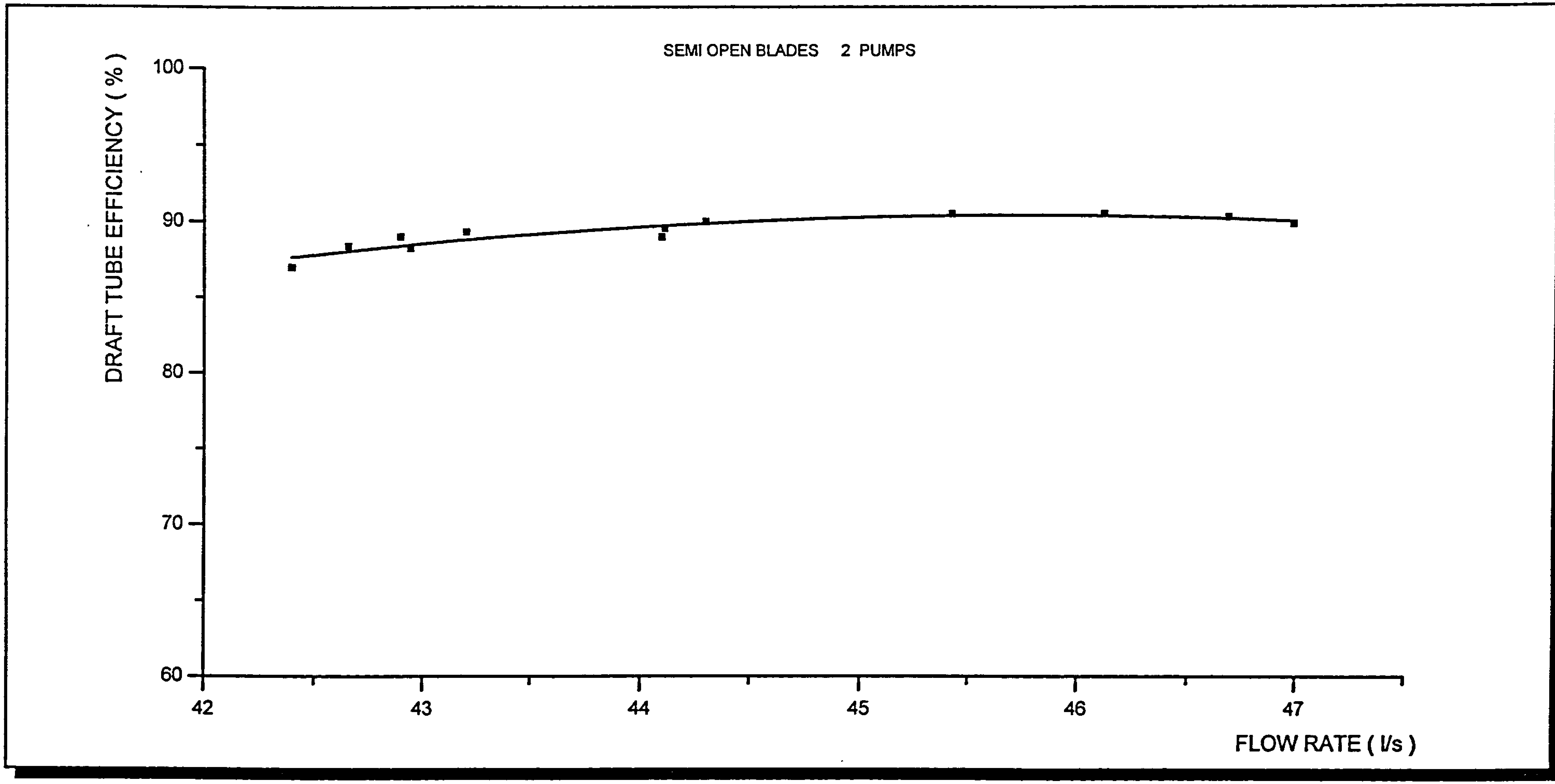

SEMI OPEN BLADES 2 PUMPS
DRAFT TUBE EFFICIENCY ( % )
100
90
80
70
60
42
43
44
45
46
47
FLOW RATE ( l/s )

Figure ( 6.48 )

