ENERGY RECOVERY DURING PUMP TESTING BY REPLACING VALVE WITH PUMP AS TURBINE

A DISSERTATION

Submitted in partial fulfillment of the
requirements for the award of the degree
of
MASTER OF TECHNOLOGY
in
WATER RESOURCES DEVELOPMENT
(MECHANICAL)

By

RAJAN MAN SHRESTHA

CENTRAL LIBRARY
Acc. : G12218
Dt. 20/12/05

DEPARTMENT OF WATER RESOURCES DEVELOPMENT & MANAGEMENT
INDIAN INSTITUTE OF TECHNOLOGY ROORKEE
ROORKEE - 247 667 (INDIA)
JUNE, 2005

CANDIDATE'S DECLARATION

I hereby declare that the dissertation titled "***Energy Recovery During Pump Testing by Replacing Valve with Pump as Turbine*** " which is being submitted in partial fulfillment of the requirement for the award of Degree of **Master of Technology in Water Resources Development (Mechanical)** at department of Water Resources Development and Management (WRD&M), Indian Institute of Technology, Roorkee is an authentic record of my own work carried out during the period of July 2004 to June 2005 under the supervision and guidance of **Professor Devadutta Das,** WRD&M, IIT, Roorkee and **Professor Gopal Chauhan,** WRD&M, IIT, Roorkee.

I have not submitted the matter embodied in this dissertation for the award of any other degree.

Rajan

Place: Roorkee. ....

Dated: June, 2005 **Rajan Man Shrestha**

This is to certify that the above statement made by the candidature is correct to the best of our knowledge.

29/6/05

Gopal Chauhan
Professor WRD&M
IIT, Roorkee
Roorkee-247667, India

Devadutta Das

Devadutta Das
Professor WRD&M
IIT, Roorkee
Roorkee-247667, India

ACKNOWLEDGMENT

I express my deep sense of respect and gratitude to Prof. Devadutta Das and Prof. Gopal Chauhan, Department of Water Resources Development and Management, Indian Institute of Technology Roorkee, for their thorough guidance in bringing out this Dissertation. They have been constant source of inspiration and encouragements; the valuable hours of discussions and suggestions that I have from them have undoubtedly helped me in supplementing my thoughts in right directions for attaining the desired objectives of completing this Dissertation in its present form. Working under their guidance will always remain a cherished experience in my memory and I will revere it throughout my life.

I express my sincere gratitude to Dr. J.T. Kshirsagar, Associate Vice-President, Kirloskar Brothers Limited, Pune for providing me with valuable suggestions, information and time. I am highly indebted to him for promptly and tirelessly answering my queries. Thanks are also due to Mr. S. N. Shukla, Manager, Kirloskar Brothers Limited, Pune for providing necessary reference materials.

Thanks to all faculty of WRDM for their encouragement, constructive criticisms and suggestions. Their keen sense of duty has contributed directly or indirectly in a significant manner towards completion of this Dissertation.

I am indebted to Government of India for providing financial support to pursue M. Tech in IIT, Roorkee, India under TCS Colombo Plan.

My special thanks are due to my colleagues for creating friendly environment both in classes and outside throughout the period of my study in IIT, Roorkee.

I highly appreciate co-operation extended to me by all the staffs of River Engineering Lab. and Computer Lab., WRDM, IIT Roorkee during experiment and compilation of this dissertation.

I am also thankful to my parents, sister and relatives for their whole-hearted support and encouragement.

IIT, Roorkee, India, (Rajan Man Shrestha)

Dated: June, 2005

CONTENTS

List of Figures

List of Notations (Symbols)

S. No.	Notation	Details
1	a	turbine model constant
2	a_1	inlet area of venturimetere (m^2)
3	a_2	throat area of venturimeter (m^2)
4	bhp	brake horse power
5	C_d	coefficient of discharge
6	C_h	conversion factor for head
7	C_q	conversion factor for flow
8	C_{u1}	component of absolute velocity of flow in impeller direction at inlet of pump and out let of turbine (m/s)
9	C_u	component of absolute velocity of flow in impeller direction at outlet of pump and inlet of turbine (m/s)
10	C_m	radial or meridian velocity (m/s)
11	C_v	coefficient of discharge for venturimeter
12	D	diameter of impeller or runner (m)
13	g	acceleration due to gravity (m/s^2)
14	H	total head (m)
15	H_e	Euler head (m)
16	H_{nt}	normal turbine head (m)
17	H_{np}	normal pump head (m)
18	$H_{np(np)}$	nominal pump head at pump speed (m)
19	$H_{np(nt)}$	nominal pump head at turbine (PAT) speed (m)
20	H_t	design head for turbine (m)
21	Δh	head loss (m)
22	I	current in (A)
23	K	pressure loss coefficient
24	K_l	pressure-loss coefficient for partly open valves
25	k_1	constant
26	k_2	constant
27	N	speed of pump or turbine (rpm)
28	N_{sp}	specific speed of pump

29	N_{st}	specific speed of turbine
30	N_t	speed of turbine (rpm)
31	n_p	nominal pump speed (rpm)
32	n_t	nominal turbine speed (rpm)
33	P	Power (W)
34	Δp	pressure loss (N/m^2)
35	Q	discharge in m^3/s
36	Q_t	design discharge for turbine (PAT) (m^3/s)
37	$Q_{np(np)}$	nominal pump flow at pump speed n_t (m^3/s)
38	$Q_{np(nt)}$	nominal pump flow at turbine (PAT) speed n_t (m^3/s)
39	r_2	outer radius of turbine runner (m)
40	r_1	inner radius of turbine runner (m)
41	s	number of stage of pump as turbine
42	T	torque (N-m)
43	TDH	total differential head (m)
44	u_2	peripheral velocity of impeller at inlet of pump and outlet of turbine (m/s)
45	u_1	peripheral velocity of impeller at outlet of pump and inlet of turbine (m/s)
46	V	voltage (Volt)
47	y	height of mercury column (m)
49	ρ	density of fluid (Kg/m$^{3)}$
50	η	efficiency
51	β_2	blade angle of pump at outlet (degree)
52	ω	angular velocity (rad/s)
53	γ	specific weight of mercury (N/m^3)
54	γ_w	specific weight of water (N/m^3)

List of Tables

SYNOPSIS

The pumps are hydro mechanical devices that transfer mechanical energy from some external sources to the liquid flowing through it. They have wide spread use and are produced in a seemingly endless variety of sizes and types. They are applied to an apparently endless variety of services and are probably the second most common machine in use exceeded in number by the electric motors. Such extensive coverage has required the establishment of theoretical analysis and to be supported by graphical representation of actual test.

Pump manufacturers test run their product to confirm and to verify whether the predicted characteristics match with the actual. The pumps that are tested can be very large and require more power and time. Valves are used in a shop performance test that alters the variable friction-head portion of the total system head curve and consequently the pump flow. Most of energy developed by pump, during test, in form of head and discharge is dissipated in valve. It is worthwhile to recover a portion of this energy by replacing valve with pump as turbine that is going un-harnessed, particularly because pumps have been used as turbines for power generation for many years in the field of small hydropower development. Pumps are readily available and many sizes are stock items and are several generations ahead of conventional turbines in cost reduction. They are less sophisticated, easy to install, maintain, simpler to operate and are readily available in a broader range of configurations than conventional turbines.

Study of h-q characteristic of pump as turbine, at certain speed, shows similarity with the h-q characteristic of valve at a certain valve opening position. However, varying the speed of pump operating as turbine can replicate h-q characteristic of valve to a limited extent at different valve opening positions but not entire valve stem or opening positions. With more number of pumps as turbines, when connected in parallel and operated in different combinations and speed results in replication of valve characteristics for different valve opening positions. Simultaneous operation of pumps as turbines and speed regulation by varying loads on the energy recovery system comprising of pumps, (in parallel) as turbines, replicates the valve characteristics and can replace valve used during pump testing recovering a portion of energy that is used in test. A system is proposed that recovers energy using reverse-running pumps acting as turbines during performance testing of 24UPH3 pump of Kirloskar, India. The same system can be used for testing pumps of different capacities, which recovers energy during test.

1. INTRODUCTION

1.1 GENERAL

Man has used energy at an increasing rate for sustenance and well being. With industrial revolution, consumption of energy in form of fossil fuels, oil and natural gas has increased extremely. The invitation of heat engines and the use of fossil fuels made energy portable and introduced the much-needed flexibility. The flexibility was enhanced with the discovery of electricity and the development of central power generating stations either using fossil fuel or water supply. The increased demand of electric power, with rapidly depletion of fossil fuels caused uncertainties about the future availability and cost of fuel. This leads to worldwide energy crisis causing increase in oil price. In this scenario of energy, we are forced to think in innovative ways to use renewable energy and to recover energy in every possible manner.

Industries are constantly looking for opportunities to reduce cost and increase productivity in addition to more effective methods to better integrate their operation to reduce energy wastage. In many industrial processes, the liquid remains at a higher pressure even after it has completed its utility in the process and has to be brought down to a lower pressure before further processing. The conventional method is to passes through a pressure-reducing valve to affect the required pressure drop. Instead of loosing the capacity of the high-pressure liquid, which is capable of work, an economically and environmentally attractive alternative is to install an energy recovery turbine at lower point and turn waste energy into electricity.

Similarly, pump manufacturer test run their product to confirm the design condition. Energy developed by pump in form of pressure head and discharge is dissipated by valves during test. Recovery of energy replacing throttle valve or orifice by turbines is the beneficial use of head and discharge energy available that would otherwise dissipated. Technologies that recover energy reduce the cost and consumption of energy. The recaptured energy is potentially useful for driving pump or other piece of rotating equipment. The recovered energy can also be used for heating and/or cooling, dehumidifying outdoor air brought into a building/factory for ventilation, space heating, lighting, water heating and many more.

1.2 NECESSITY OF ENERGY RECOVERY

There are many irrigation canals, wastewater treatment plants, cooling system in thermal power stations where hydraulic energy is dissipated or otherwise wasted. Pressure reducing valves, free discharge valves, etc waste energy [12]. Industrial processes where energy is reduced in pressure reducing valves are cryogenic system, petrochemical refinery processes and seawater desalination [1]. Similarly large amount of energy is used in hydraulic laboratory during testing of hydraulic machines is dissipated in valves.

There has been increasing gap between energy supply and demand and this energy deficit is increasing alarmingly. The increasing demand and cost of energy can be tackled by using energy in efficient and effective ways. Energy recovery is one of the ways of efficiently utilizing energy. Energy recovery during pump- testing, seawater desalination, petrochemical refinery process and cryogenic system will contribute significantly to cost reduction and energy utilization.

1.3 UTILITY OF WORK

Evaluations of performance of newly designed and manufactured pumps are done by test running them at their design speed and out put. The pumps that are to be tested can be very large and require more power. Valves are used to change the system head for the purpose of varying the pump flow during a shop performance test. Once the flow enters the valve, most of the energy that the pump develops in the form of head and discharge is wasted.

Previous works on energy recovery focus on recovering energy using pump as turbine from constant head and discharge rather than replacing valve totally by turbines in parallel. Replication of valve characteristics by turbines in parallel can regulate flow through them while evaluating the performance of the test pump and recovering at least a part of energy that is being wasted.

Centrifugal pumps as turbines may be used to advantage as valve for flow regulations when they are run in parallel. Energy recovery generally deals with small power generation. Design and construction of suitable turbines for small power generation is associated with difficulties. It is therefore worthwhile to investigate the use of relatively cheaper, volume produced and in varieties, centrifugal pumps as turbine and their operation in parallel. The advantage of using the centrifugal pump has is low capital cost and disadvantage lies in fixed guide apparatus. At a time, when industrial production is affected to a great extent because of high power cost and shortage, employing pumps as turbines operating them in parallel for flow regulation and power generation can be of important use. Industries that deal with hydraulic machines use large amount of power while testing hydraulic machines that they manufacture. The energy which is going waste can be exploited by employing suitable numbers of pumps to run as turbine in parallel. With the power recovered, it can be used for various purposes from heating to supplement the power requirement of industries.

Another example is in water supply pipelines where pressure available is very high; pumps as turbines in parallel may be implemented for flow regulation and pressure release recovering energy which would be wasted by valve in the process of pressure release.

1.4 AIMS AND SCOPE OF PRESENT WORK

When proto type pumps are tested to confirm their head-capacity, power out put and efficiency, a large amount of energy is used in developing head and discharge. This energy, which in form of pressure head and discharge, is dissipated in form of heat once it passes through flow regulating valve. So a huge amount of energy and money is spent in testing pumps. If the developed head and discharge could be used for power generation, it would contribute to cost reduction in significant manner.

This work is carried out to study the possibility of replacing valve by pump working as turbine. If gangs of pumps, which work in parallel, as turbines could replicate valve characteristics; they can regulate flow and help recovery energy, which is otherwise wasted.

The present work emphasizes on following studies.

- Study of various performance curves of different types of pumps
- Study of characteristics performance of valves for various opening positions
- Study of characteristics performance of pump as turbine
- Study to find out whether the turbine can replicate valve characteristics and can be used in pump testing
- Designing an energy recovery system that test performance of pumps

1.5 ORGANIZATION OF DISSERTATION

The study is organized to achieve its objectives in six chapters. The contents in this dissertation are presented briefly as following chapters.

Chapter – 1	is an introduction. It gives general information about need of using pump as turbine in flow regulation. It also deals with the problem definition and objective of present study.
Chapter – 2	is literature review. It studies the works done in using pump as turbine and use of pump as turbine in pressure regulation and energy recovery.
Chapter – 3	studies characteristics of pumps, valves, pump as turbine and theoretically considers replication of valve characteristics by pump as turbine.
Chapter – 4	is experimental set-up and test results.
Chapter – 5	discusses results of tests and their implication for designing test system that recovers energy.
Chapter – 6	is conclusions of the study along with recommendations.

2. LITERATURE REVIEW

2.1 GENERAL

Apart from normal operation of pumps in their usual head-capacity and speed range, some of the special operating condition are unavoidable, others occurs accidentally, and still others can be reproduced only in the laboratory or they develop during a transient period while changing from one state to another. Operation of a centrifugal pump as a hydraulic turbine, behavior of a pump in the event of power failure, or starting a pump running in reverse are example of unusual operating conditions [17].

Centrifugal pump as turbine is long being used as power recovery turbine in process industries [3]. Energy has been extracted using pump in reverse mode from water supply network where excessive pressure head was dissipated by throttling through sluice gates or valves.

Rising energy costs and predicted energy shortages, forced industries to eye on every possible energy resources. Use of pump as hydraulic turbine is a practical and economical approach for energy generation and recovery in small scale. Multistage radial-flow units have been used as power recovery turbine in process industries [3]. A Great interest was shown in study of operation of centrifugal pump as hydraulic turbine, as centrifugal pumps may be used to advantage as small hydraulic turbines where low initial cost is imperative. Experience and modern technology made pump as turbine a viable alternative for small hydro plant potential and if properly selected, pump operating in reverse yields good efficiency.

Success of pump as turbine, both technically and financially, made industries to exploit the benefit of pump by using it in energy recovery. In conventional liquefaction plants, the high pressure LNG is expanded in a throttling valve. The objective of throttling is to decrease the pressure of LNG to the levels manageable for economic transportation and allow the refrigeration cycle to be completed. A throttling valve can be replaced by turbine. This way the same pressure drop can be exploited by producing power. An investigation of cryogenic turbine revealed that replacing the throttling valve with the cryogenic turbine can save an LNG liquefaction plant about half a million dollars a year in electricity costs [13].

A major operating cost of South African deep gold mines is refrigeration, which is mainly done with water piped down the mines. The descent of the water releases large amounts of potential energy. This energy is either dissipated into heat by pressure release valves or can be party be recovered with energy recovery equipment such as turbines. A study finds and proposed a system that uses a reverse-running multistage pump acting as a turbine to do simultaneous pressure regulation and pressure recovery in secondary cooling water system in deep mines. The turbine drives a standard motor that acts a generator. A constant drop across the system is maintained with bypass and inline valves [1].

2.2 PUMPS AS TURBINE

In 1931 Kittredge, C. P. and Thoma published an article on "Centrifugal Pump Operated Under Abnormal Condition" [8]. The paper described experiments carried on with a small pump for the purpose of obtaining performance characteristics from which the behavior of the pump during sudden changed of operating conditions could be predicted. In these experiments the pump was operated under condition of negative head, delivery, and speed, in addition to the normal range of performance.

As an outgrowth of the work, a series of investigation was undertaken in the hydraulics laboratories of the California Institute of Technology under the direction of Knapp, R. T. and in his paper "Complete Characteristics of Centrifugal Pumps and Their Use in the Prediction of Transient Behavior" [10] describe the technique of determining the complete characteristics of a hydraulic machine such as a centrifugal pump or a turbine, together with a method of presenting these characteristics in a convenient manner on single diagram. It has been found that the possible operating conditions of hydraulic turbine and centrifugal pump when compared, the pumps are subjected to much wider and more involved variation than are the turbines, especially

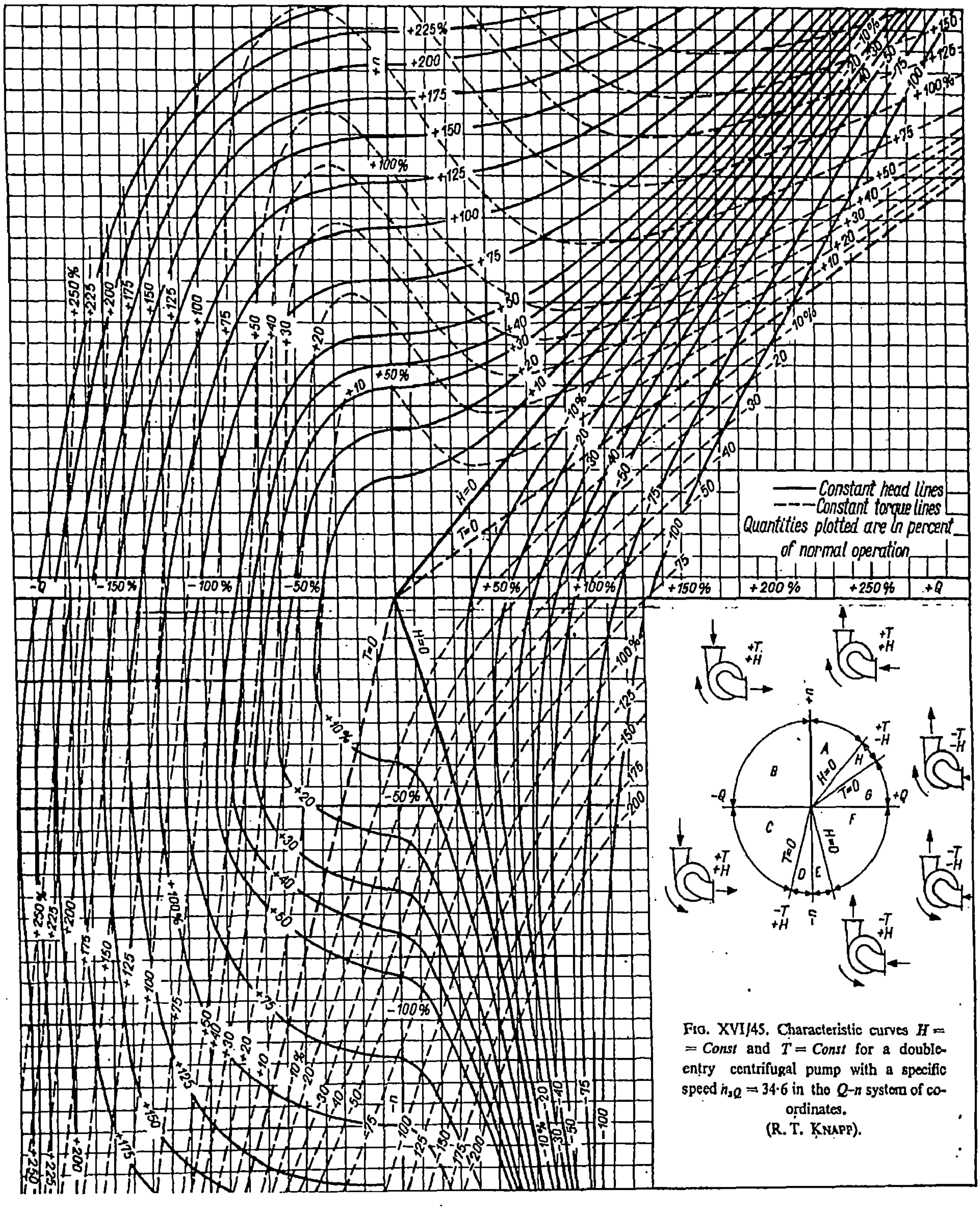


FIG. XVI/45. Characteristic curves $H = Const$ and $T = Const$ for a double-entry centrifugal pump with a specific speed $n_{sQ} = 34{\cdot}6$ in the Q-n system of co-ordinates.
(R. T. KNAPP).

Fig.. 2.1 Complete pump characteristics, double suction pump; N_s=1800 (FPS unit) (Stepanoff, A. J) [17]

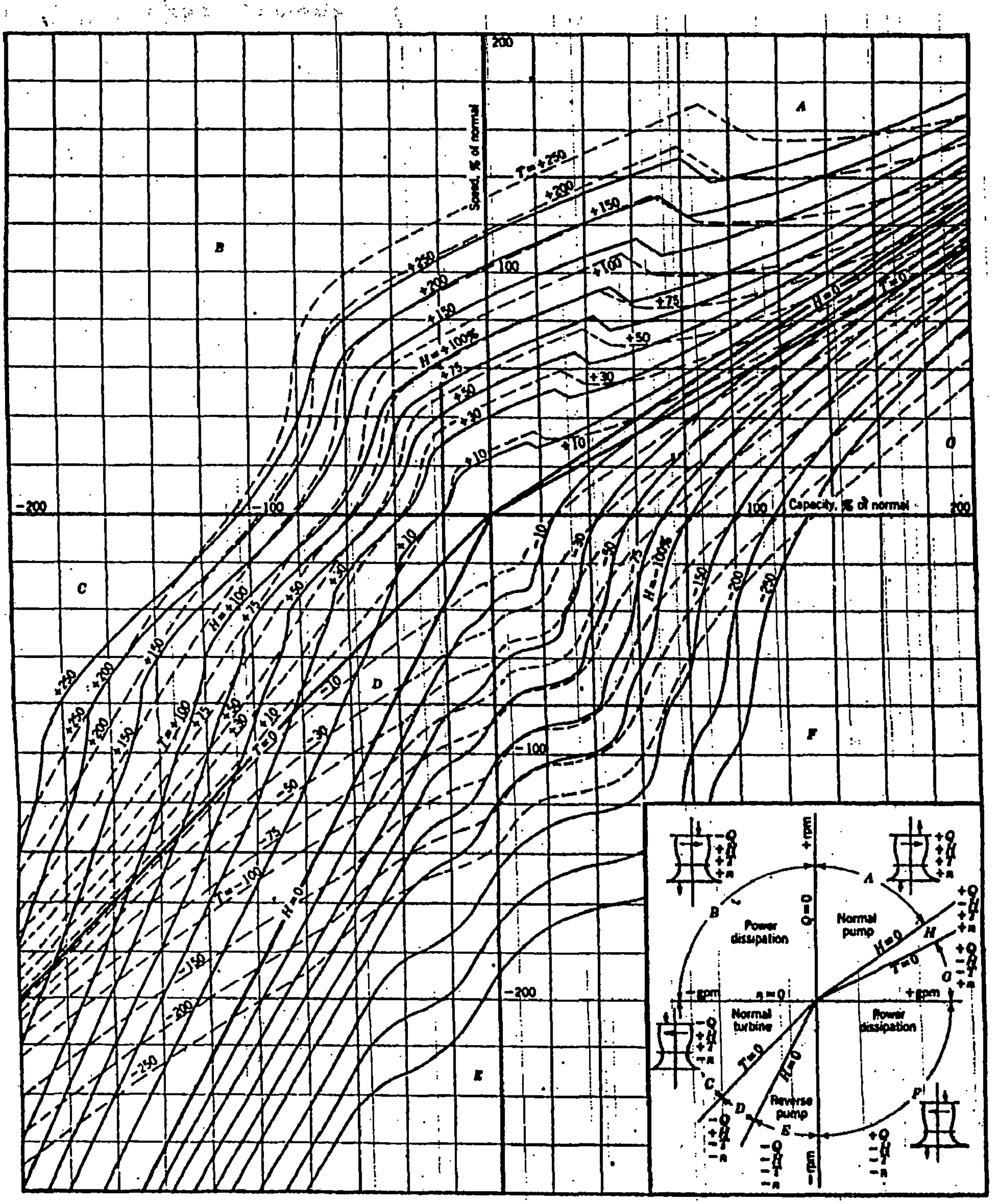


Fig. 2.2 Complete pump characteristics, mixed flow pump; N_s=7500 (FPS unit) (Stepanoff, A. J) [17]

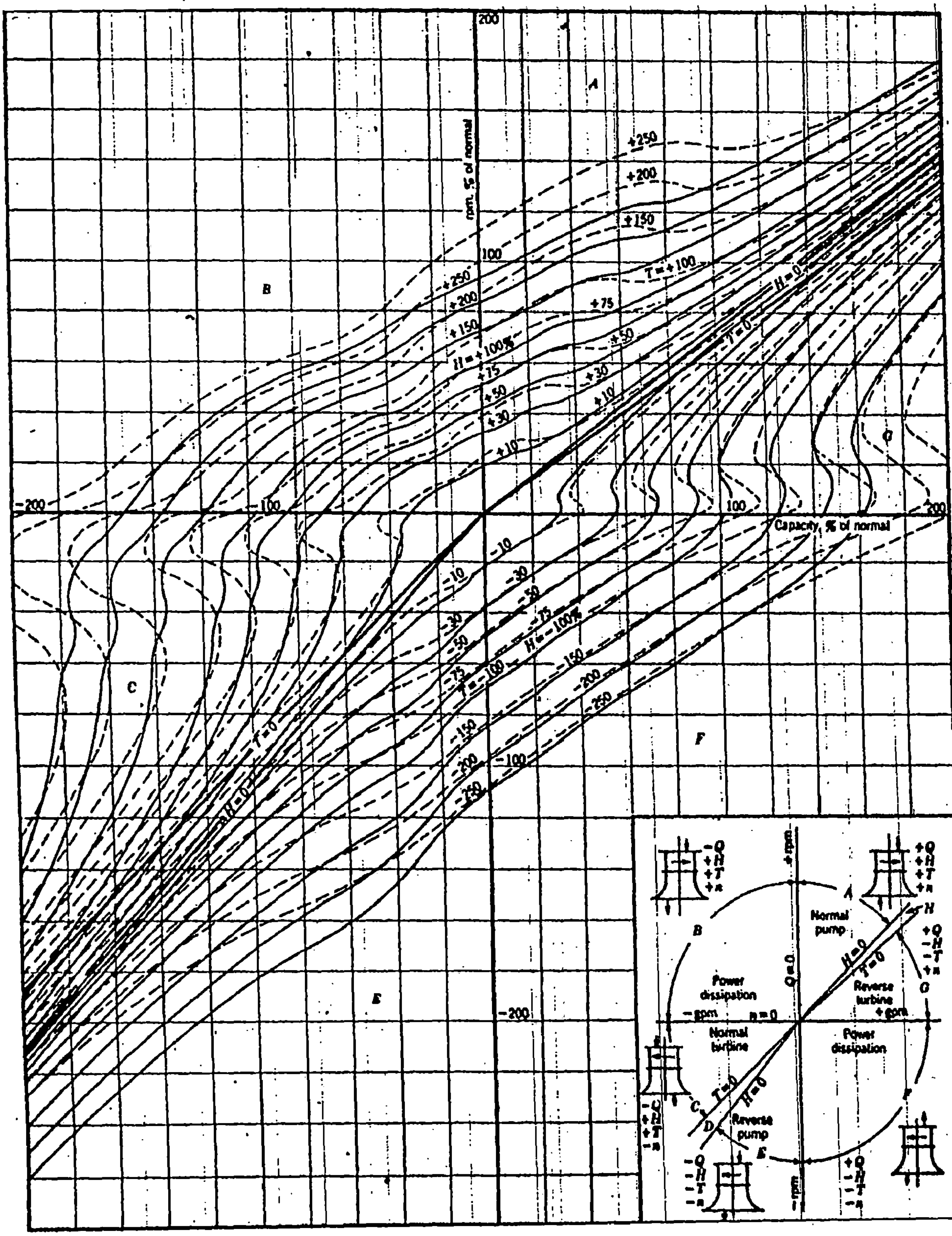


Fig. 2.3 Complete pump characteristics, axial flow pump; N_s =13500 (FPS unit) (Stepanoff, A. J) [17]

during the transient states of staring, stopping, or emergency operation. In order to graphically present the various operation conditions, the test results were presented on a single four-quadrant diagram having as coordinates the discharge and the speed. Fig. 2.1, 2.2 and 2.3 are complete characteristic curves presented on a single four-quadrant diagram for pump of specific speed 1800, 7500 and 13500 (FPS unit) respectively. The data are presented as series of contour curves of constant values of head and torque, the full lines being of constant head and the dotted ones lines of constant torque..