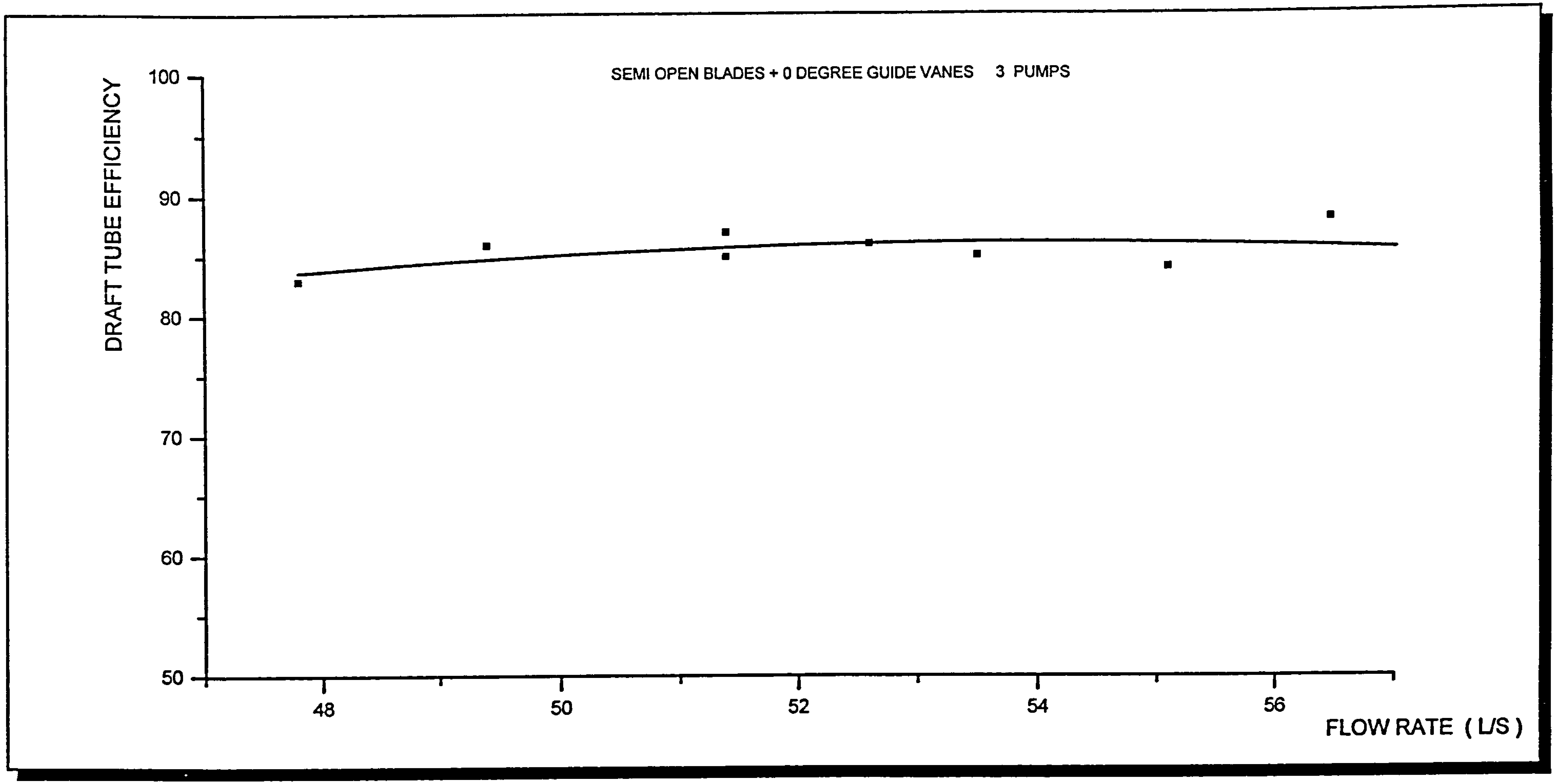