Kittredge, C. P. in his article "Centrifugal Pumps Used as Hydraulic turbines" [9] describes that centrifugal pumps may be used to advantage as hydraulic turbines in applications where the required power is rather small and low initial cost is more important than high efficiency. An estimate of the performance of a centrifugal pump when used as a turbine can be made from published curves of complete characteristics for typical pumps. His paper presents some of the available data on pumps and shows how to select a pump that will meet specified requirement when used as a turbine.

Chapallaz, J. M in GTZ publication titled "Manual on Pumps Used as Turbines" [2] provides a practical method to select a pump as turbine for a specific purpose. The application of pumps as turbines is not yet as widespread as the use of the same machine in its normal operation; performance curves are usually available for the pump mode. Methods to predict turbine mode performance from pump mode data have been developed. But due to the different pump designs, none of the methods developed so far are 100% reliable and errors from predicted to actual flow and head in turbine mode may be as high as 10% or more. The manual presents a pump selection method to use as turbine based on two parameters of the pump mode performance; they are, best efficiency and specific speed. Specific speed is a measure for the shape of the runner; efficiency may represent the different design features of a machine including the volute casing, which plays an important role in the turbine mode.

The selection charts presented in the manual are based on more than 80 tests results and considerable amount of documents and data on pump operated as turbines has been gathered and incorporated critically.

Summary of the selection procedure of pump as turbine based on reference [2] is as follows;

After finding head and discharge of site and deciding the speed of turbine, specific speed is found by following equation.

$$N_{st} = N_t \frac{\sqrt{Q_t}}{H_t^{3/4}} \quad \text{.......2.1}$$

where,

N_{st} is specific speed of turbine

Q_t design discharge for turbine (m^3/s)

H_t is design head for turbine (m)

N_t speed of turbine (rpm)

turbine specific speed is converted into pump by using relation

$$N_{sp} = \frac{N_{st}}{0.89} \quad \text{.......2.2}$$

where, 0.89 is a coefficient found from several tests and is relation between turbine and pump-mode specific speed which take fairly constant value that is, 0.89

Using pump specific speed and efficiency, conversion factors for head C_h and flow C_q is found and using following relation conversion from turbine design conditions into pump condition is obtained.

$$\frac{H_{nt}}{H_{np}} = C_h \quad \text{.......2.3}$$

$$\frac{Q_{nt}}{Q_{np}} = C_q \quad \text{.......2.4}$$

where, H_{nt} and H_{np}, are nominal turbine head and nominal pump head in metres, Q_{nt} and Q_{np} are nominal turbine flow and pump flow in m^3/s respectively.

Nominal pump head at turbine speed n_t is given by

$$H_{np(n_t)} = \frac{H_{nt}}{C_h} \quad \text{.......2.5}$$

Nominal pump flow at turbine speed n_t

$$Q_{np(n_t)} = \frac{Q_{nt}}{C_q} \quad \text{.......2.6}$$

Fig. A1 shows the conversion factors for head and flow and are dependent on specific speed and efficiency of pump.

Converting pump design condition at turbine rated speed into pump rated speed

Nominal pump head at pump speed n_p

$$H_{np(n_p)} = H_{np(n_t)} * \left[\frac{n_p}{n_t}\right]^2 \text{ ....2.7}$$

Nominal pump flow at pump speed n_p

$$Q_{np(n_p)} = Q_{np(n_t)} * \left[\frac{n_p}{n_t}\right] \text{ ....2.8}$$

Saini, R. P. and Ahmed, N. [15] in their paper titled "Method for Selection of Pumps used in Turbine Mode" presented a nomogram developed on the basis of actual test data of centrifugal pumps in turbine mode. The nomogram avoids lot of calculation work required in selection of pumps for turbine use; it has been found that the nomogram could be a good tool to assist selection of pumps for turbine applications.

The nomogram has been developed on the basis of actual test data obtained by conducting test on different sizes pumps in turbine mode, ranging specific speed form 10 to 300 (metric). The specific speed N_s in metric unit was computed using the following equation.

Specific Speed $N_s = \dfrac{N\sqrt{Q_T}}{H_T^{0.75}}$ 2.9

The best efficiency point of pumps in turbine mode has been determined and subsequently conversion factors for head and discharge were determined using following equations.

Conversion factor for head $h = \dfrac{H_T}{H_P}$ 2.10

And

Conversion factor for discharge, $q = \dfrac{Q_T}{Q_P}$ 2.11

Where,

N are rpm of machine

H_T is head on machine in turbine mode at best efficiency point (m)
H_P is head on machine in pump mode at best efficiency point (m)
Q_T is discharge through machine in turbine mode at best efficiency point (m^3/s)
Q_P is discharge through machine in pump mode at best efficiency point (m^3/s)
Finally plots of conversion factors versus specific speed have been prepared and other parameters required for selection of pumps were computed and plotted in different figures.

Nomogram comprises the following components.

Fig. 2.4 shows the specific speed of pumps computed using Eq. 2.9 at different rpm (750 rpm, 1000rpm, 1500rpm and 3000rpm)

Fig. 2.5 shows the specific speed versus site discharge plots. Different plots in the figure are drawn corresponding to different values of site head, ranging from 5.0m to 150.0m.

Fig. 2.6 shows the relationship of specific speed versus correction factors for head and discharge, determined by using Eqs 2.10 and 2.11 Curve (h) and (q) correspond to the correction factor for head and discharge respectively.

Fig. 2.7 shows the relationship between the pump head and correction factor for head. The curves in the figure corresponding to different values of site heads, ranging from 5m to 150m.

Fig. 2.8 shows the relationship between the pump discharge and correction factor for discharge. The curves corresponding to different values of sites discharge, ranging form 0lps to 500lps.

1. Take the value of given site head and site discharge (e.g., H_T =50m, Q_T =150lps). Draw a line parallel to X-axis through the given discharge on Y axis in Fig. 2.9 and locate the point in the corresponding curve head.
2. Draw an upward vertical line form the located point to intersect the specific speed scale for given rpm of the generator (N=1500). The value will be the specific speed of turbine (N_s = 40).
3. Take this values of specific speed (N_s = 40) on X axis of the Fig. 2.9. Now draw an upward vertical line from this point to cut the correction factor curves (h) and (q) for head and discharge. The located point on curve (h) will

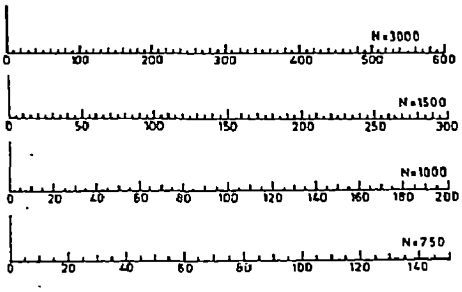


Fig. 2.4 Specific seed at different rmp (Saini) [15]

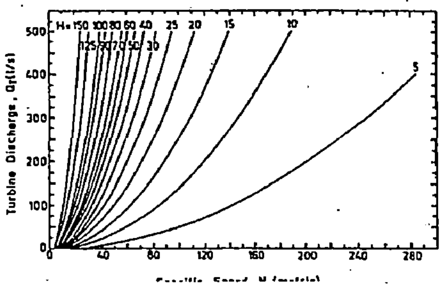


Fig. 2.5 Specific speed vs. site discharge (Saini) [15]

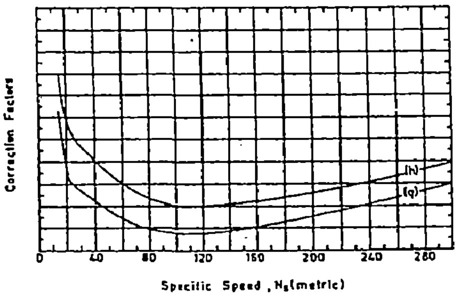


Fig. 2.6 Specific speed versus correction factors for head and discharge (Saini) [15]

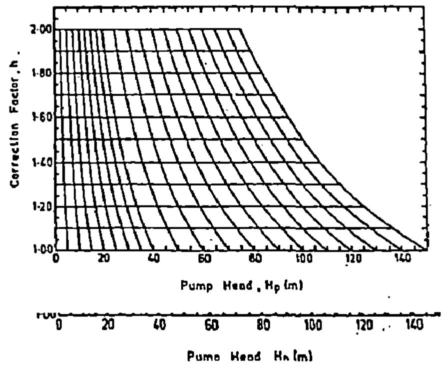


Fig. 2.7 Head correction factor vs. pump head (Saini) [15]

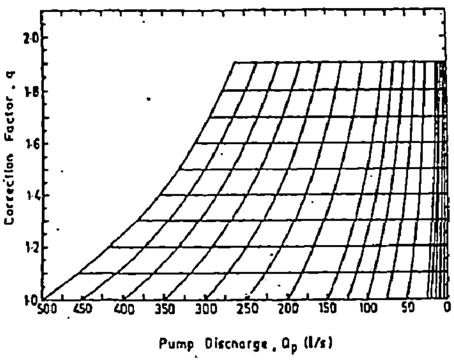


Fig. 2.8 Discharge correction factor vs. pump discharge (Saini) [15]

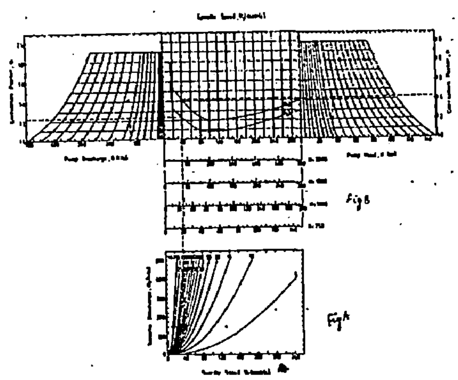


Fig. 2.9 Nomogram for selection of head and discharge of pump used in turbine mode (Saini) [15]

correspond to the correction factor for head and the point located on curve (q) will correspond to the correction factor for discharge (h=1.41 and q= 1.23)

4. To determine the required head of the pump draw a horizontal line through, the located point on head correction factor curve (h). locate the point on the curve corresponding to site head. Now draw vertical line through this located point to meet at X axis. This point will give the value of required head of the pump (H_p =35m).
5. Similarly to determination the required discharge of the pump, draw a horizontal line through the located point on discharge correction factor curve (q). locate the point on the curve corresponding to site discharge. Now draw downward vertical line through this located point to meet X-axis. This point will give the value of required discharge of the pump (Q_p =125lps).

The paper titled "Experimental and Numerical Studies on a Pump as a turbine" [16] presents finding from the experimental work at IWK, University of Karlsruhe, Germany and the computational fluid dynamics (CFD) studies at Kirloskar Brothers, Pune, India. The experiment work throws light on overall performance of the pump in turbine mode, whereas CFD analysis brings out details of flow pattern throughout the geometry. The experiment was carried out with single staged, end-suction standard centrifugal pump with a medium specific speed (35 rpm-SI). The impeller diameter was 206 mm with 8 radial vanes. In pump mode, the best efficiency flow was 25.4 l/s at 12.8 m head, and efficiency was 77% at 1500 rpm. The test was carried out at different operating speeds. The characteristics were determined for the whole operating range starting from no load to maximum load reached in the running condition. The maximum load was limited by the input energy supplied as well as the loading limit of the DC Generator. The experimental characteristics were determined at five different operating speeds, namely 900rpm, 1000 rpm, 1200 rpm and 1300 rpm covering all regions of operation. The maximum efficiency achieved was over 80% for most of the operating speeds. The over all performance was converted to non-dimensional parameters namely the Discharge number Q/ND^3, Head number gH/N^2D^2, and Power number $P/\rho N^3D^5$. The value of peak efficiency at different operating speeds was different but occurred at a constant Q/ND^3 of '0.152'. The η_{max} for 900 rpm, 1000 rpm, 1100 rpm, 1200 rpm and 1300 rpm were 79.7%, 79.7%, 80.1%, and 81.3% respectively. The curves presented the remarkable collapsibility of the various constant speed curves. Fig. 2.10 is overall performance of the turbine in non-dimensional parameter.

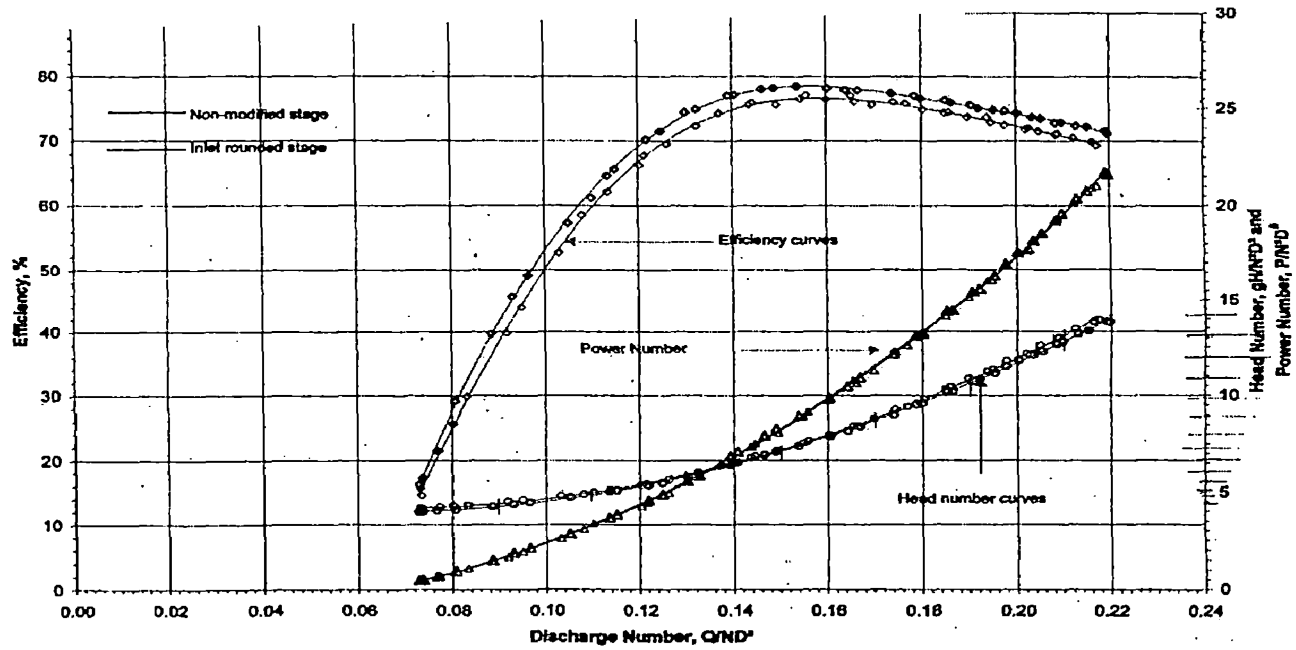


Fig. 2.10 Overall performance of the turbine in non-dimensional parameter (Singh, P.) [16]

Pumps are designed as fluid movers. However, pumps have been used as turbines for power generation for many years. Industrial power recovery instillations have been the most prevalent. A familiarity with pumps and a concern for reducing process energy consumption also has been a contributing factor for using pump as power generating unit [11].

In an application of a pump used as a fluid mover, the liquid enters the suction part at lower pressure, absorbs shaft work in the impeller, and leaves the discharge nozzle at high pressure. When a pump is used as a turbine, the liquid enters at high pressure, drives the impeller in reverse, and leaves at a lower pressure [3].

A basic difference between conventional pumps and hydraulic turbines is that pumps generally do not have a flow control device. Pumps are usually designed for a specific total dynamic head and discharge requirement. Thus, pumps as turbines are best suited to constant flow applications [3].

As centrifugal pumps are designed as fluid movers, they may be less efficient as hydraulic turbines than equipment exclusively designed for that purpose. However, using pumps as turbines have the advantage of being mass-produced in many countries. Pumps are readily available and many sizes are stock items. Pumps are several generation ahead of conventional turbines in cost reduction; they are less sophisticated, easy to install and maintain, and simpler to operate. They are readily available in a broader range of configurations than conventional turbine. When used with an integral induction motor, they can be installed as a combines turbines and generator unit. From the experiences we have no doubt that a pump can be operated as a power generating unit and our concerns are more related to equipment application, efficiencies, operating range, cavitation characteristics, transients, costs, etc [3].

2.3 USE OF PUMP AS TURBINE IN ENERGY RECOVERY

Pacific Pumps in their publication titled "Hydraulic Power Recovery Turbines (HPRT)"[14] states that HPRT is an alternative drive for pumps and other rotating equipment in very selective applications. HPRT are centrifugal pumps that have been modified mechanically to operate with rotation reversed and modified hydraulically to accept reverse flow. System flow in HPRT should be such that flow through it is never less than the minimum specified. Constant speed units generally require at least 20 percent of the design flow to prevent noise or overheating. Also flow less than about 40 percent of design flow will create a drag on the auxiliary drivers. The minimum flow required will be reduced if an over –running clutch is used or variable speed operation is contemplated.

The paper compares the performance characteristics of an HPRT typical of current instillation in process plants with the characteristics of the same machine functioning as a pump; both machines are assumed to have a best efficiency of 80 percent. The most obvious difference between the pump and the HPRT is the reverse trend of the head-capacity curve. Beyond the flow corresponding to zero brake horsepower (bhp) output, the HPRT total Differential Head (TDH) rises rapidly with increasing flow and the HPRT bhp output curve also rises with increasing flow more rapidly. The paper also states that the typical HPRT performance change with speed and diameter of impeller. The change in performance characteristics can be determined by using Pump "Affinity Laws." That is any given flow will vary as the ratio of speeds; TDH will vary as the ratio of speed squared; and bhp will vary as the ratio of speed cubed.

$$Q_1 * \frac{N_2}{N_1} = Q_2 \qquad \text{2.12}$$

$$TDH_1 * \left(\frac{N_2}{N_1}\right)^2 = TDH_2 \qquad \text{2.13}$$

$$bhp_1 * \left(\frac{N_2}{N_1}\right)^3 = bhp_2 \qquad \text{2.14}$$

But it is less effective to change speed to reduce TDH at a fixed flow corresponding to best efficiency point. The only significant change in TDH occurs at low flows where an HPRT is usually not operated. The head-capacity curves tend to converge into common curve in the range of normal HPRT operation regardless of speed change.

Antwerpen Van H. J. and G.P. Greyvenstein in their paper titled "Use of turbines for simultaneous pressure regulation and recovery in secondary cooling water systems in deep mines"[1] proposed a system that uses a reverse-running multistage pump acting as a turbine to do simultaneous pressure regulation and energy recovery in secondary cooling water system in deep mines. The turbine drives a standard induction motor that acts as a generator. A constant drop across the system is maintained with bypass and inline valves. As operation requirement demands, the pressure relief system that change its own pressure drop characteristic according to the down stream flow demand, so that the down stream pressure remains constant. The two viable options are to regulate the pressure by varying the turbine speed or to operate the turbine at constant speed with inline and bypass pressure regulating valves.

In variable turbine speed system, the flow rate is varied while the pressure drop is kept constant at 30m. To vary the flow rate at 30 m head, the turbine speed has to vary from 2200 rpm for 15l/s to 900 rpm for 31 l/s. Characteristics of turbine at speed other than synchronous speed is obtained by using mathematical turbine model. The model is relationship between the head/speed/flow (H_t, N, Q) of a reverse running pump and is given as

$$H_t = s(a_1Q^2 - a_2QN + a_3N^2) \qquad 2.15$$

Where, a_1, a_2 and a_3 are empirical constants and s the number of stages.

The turbine model is demonstrated by fitting it to experimentally determined characteristics. Fig. 2.11 is predicted head/flow data fitted with experimental head/flow curves at various speeds. Variation of speed is viable by using a doubly fed induction generator system in which the exacting frequency changed according to the rotation speed in order to always maintain the synchronous output frequency.

In constant turbine speed system, pressure is controlled by inline and bypass pressure reducing valves. The optimal flow rate is at 28 l/s and for flow rate less than optimal, an inline valve is needed to sustain the reduced pressure drop. A flow rate larger than optimal, the turbine pressure drop exceeds 30 m so that a bypass valve is required as in the case with the variable speed system. At low flow rates, when the turbine is shut down, the inline pressure-reducing valve also acts as an isolating valve.

Study indicates that the variable speed system has a marginally wider peak efficiency range, while the constant speed system has a wider low efficiency range, so depending on the shape of the flow volume/ flow rate distribution, one may recover more energy with one configuration than with the other.

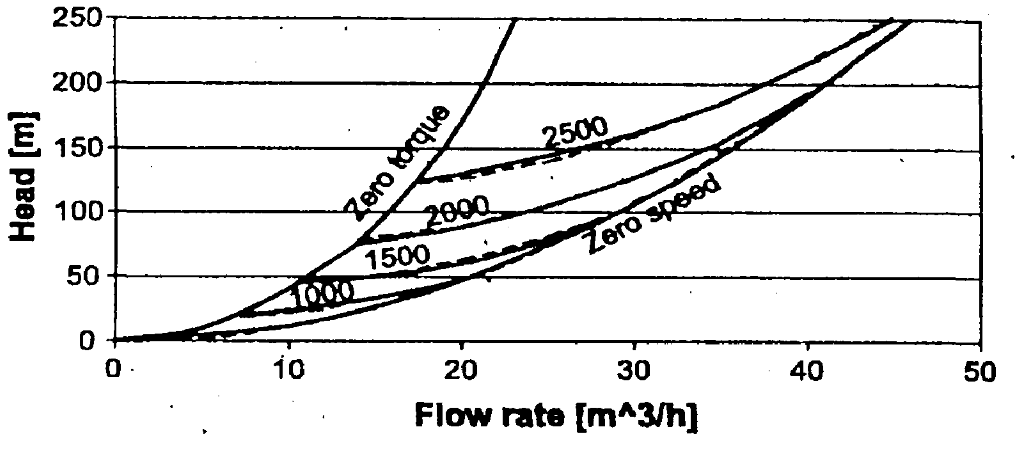


Fig. 2.11 Experimental (solid) and predicted (dashed) head/flow curves for the turbine data presented by Bohl (Antwernpen, H. J.) [1]

CENTRAL LIBRARY
Acc.: G12218
Dt.: 20/12/05

2.4 SUMMARY

Above literature review over last decades reveals lack of study in energy recovery during pump testing by replacing valve with pump as turbine. In the line of above, energy recovery based on using pump as turbine during pump-testing is placed a premium scope to asses the suitability of energy recovery in hydraulic machinery industries; success of it enables lowering the testing cost and energy demand of hydraulic machinery industries. The same system can be used for flow regulation in water supply networks while generating energy.

In this endeavor, a sincere effort has been made in subsequent chapters to analyze the suitability of using pump as turbine in regulating flow and using it in pump testing to recover energy.

3. THEORETICAL CONSIDERATIONS

3.1 PUMP AND ITS CHARACTERISTICS

3.1.1 Pump

Pump is a mechanical device, which transfers mechanical energy from some external sources to the liquid flowing through it. It operates by creating a pressure difference between the suction side and the delivery side of the moving element, which may be piston or impeller. Thus the pump increases the energy of the liquid that may be used to lift the liquid from low level to a high level or for delivering liquids from a region of low pressure to high pressure and to overcome the hydraulic resistance of the delivery pipeline. It is also used for pumping liquid from higher level to a lower level through a long pipeline of very high hydraulic resistance like oil pipelines.

There are various types of pumps and are classified on the basis of the application they serve, like: the material from which they are constructed, the liquids they handle, and even the orientation in space. But the more basic form of classification based on the principle by which energy is added to the fluid, goes on to identify the means by which this principle is implemented.

- *Dynamic*, in which energy is continuously added to increase the fluid velocities within the machine to value in excess of those occurring at the discharge such that subsequent velocity reduction within or beyond the pump produces a pressure.
- *Displacement,* in which energy is periodically added by application of force to one or more movable boundaries of any desired number of enclosed, fluid containing volumes, resulting in direct increase in pressure up to the value required moving the fluid through valves or ports into the discharge line.

The positive displacement pumps, working on the principle of displacing the fluid are not suitable for operating them as turbine and hence this discussion on pumps and their characteristics has been confined to rotodynamic pumps.

Rotodynamic pumps are divided into several varieties, but classification based on low specific speed pumps, medium specific speed pumps and high specific speed pumps and depending upon the direction of flow through the impeller of the pump, centrifugal pump (where flow is nearly radial), the propeller pumps (where flow is nearly axial) and the mixed flow pump (where the flow is characterized by radial and axial) have relevance to the present discussion.

3.1.2 Study of Characteristics of Rotodynamic Pumps

a) Ideal pump

An ideal impeller contains a very large numbers of vanes of infinitesimal thickness. The particles of an ideal fluid move exactly parallel to such vane surface without friction. An analysis of power transmitted to an ideal fluid by such an impeller leads to Euler's pump equation.

$$H_e = \frac{u_2 c_{u2}}{g} - \frac{u_1 c_{u1}}{g} \quad \text{..............................3.1}$$

Where,

H_e is Euler head and is the work done on a unit weight of the fluid by the impeller.

u_2 is the peripheral velocity of impeller at outlet

u_1 is the peripheral velocity of impeller at inlet

c_{u1} is component of absolute velocity of flow in impeller direction at inlet

c_{u2} is component of absolute velocity of flow in impeller direction at outlet

g is acceleration due to gravity

The second term in Euler's equation frequently is small compared to the first term and may be neglected so that The Euler equation becomes

$$H_e \cong \frac{u_2 c_{u2}}{g} \quad \text{......................................3.2}$$

Fig. 3.1 shows the peripheral velocity u of any point on a vane and w is the velocity of a fluid particle relative to the vane as seen by an observer attached to and moving with the vane. The absolute velocity of a fluid particle, c, is the vector sum of u

and w. The subscripts 1 and 2 refer to the entrance and exit cross sections of the passages.