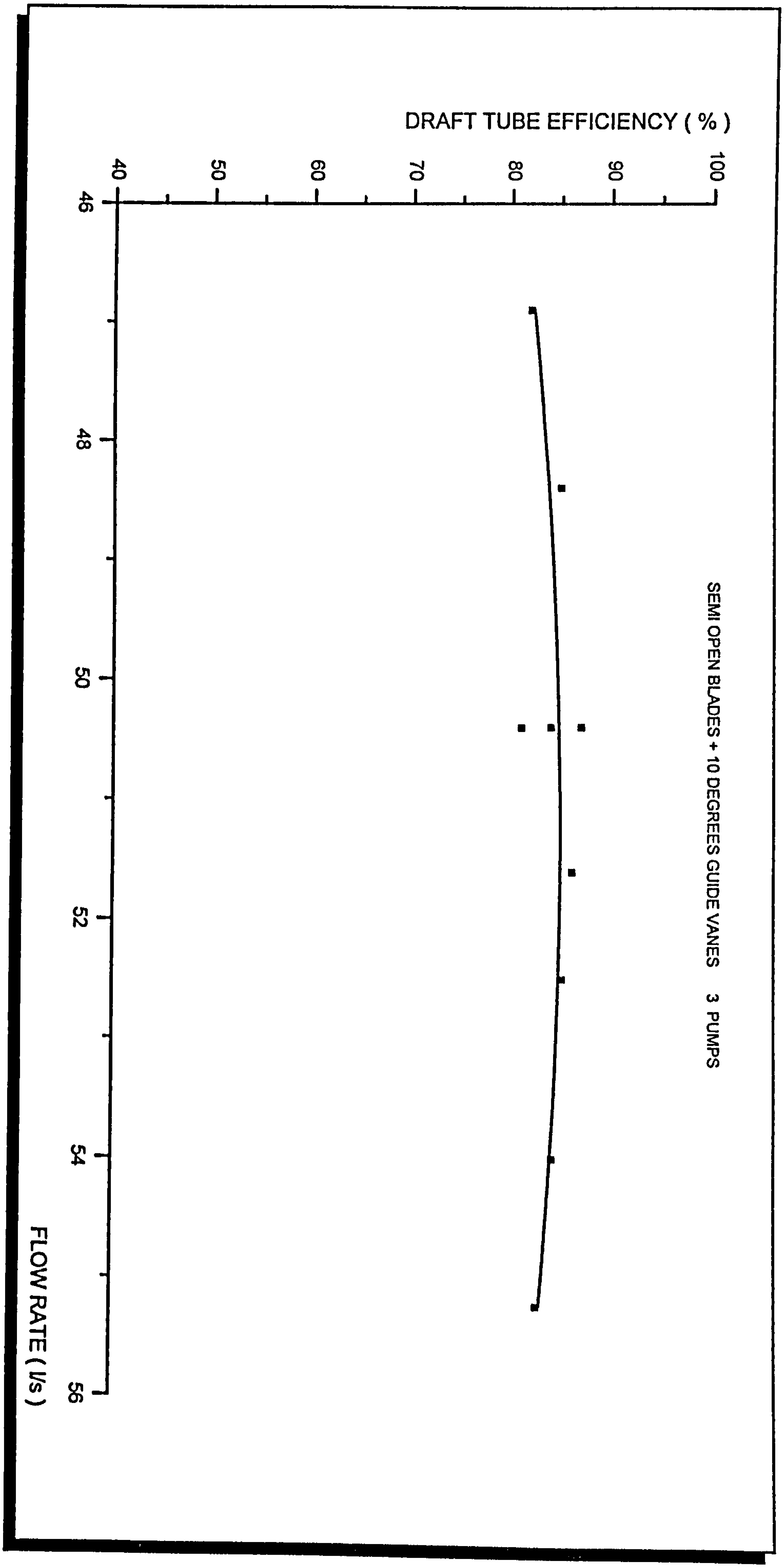