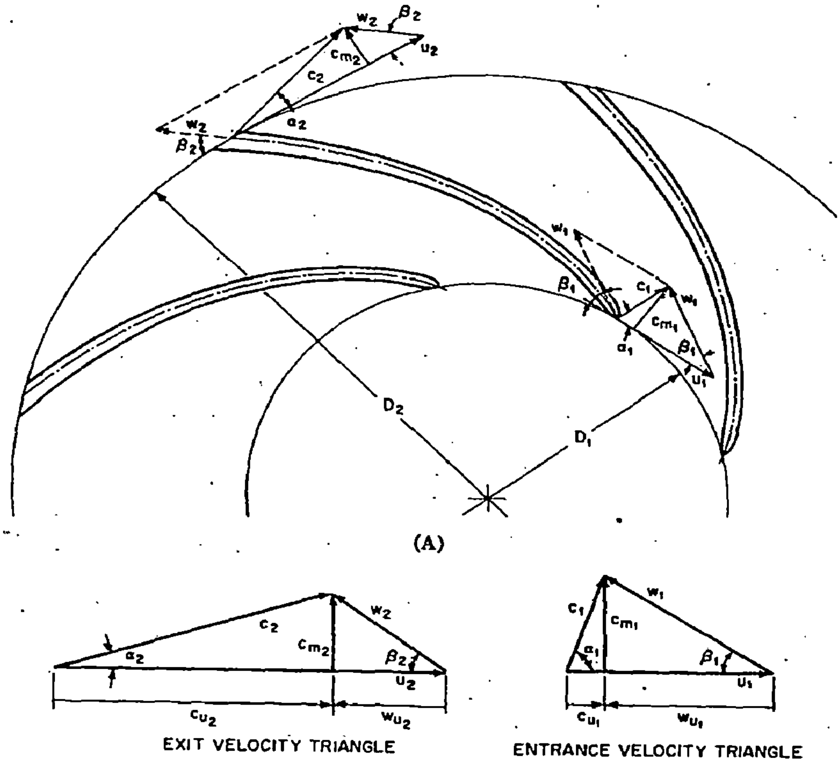


Fig. 3.1 Velocity diagram for radial flow impeller (Karassik) [6]

The absolute velocity c may be resolved into the meridian or radial velocity, c_m, and the peripheral velocity c_u. From the geometry of the figure

$$c_{u2} = u_2 - \frac{c_{m2}}{\tan \beta_2} \quad \text{.............................3.3}$$

thus,

$$H_e = \frac{u_2^2}{g} - \frac{u_2 c_{m2}}{g \tan \beta_2} \quad \text{............................3.4}$$

Neglecting leakage flow, the meridian velocity c_m must be proportional the capacity Q. With the additional assumption of constant impeller speed, the above equation (eq. no. 4) becomes

$$H_e = k_1 - k_2 Q \quad \text{....................................3.5}$$

In which k_1 and k_2 are constant with the value of k_2 dependent on the value of the vane angle β2 Fig. 3. 2 shows the Euler head–capacity characteristics for the three possible conditions on the vane angles at exit β2. The second term in equation no. 1 may be treated in like manner to the foregoing and included in equations no. 4 and 5.

The effect on Fig. 3. 2 would be to change the value of $H_e = \frac{u_2^2}{g}$ at $Q = 0$ and the slopes of the lines but all head-capacity characteristics would remain straight line.

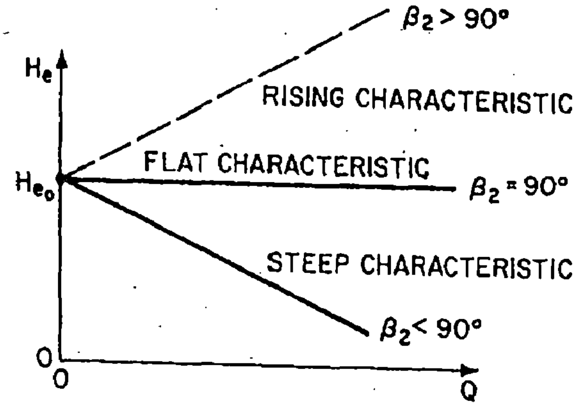


Fig. 3.2 Euler's head-capacity characteristics. (Karassik) [6]

b) Real pump

The vanes of real pumps impellers have finite thickness and are relatively widely spaced. Fluid does not flow parallel to the vane surface even at the point of the best efficiency so that the conditions required for Euler's equation are not fulfilled. The head-capacity curve of an idealized pump is though a straight line; the shape of the head-capacity curve of an actual pump is determined by 1) The Mechanical efficiency which accounts for the bearing, stuffing box, and all disk friction losses including wearing rings and balancing disk or drum if present. 2) The volumetric efficiency, accounts for leakage through the wearing rings, internal labyrinths, balancing devices, and glands. 3) The hydraulic efficiency, accounts for fluid friction losses in all through-flow passages, including the suction elbow or nozzle, impeller, diffusion vanes, volute casing, and the crossover passage of multistage pumps. The head is always less than predicted by Euler's equation and the head-capacity characteristics frequently in an irregular curve. Analyses so far have failed to predict the characteristics of real pumps with requisite accuracy so that graphical representation of actual test is in common use. Fig.3. 3 show an estimate of the losses from various sources.

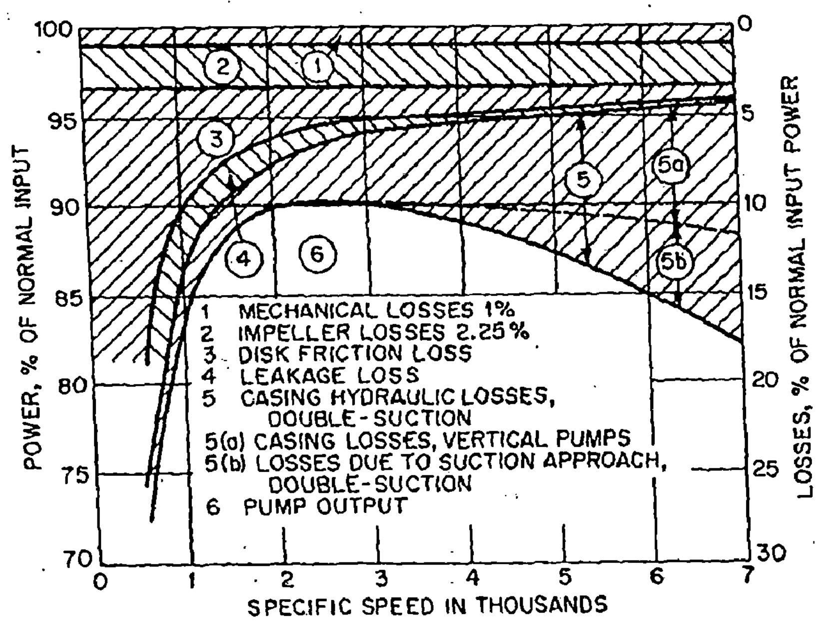


Fig. 3.3 Power balance for double suction pumps at best efficiency. (Stepanoff, A. J.) [17]

Fig. 3.4, 3.5 and 3.6 shows head-capacity, efficiency and brake horsepower curves normalized on the conditions of best efficiency and for wide range of specific speeds. These curves are applicable to pumps of any size because absolute magnitudes have been eliminated.

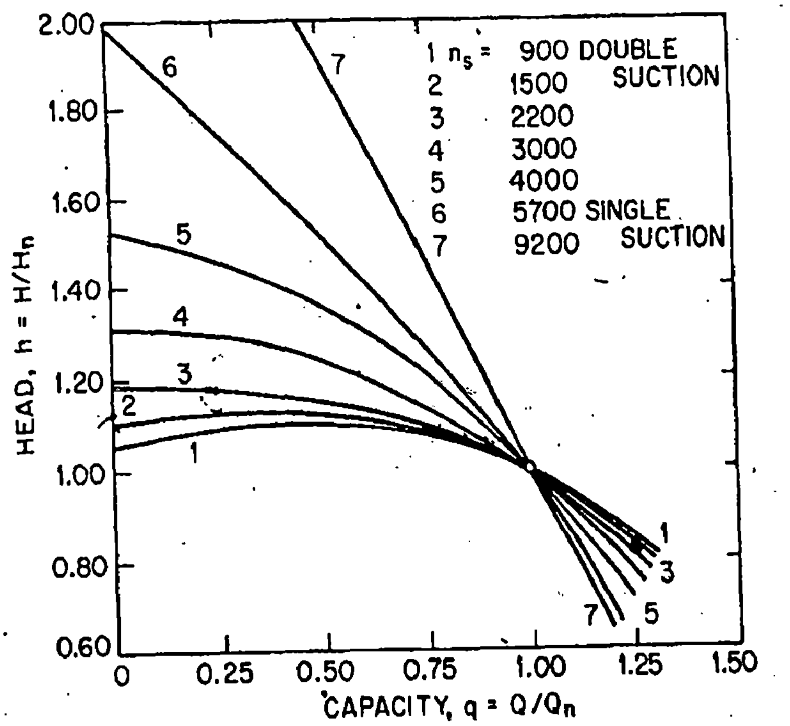


Fig. no. 3.4 Head-capacity curves for several specific speeds (FPS unit). (Stepanoff, A. J.) [17]

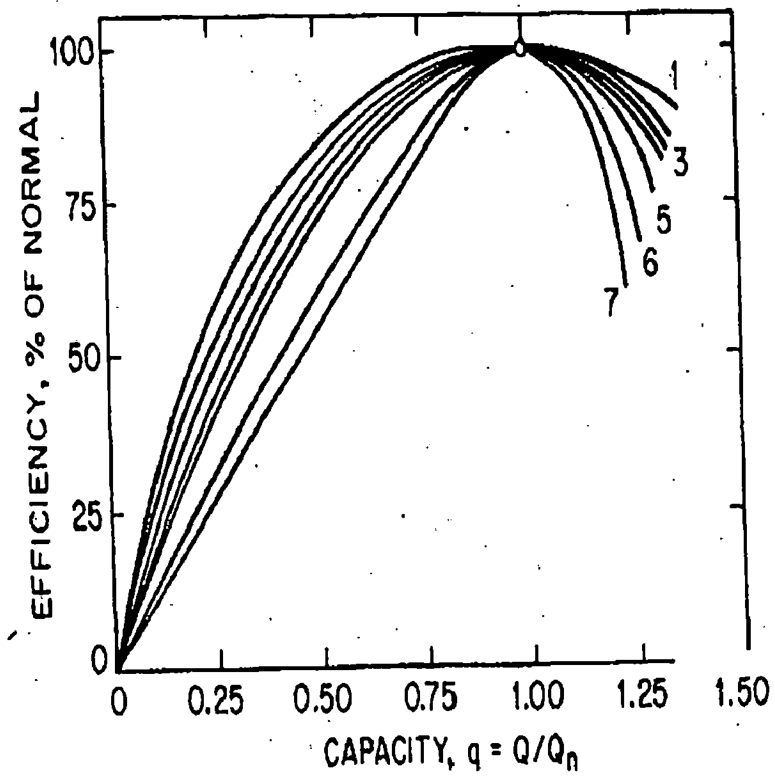


Fig. 3.5 Efficiency-capacity curves for several specific speeds(FPS unit). (Stepanoff, A. J.) [17]

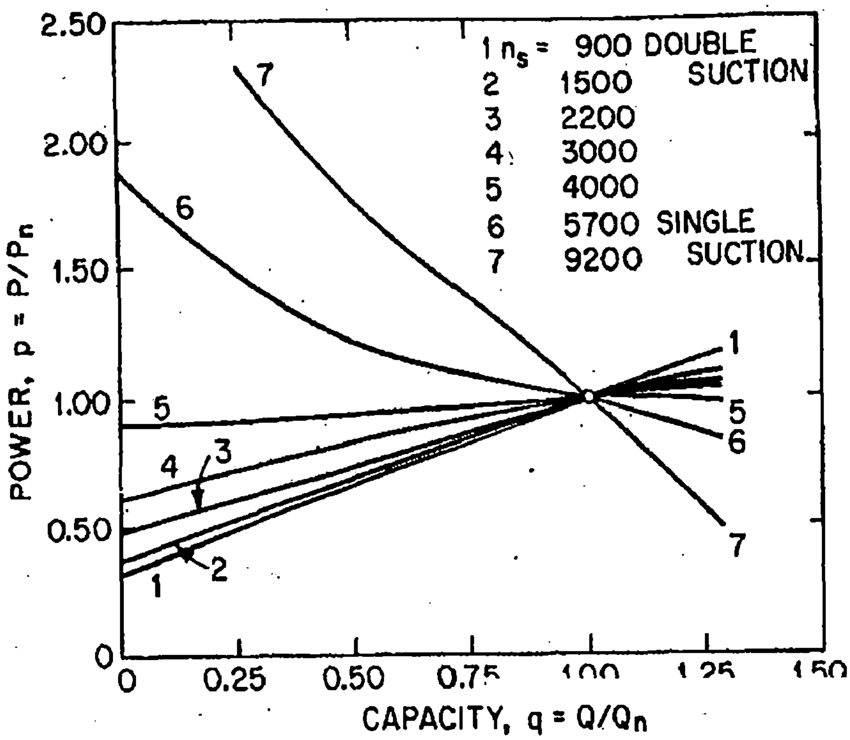


Fig. 3.6 Power-capacity curves for several specific speeds(FPS unit) (Stepanoff, A. J.) [17]

i) *Affinity Law*

The variation of head, capacity, and power with speed follows definite rules know as affinity laws which states when speed (N) is changed, capacity (Q) varies directly as the speed; the head (H) varies directly as the square of the speed and power (P) varies directly as the cube of the speed. The cube of the speed is based on the assumption that efficiency stays constant with speed for each point. The affinity laws are expressed by the following equations

$$\frac{N_1}{N_2} = \frac{Q_1}{Q_2} = \left(\frac{H_1}{H_2}\right)^{1/2} = \left(\frac{P_1}{P_2}\right)^{1/3} \quad \text{.............3.6}$$

ii) *Model law*

The variation of speed (N), capacity (Q), and power (P) with impeller diameter (D) of two geometrically similar pumps follows definite rules know as model laws which states at constant head, when impeller diameter is changed, capacity varies directly as square of the impeller diameter; the speed varies inversely as the impeller diameter, power varies directly as the square of the impeller diameter. The model laws are expressed by the following equations

$$\frac{N_2}{N_1} = \frac{D_1}{D_2} = \left(\frac{Q_1}{Q_2}\right)^{1/2} = \left(\frac{P_1}{P_2}\right)^{1/2} \quad \text{........................3.7}$$

c) Specific speed

Kinematic specific speed [5] of a rotodynamic pump is defined as the speed of homologous pump that delivers unit discharge (1 m^3/s) against unit head (1 m). It is expressed as

$$n_s = \frac{N\sqrt{Q}}{(H)^{\frac{3}{4}}} \quad \text{...3.8}$$

Dynamic specific speed [5] of a rotaodynamic pump is defined as the speed of homologous pump capable of raising .075m^3/s to a height of 1 meter absorbing a power of one metric horse power. It is expressed as

$$n_s = \frac{N\sqrt{P}}{(H)^{\frac{3}{4}}} \qquad \ldots\ldots 3.9$$

The relation between the kinematic specific speed and dynamic specific speed is Eq. 3.3

$$Dynamic\ specific\ speed(n_s) = 3.65N\frac{\sqrt{Q}}{H^{3/4}} \qquad \ldots\ldots 3.10$$

Specific speed as defined by ISI [5] is the speed expressed in rpm of an imaginary pump (proto model) geometrically similar in every respect to actual pump under consideration and which is capable of raising 75 kg (75 liters) of water per second to a height of 1 meter .The specific speed n_s is a useful dimensional parameter for classifying the overall geometry and performance characteristics of impellers. It can be show that in dimensionless form the specific speed n_s is given by equation

$$n_s = N\frac{\sqrt{Q}}{(gH)^{3/4}} \qquad \ldots\ldots 3.11$$

$$n_s = \frac{N\sqrt{P}}{(gH)^{\frac{5}{4}}} \qquad \ldots\ldots 3.12$$

When N is in rpm, Q in gpm, and H in feet (that is in FPS unit). Conversion to SI units may be accomplished by following relation.

$$n_s = 51.64rpm\frac{\sqrt{m^3/s}}{m^{3/4}} \qquad \ldots\ldots 3.13$$

Predication of pump behavior from theoretical consideration alone is impossible. Therefore, for solution of such problems one has to rely on experimental results. Knapp conducted several investigations at the California Institute of Technology.

Fig. 3. 7, Fig.3.8, and Fig. 3. 9 show the characteristics of pumps whose specific speed N_s=1800 (FPS unit), N_s=7500 (FPS unit) and N_s=13500 (FPS unit) obtained from complete characteristics curve in Fig.2.1, 2.2 and 2.3 respectively.

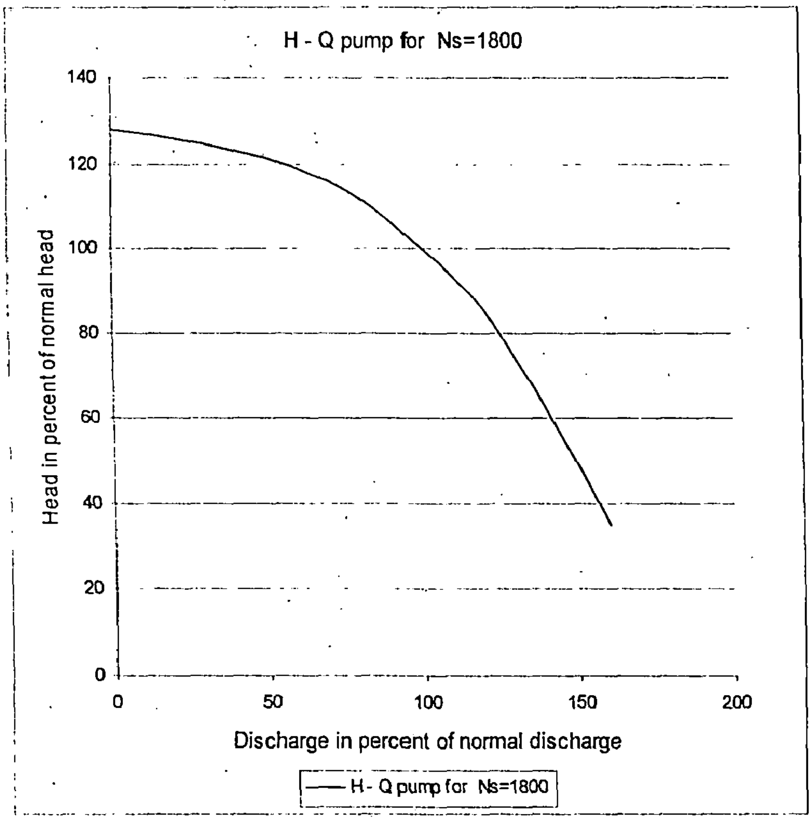


Fig. 3. 7 Head-capacity curve for Ns= 1800 pump (FPS unit). (Stepanoff, A. J.) [17]

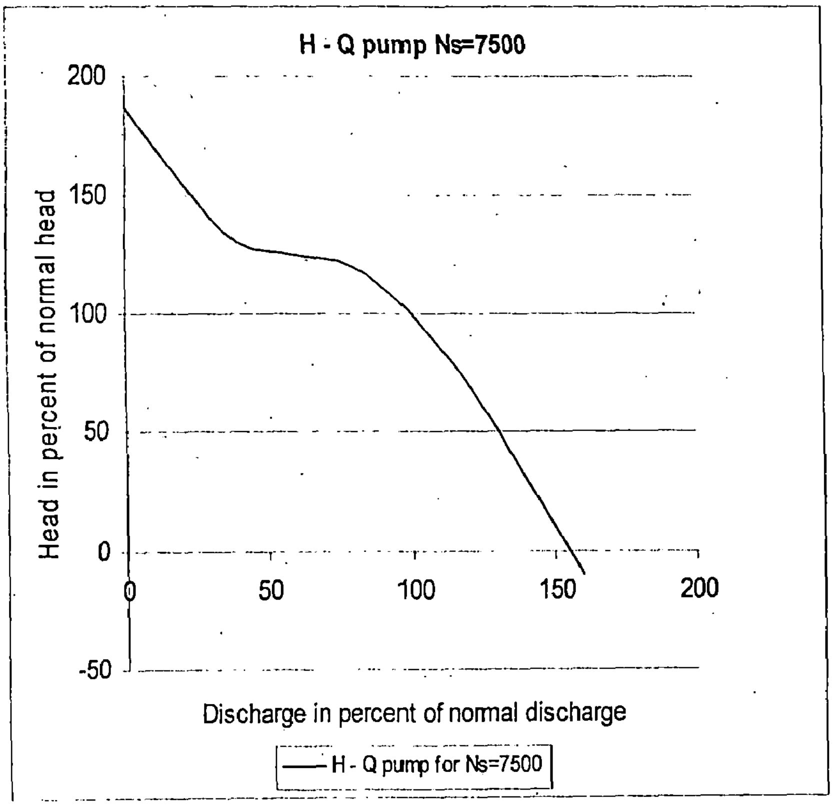


Fig. 3..8 Head-capacity curve for Ns= 7500 pump (FPS unit). (Stepanoff ,A. J.) [17]

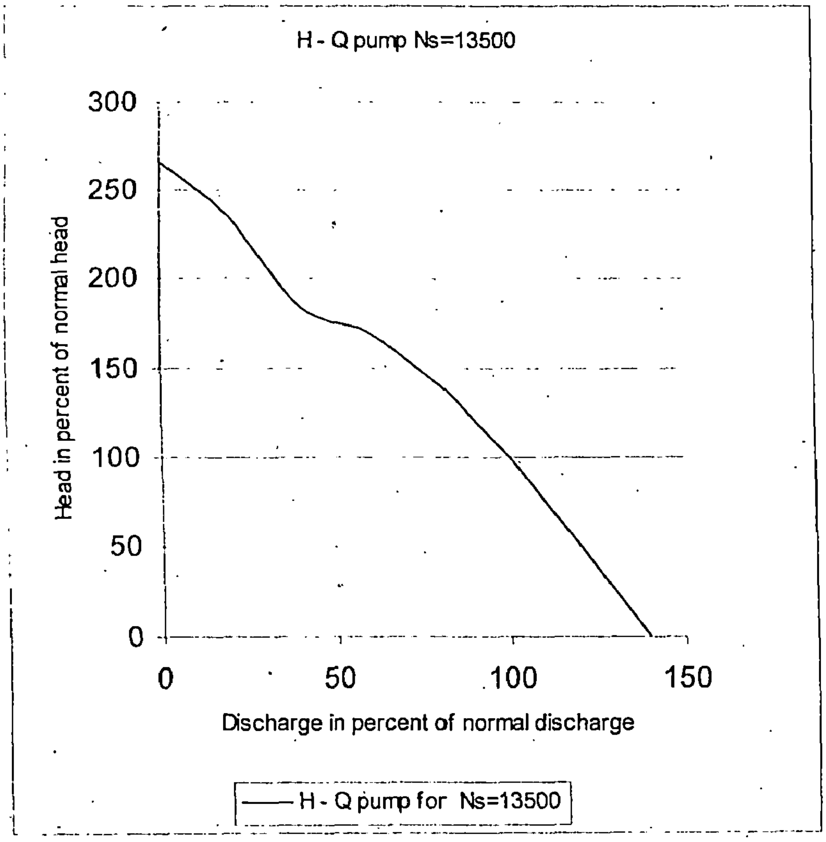


Fig. 3.9 Head-capacity curve for Ns= 13500 pump (FPS unit) (Stepanoff ,A. J) [17]

3.2 VALVES AND ITS CHARACTERISTICS

3.2.1 Valve

Valves are the components in a fluid flow or pressure system, which regulate either the flow or the pressure of the fluid. This duty may involve stopping and starting flow, controlling flow rate, diverting flow, preventing back flow, controlling pressure, or relieving pressure. These duties are performed by adjusting the position of closure member in the valve. This may be done either manually or automatically; manual operation includes operation of the valve by means of a manually controlled power operator. Manually operated valves are for stopping flow, controlling flow rate, and diverting flow; and automatically operated valves for preventing back flow and relieving pressure. The manually operated valves are referred to as manual valves, while valves for the prevention of back flow and relief of pressure are referred to as check valves and pressure relief valves, respectively.

Most valves consist of a body containing a flow control element (discs, plug, gates, etc) attached to and operated by rotation of a stem. The stem together with any stem seals is enclosed within a bonnet. The top of the stem is fitted with a hand wheel (or lever) for rotation of the stem and some stems may have a sliding operation for quick action.

With threaded stems, the threaded portion may be fully enclosed by the bonnet, known as inside-screw; or exposed beyond the bonnet, known as outside-screw. The former obviously provides maximum protection for the screw thread. Outside screws have the advantage of being easier to lubricate.

With rising-stem valves, the hand wheel and stem together, give a visual indication of the degree of valve opening. With a non-rising stem the hand wheel does not rise or fall with the turning movement. The advantage of this type is that it can be installed in situation providing only minimum headroom above the hand wheel.

Various types of bonnet may be used- e.g. screw-in, screw-on, union style and bolted or flanged bonnet. Screw-in or screw-on bonnets are the simplest and cheapest, but largely limited to smaller valve used on low-pressure services. Union bonnets generally provide tighter sealing and are particularly suitable for valves that are dismantled frequently for servicing. Plain flange and male and female flanged bonnets are generally preferred for high temperature or high-pressure valves, and also for large sizes valves. An alternative type for high-pressure and/ or high temperature services is the breech-lock bonnet [18].

While testing the performance of pumps, a valve or valves in the discharge line of a centrifugal pump control/s the pump flow and consequently alters the variable friction–head portion of the total system head curve. Fig. 3.10 illustrates the use of a discharge valve to change the system head for the purpose of varying the pump flow during shop performance test.

The maximum flow is obtained with a completely open valves and the only resistance to flow is the friction in the pipes, fittings, and flow meter. A closed valve results in the pump operation at shutoff conditions and produces maximum head. Any flow between maximum and shutoff can be obtained by proper adjustment of the valve opening.

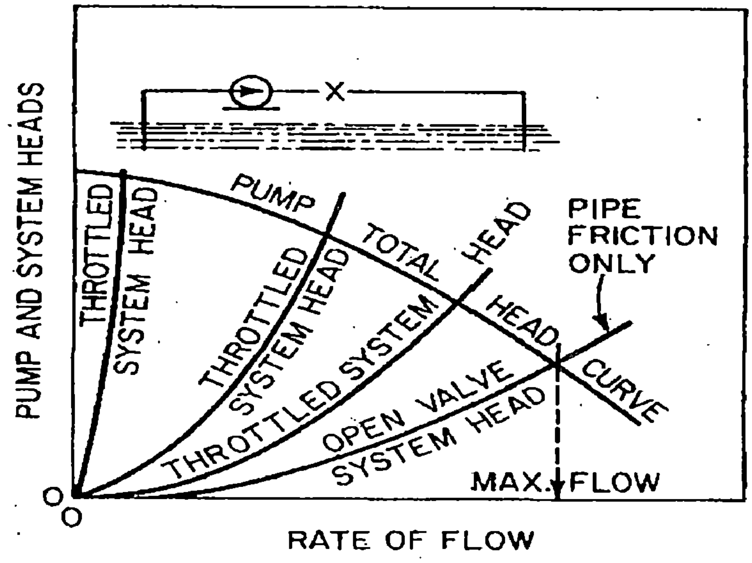


Fig. 3.10 Construction of system total head curves for various valves openings (Karassik) [6]

Throttling the inlet line to the pump will reduce inlet head and cause the pump curve to start at the same point but slope downward at a faster rate.

3.2.2 Study of Characteristics of Valves

The flow characteristics express the way in which the flow through the valve depends on percentage of valve stem travel. The latter may be translatory or rotary motion. A plot of percentage of maximum flow at various percentage of stem travel is the usual quantitative way of showing a characteristic.

The linear characteristic is a straight line, with flow percentage always equal to stem-travel percentage. A quick-opening characteristic, on the other hand, produces proportionately more flow in the early stages of stem travel. An equal percentage characteristic gives a change that, for a given percentage of lift, is a constant percentage of the flow before the change.

In linear characteristic, the rate of flow change is uniform for a given stem-travel change. But incorporation of the valve into a piping system affects the overall characteristic. Since resistance to flow in a given piping system is roughly proportional to the square of the flow rate, the curve of the piping system head loss plotted against flow rate will be a parabola, with resistance increasing at a faster rate than flow. If the piping system and valve are consider together, with the flow rate in the system plotted against percentage of valve-stem travel, the overall system characteristics will differ from the valve characteristic. The overall system characteristic is displaced upward toward the quick-opening valve characteristic but can have point of flexure. The amount of displacement depends on what part of the total system pressure drop is taken by the valve. Only with a very short outlet pipe would the valve take the entire pressure drop, and then its characteristic would be that of the valve.

In many systems the pressure drop across the valve is designed to be from a tenth to a third of the total system drop. If the valve in such a system has an equal-percent characteristic, then the characteristic of the overall system will be close to linear as far as the actuator of the valve is concerned. Valves with characteristic between linear and equal percentage are also useful in modulating control. Ball, plug, and butterfly valve are examples of valve having characteristic between linear and equal percentage. Butterfly valves are used in pump testing. Fig. 3.11 shows the different flow characteristics.