SEMI OPEN BLADES + 0 DEGREE GUIDE VANES 3 PUMPS
DRAFT TUBE EFFICIENCY
100
90
80
70
60
50
48
50
52
54
56
FLOW RATE ( L/S )

Figure ( 6.49 )



DRAFT TUBE EFFICIENCY ( % )
100
90
80
70
60
50
40
46
48
50
52
54
56
SEMI OPEN BLADES + 10 DEGREES GUIDE VANES 3 PUMPS
FLOW RATE ( l/s )


Figure ( 6.50 ).

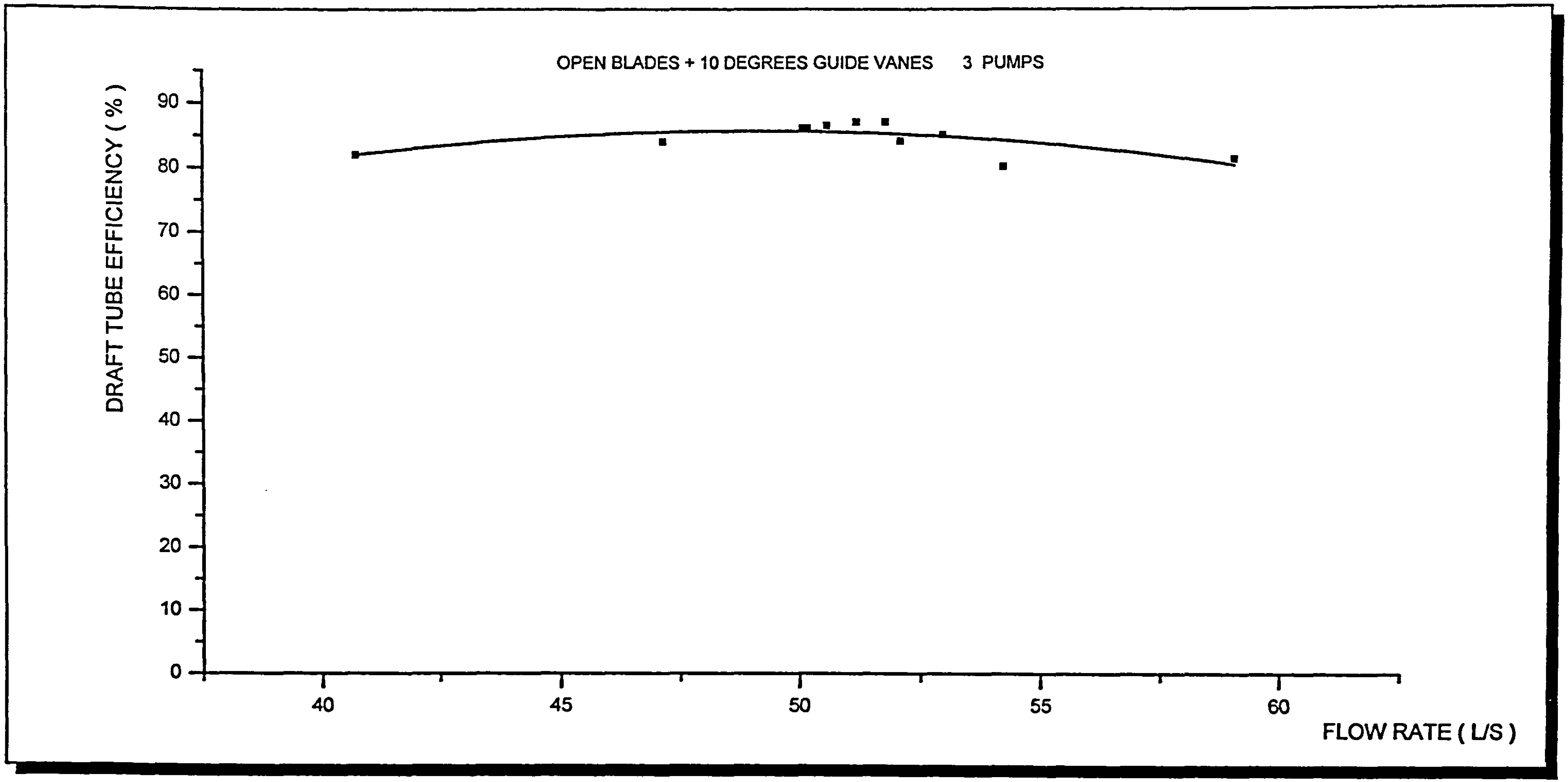

OPEN BLADES + 10 DEGREES GUIDE VANES 3 PUMPS
DRAFT TUBE EFFICIENCY ( % )
0
10
20
30
40
50
60
70
80
90
40
45
50
55
60
FLOW RATE ( L/S )

Figure ( 6.51 )


OPEN BLADES + 10 DEGREES GUIDE VANES 4 PUMPS
DRAFT TUBE EFFICIENCY ( % )
90
80
70
60
50
40
30
20
50
55
60
65
70
75
FLOW RATE ( l/s )

Figure ( 6.52 )

# CHAPTER
# ( VII )

# CONCLUSIONS

## CONCLUSIONS

In this work more than 500 readings are taken in the laboratory of which less than 50% of them are presented in about 140 figures. By considering and comparing all these the following can be concluded:

1. The designed circuit with the help of six pumps is capable of generating precise heads and flow rates within its specified range of micro hydro turbine requirements.
2. Installation of Guide Vanes upstream of the Runner (this includes a second support bearing) can increase the performance of the turbine significantly. The improvement in efficiency for one typical condition is over 20% (runner blades semi-open and guide vane angle zero, Fig. 6.4.5, best efficiency 85%, compared with original turbine, Fig. 6.4.2, best efficiency 63%).
3. Application of the guide vanes yields extension of the high efficiency domain of performances. It can be seen from Fig. 6.14.5 that the turbine (with guide vanes) can operate from 1000 to 4800w with 81% efficiency as compared to Fig. 6.14.2 (original turbine) which is 700 to 3800w with 61% efficiency.
4. A vent hole at the highest point of the main casing of the turbine, for extracting the air bubbles, can effectively result in removing instability due to air in the system and cavitations.

   Such vent hole should be incorporated to the practical turbine designs.

5. It was confirmed that application of the draft tube plays an important role in attaining higher levels of efficiency.
6. Changing of the angle of the guide vanes can lead to an increase of the generated power, but the efficiency of the turbine may remain unaffected or fall slightly.
7. The Agnew turbine is not suitjable for low speed conditions (ie. less than 800 rpm). The optimum conditions for this modified model are as follows:

   Q: 0.045 to 0.055 $m^3/s$

   H: 3 to 6 m

   N: 800 to 1700 rpm

   P: 1000 to 4600 w

8. For further work it is suggested that the following features of the Agnew Turbine be further studied separately:

(a) Design of the guide vane geometry and matching the performance with that of the turbine.

(b) Second support bearing for main shaft

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