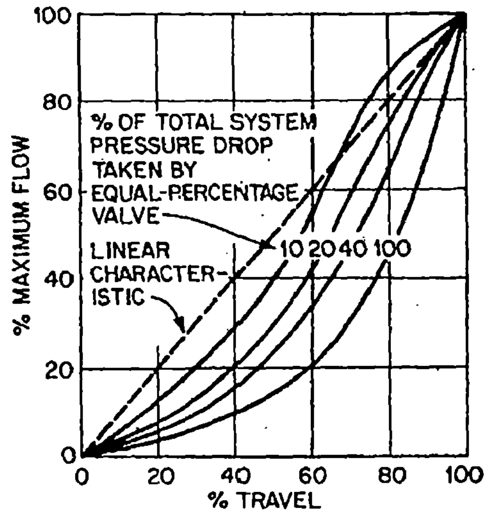


Fig. 3.11 Different types of valve characteristics (Karassik) [6]

a) Flow through valves

Valves may be regarded as analogous to control orifices in which the area of opening is readily adjustable. As such, the friction loss across the valve varies with flow, as expressed by the general relationship:

$v \propto (g\Delta h)^{1/2}$ 3.14

$v \propto \left(\frac{g}{\gamma}\Delta p\right)^{1/2}$ 3.15

where,

v is flow velocity (m/s in SI unit)

Δh is head loss is in (m in SI unit)

Δp is pressure loss (N/m^2 in SI unit)

For any valve position, numerous relationship between flow and flow resistance have been established, using experimentally determines resistance or flow parameters. Common parameters so determined are the resistance coefficient K and, dependent on the system of units, the flow coefficients C_v, K_v and A_v. It is standard practice, to base these parameters on the nominal valve size.

b) Resistance coefficient

The resistance coefficient K defines the friction loss attributable to a valve in a pipeline in terms of the velocity head or velocity pressure, as expressed by the equations.

$\Delta h = K\frac{v^2}{2g}$ (Coherent unit)..............................3.16

and

$\Delta p = K\frac{v^2\rho}{2}$ (Coherent unit)..............................3.17

where,

ρ is density of fluid

g is acceleration due to gravity

Δh is head loss

Δp is pressure loss The above equation is valid for single-phase flow of Newtonian liquids and for both turbulent and laminar flow conditions.

The value of the pressure-loss coefficient of valve varies with the type of valve and, with most types, also with size and make. The following are approximate mean range of pressure-loss coefficients of fully open valves for conditions of fully turbulent flow used in test of pumps.

Gate valve, full bore: $K = 0.1 - 0.3$

Butterfly valve, dependent on blade thickness: $K = 0.2 - 1.5$

As flow is modulated by using valve which is achieved by translatory or rotary motion of valve stem. The pressure-loss coefficient for partly open gate and butterfly valves may be found by multiplying the K-values of the fully open valves with a factor K_1, which may be obtained from Fig. 3.12 and 3.13.

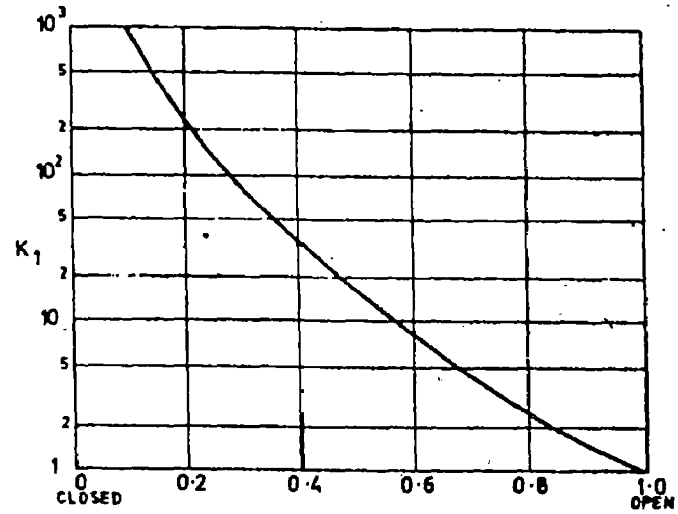


Fig. 3.12 Approximate effect of partial opening on pressure-loss coefficient of gate valves (Zappe R. W.) [19]

The relation between the pressure-loss coefficient and the valve opening position represents the relationship between flow through the valve and the valve opening position on the basis of constant pressure loss in the valve. The flow characteristic based on this relationship is referred to as inherent flow characteristic. In the case of flow-control valves, the valve closure member and/or the valve-seat orifice are frequently shaped to achieve a particular inherent flow characteristic. Fig. 3.14 highlights some of the more common flow characteristics like: Quick opening (1), Square Root (2); Linear and (3) and equal Percentage (4).

In most practical applications, however, the pressure loss across the valve varies with the valve opening position. The pressure drop across the installed valve diminished with rising flow rate, the required rate of valve opening with rising flow rate is higher than indicated by the inherent flow characteristic. This is demonstrated for a pumping system shown in Fig. 3.15. The lower portion of the figure shows the flow rate versus valve opening position. The latter flow characteristic is referred to as installed flow characteristic and it is unique for each valve instillation.

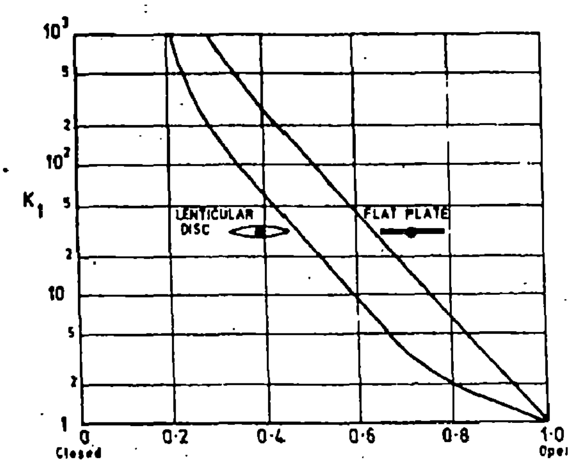


Fig. 3.13 Approximate effect of partial opening on resistance coefficient of butterfly valves (Zappe R. W) [19]

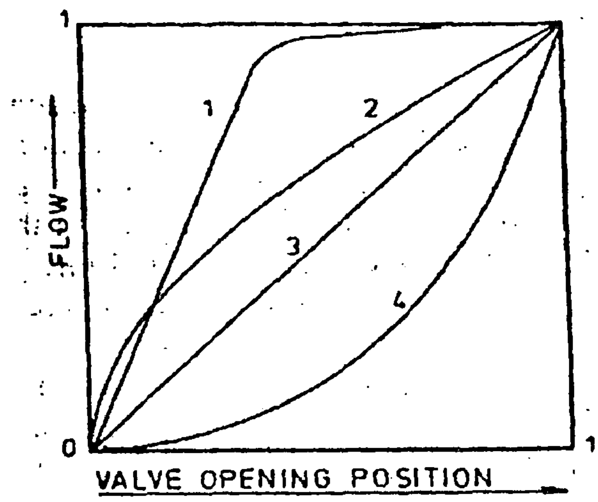


Fig. 3.14 Inherent flow characteristics of valves (Zappe R. W) [19]

Valves are used in testing the performance of pumps; valve regulates the flow resulting in pressure or head developed by pump in upstream side of the valve. Head developed by pump due to valve closure can be found by deducting system head loss form the head developed by pump at that discharge. Fig. 3.16 shows the relation between head developed by pump due to valve closure, fixed head and system head with flow.

Fig. 3.17 is obtained by deducting system head loss and fixed head from the h-q curve of pump. The curve shows the relation between head developed by pump due to valve resistance or head lost in it at various discharge.

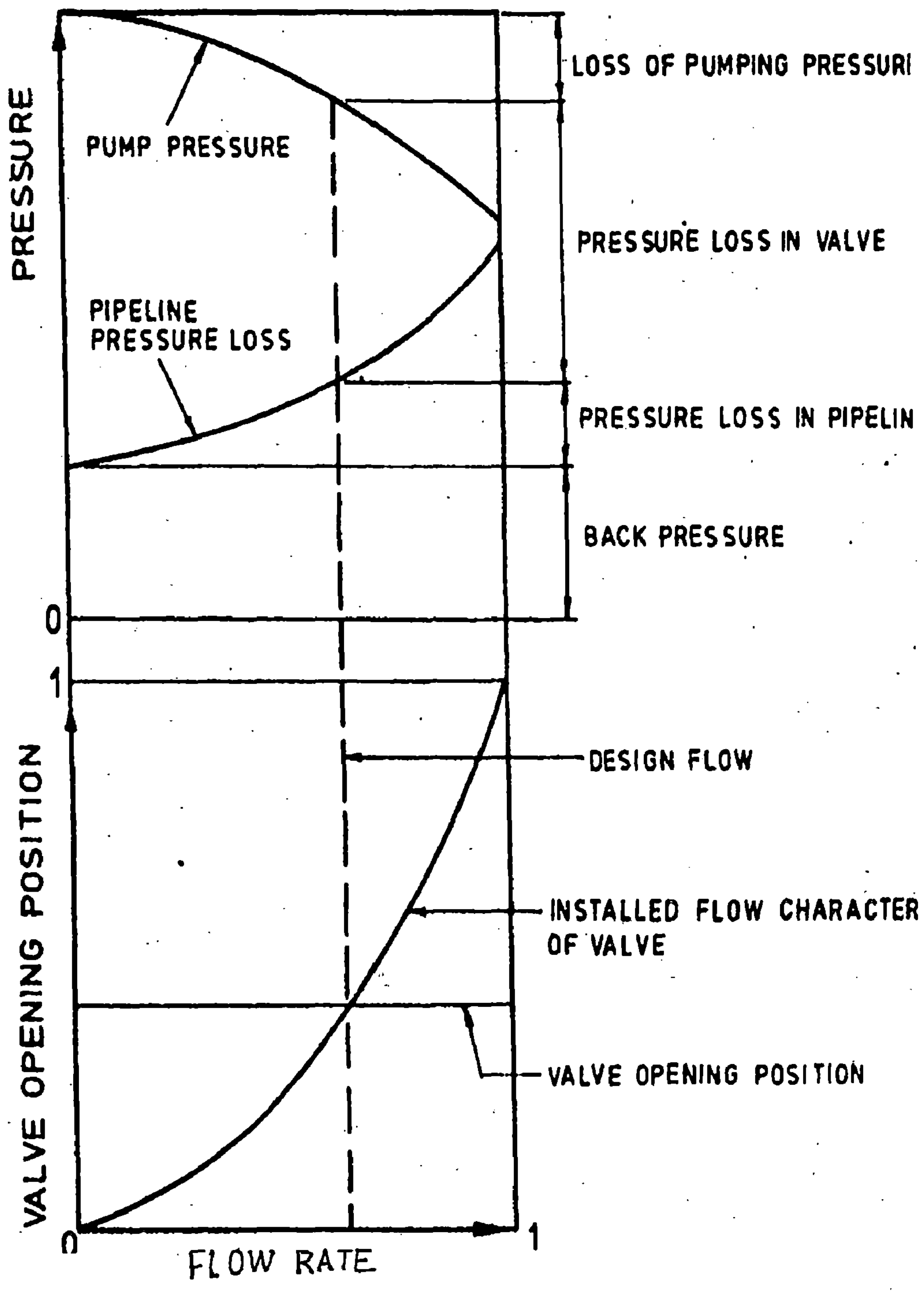


Fig. 3.15 Relationship between flow rate, valve opening position, and pressure loss in pumping system (Zapper. R. W.) [19]

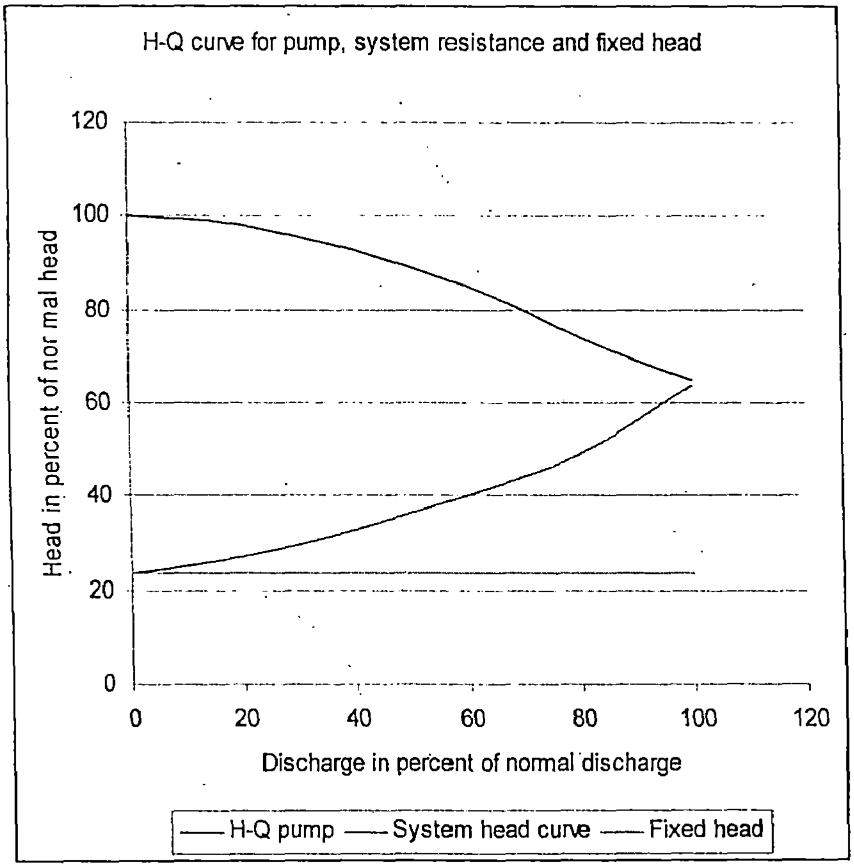


Fig. 3.16 H-Q curve of pump, system resistance and fixed head (Zapper. R. W.) [19]

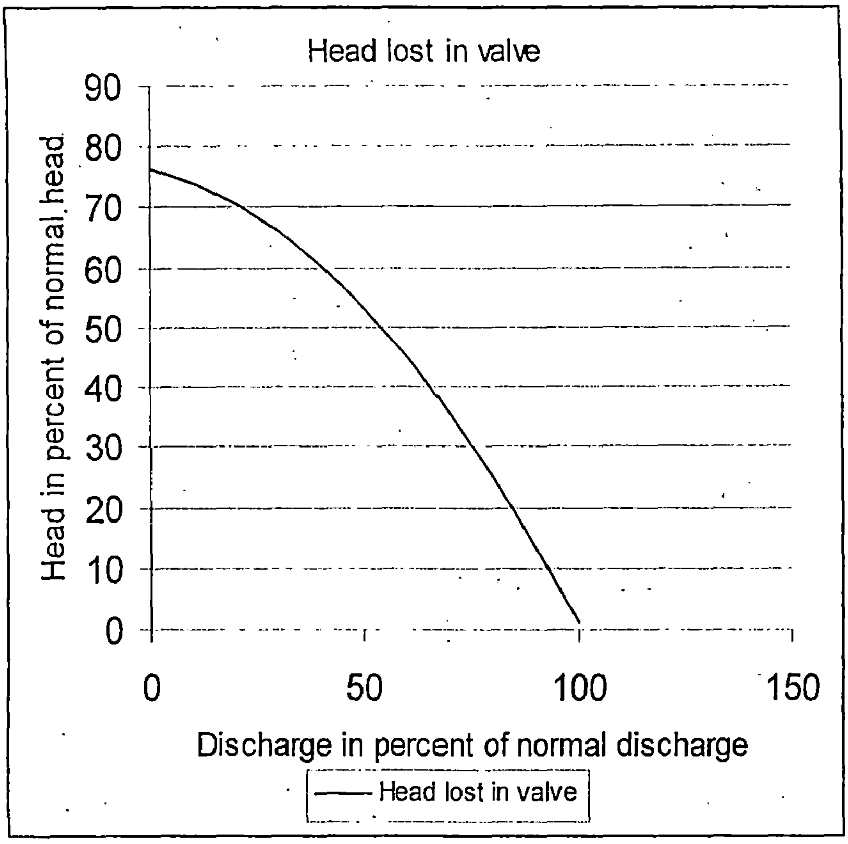


Fig. 3.17 Head lost by valve at various flow rates

3.3 TURBINES AND ITS CHARACTERISTICS

3.3.1 Turbine

Turbines are hydraulic machines that convert hydraulic energy into mechanical energy. Turbines are operated under variable head and flow conditions, flow though the turbine is adjusted to either accommodate to seasonal variation of the available water or to adjust power output according to the demand of the consumers. Adjustable guide vanes and/or runner blades regulate the flow. Unlike pumps, flow through a turbine is accelerated which is less subjected to turbulence; runner passages are therefore relatively short which reduces friction losses and ensures high efficiencies.The basic hydraulic theory is the same for rotational fluid machines (pumps and turbines).

Like pumps, turbines are also classified in different ways, like:

a) Action of water on runner blades

- **Reaction:** In reaction turbine the wheel passages are completely filled with water under a pressure that varies throughout the flow. The energy of the water leaving the stationary guide vanes and entering the runner is partly pressure energy and partly kinetic energy. During flow through the wheel both the pressure and the absolute velocity of the water are reduced as the water gives up its energy to the wheel.
- **Impulse turbine:** In the impulse turbine the wheel passages are never completely filled with water. Throughout the flow the water is under atmospheric pressure. The energy of the water leaving the stationary guides and entering the runner is all kinetic. During flow through the wheel the absolute velocity of the water is reduced as the water gives up its kinetic energy to the wheel. This type of turbine is more commonly known as the Pelton wheel.

Rotodynamic pump when operated as turbine acts as reaction turbine hence this discussion on turbines and their characteristics has been confined to reaction turbines.

b) Direction of flow in the runner

- **Radial flow:** Radial flow means that the path of a particle of water as it flows through the runner lies in a plane, which is perpendicular to the axis of rotation. If the water enters at the inner circumference of the runner and discharges at the outer circumference, the turbine is called as outward flow turbine known as the Fourneyron turbine [4].

  If the water enters at the outer circumference of the runner and discharges at the inner circumference, the turbine is called as inward flow and is the original Francis turbine [4].
- **Axial flow:** If a particle of water remains at a constant distance from the axis of rotation as it flows through the runner, the turbine is known as axial or parallel flow turbine. The type of turbine falling in this class is commonly called the Jonval turbine [4].
- **Mixed flow:** If the water enters a wheel radially inward and then during its flow through the runner turns and discharges axially, the turbine is called mixed flow type. This type of turbine is called a Francis turbine, though it is not identical with the one built by Francis [4].

c) Position of shaft

The distinction of turbine into vertical shaft and horizontal shaft is due to position of shaft. The vertical shaft turbine is used where it is necessary to set the turbine drown by the water while the generator or other machinery that it drives must be above. The horizontal shaft turbine is used where the turbine can be set above the tail water level and if the generator or other machinery that it drives can be set at the same elevation.

3.3.2 Study of Characteristics of Turbines

Like pumps, the Euler equation equally applies to turbines. Since the flow pattern in a Francis turbine is basically the reverse of that in a centrifugal pump, the same parameters determine the shape of the velocity diagrams, hence turbine performance.

Fig. 3.18 shows that the flow approach the runner, the shape of the velocity diagram is determined by the guide vane angle α_2 but not by runner blade angle β_2. As the flow leaves, the shape of velocity diagram at the inner periphery of the runner is mainly by

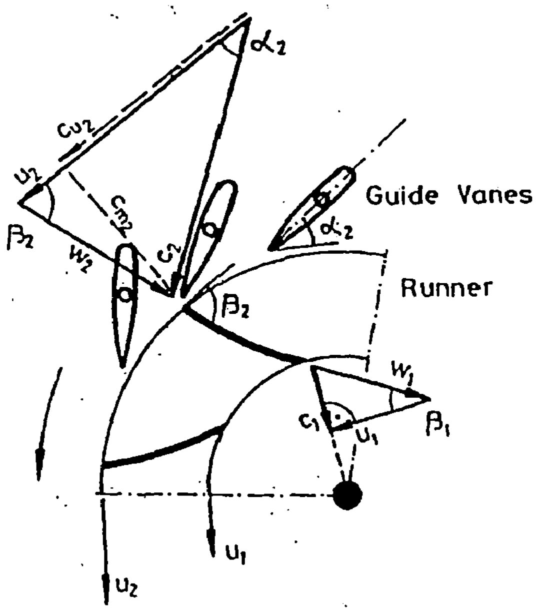


Fig. 3.18 Vector diagrams for Francis turbine (Chapallaz, J. M.) [2]

the blade angle β_1. Similar to pumps, the concept of conversion of the moment of momentum is applied on all torques acting on the turbine runner. The tangential components of the inlet and outlet velocities c_{u2} and c_{u1} contribute to the rate of change of momentum that constitutes the torque T acting on the turbine shaft. All other forces and momentums act radially on the runner and do not produce toque.

$T = \rho Q(c_{u2}r_2 - c_{u1}r_1)$ 3.18

Power (P) = $T\omega = \rho\, g\, H\, Q$ 3.19

Multiplying eq no. 3.19 by ω and equating with eq. no. 3.19 we get,

$$T\omega = \rho Q(c_{u2}r_2 - c_{u1}r_1)\omega$$

$$\rho\, g\, H\, Q = \rho Q(c_{u2}r_2 - c_{u1}r_1)\omega$$

$$H = \frac{1}{g}(c_{u2}u_2 - c_{u1}u_1) \text{ ....3.20}$$

Considering whirl free exit of flow, then equation 3.20 becomes

$$H = \frac{1}{g}(c_{u2}u_2) \text{ ....3.21}$$

Where,

ω is angular velocity (rad/s)

ρ is density of fluid (kg/m3)

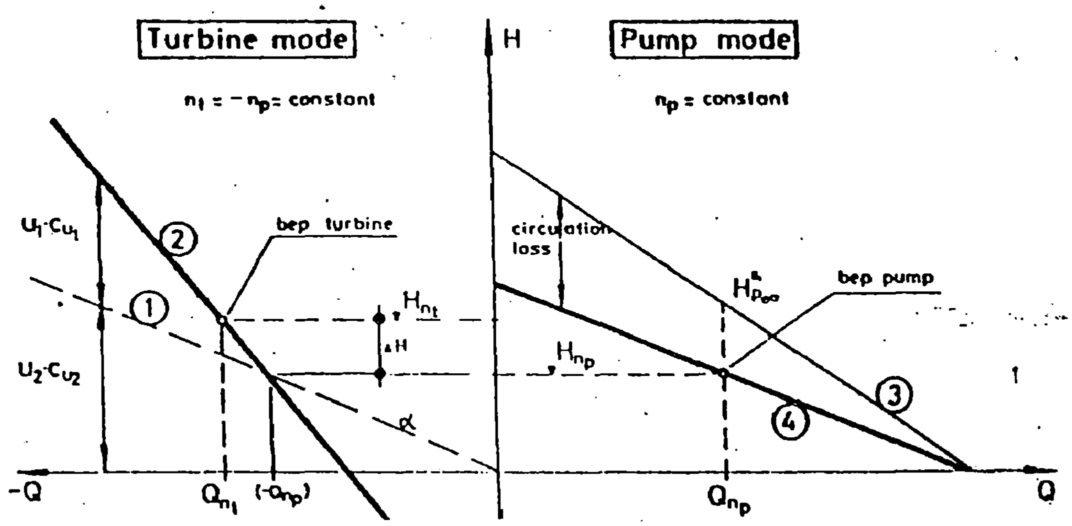
c_{u1} is component of absolute velocity of flow in runner direction at outlet (m/s)

c_{u2} is component of absolute velocity of flow in runner direction at inlet (m/s)

r_1 is radius of runner at outlet (m)

r_2 is radius of runner at inlet (m)

g is acceleration due to gravity (m/s^2)



(1) Euler line considering inlet triangle only u2*cu2
(2) Euler line considering inlet and outlet triangles (u2*cu2-u1*cu1)
(3) Ideal Euler line with infinite number of impeller vanes
(4) Euler line with finite number of impeller vanes including circulation losses

Fig. 3.19 Friction less and shock less performance of turbine and pump (Chapallaz, J. M.) [2]

Appling the Euler equation for ideal machines, we can draw ideal performance curves in both pump and turbine modes. For ideal conditions, the design flow and design head in both pump and turbine mode are identical. Fig. 3.19 shows the ideal performance curves for both pump and turbine modes.However, for real fluids and machines, the following two effects are considered:

1. geometry of the pump or turbine
2. hydraulic losses of the real fluid

The behavior for real fluid flow including friction and turbulence results in different rules for the design of pumps and turbines thus the performances in both modes are not identical although the theory of ideal fluids would predict the same.

When pump is operated as turbine, circulation loss that occurs at the inner periphery of the impeller at the suction side is practically negligible due to the smaller diameter of the impeller at the suction section side. This frictionless performance in turbine mode therefore corresponds to the ideal Euler condition and both head and flow will be bigger than in pump mode. This phenomenon is geometric effect since it is basically caused by the different geometric parameters determining the energy transfer between the fluid and the impeller of a machine. This increase in flow will affect the outlet triangle and outlet flow and the flow is no longer whirl free since the vane angle at the suction side is not designed for higher flow

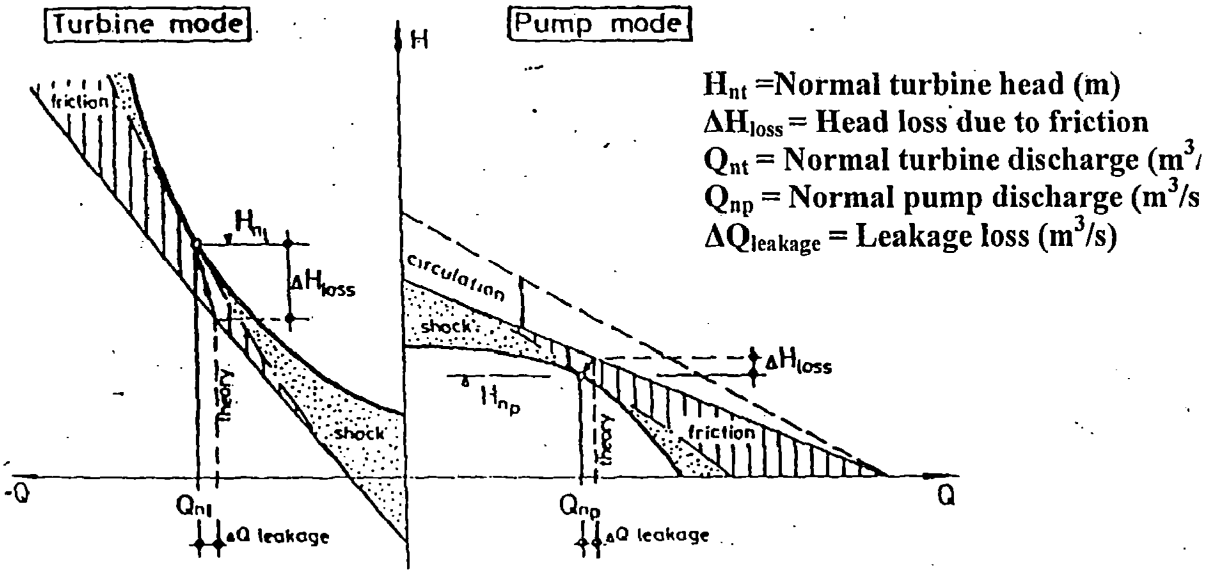


Fig. 3.20 Performance of turbine and pump mode including hydraulic losses (Chapallaz, J. M.) [2]

As the fluid passes through the impeller, it is subjected to friction and shock loss. Due to these losses, the ideal energy transfer form the fluid to the rotating impeller as by the Euler equation is not achieved. Therefore, friction and shock losses must be added to the ideal head according to Euler. Fig. 3.20 shows the performance of turbine and pump mode including hydraulic losses.

The performance of a pump or a turbine is usually presented in head versus flow diagram. Pump and turbine mode performance can be plotted in a single diagram by extending the flow axis into negative values representing the reverse operation of the pump, i.e., the turbine mode performance. These diagrams are usually referred to as complete or total characteristics of a machine. Fig. 3.21 shows the total head-flow characteristics of a pump and pump as turbine of the same speed in pump and turbine operation. Unlike pump, turbine flow increases continuously with increase in head. These curves are compatible to those in Fig. 3.22 by plotting discharge in negative side.

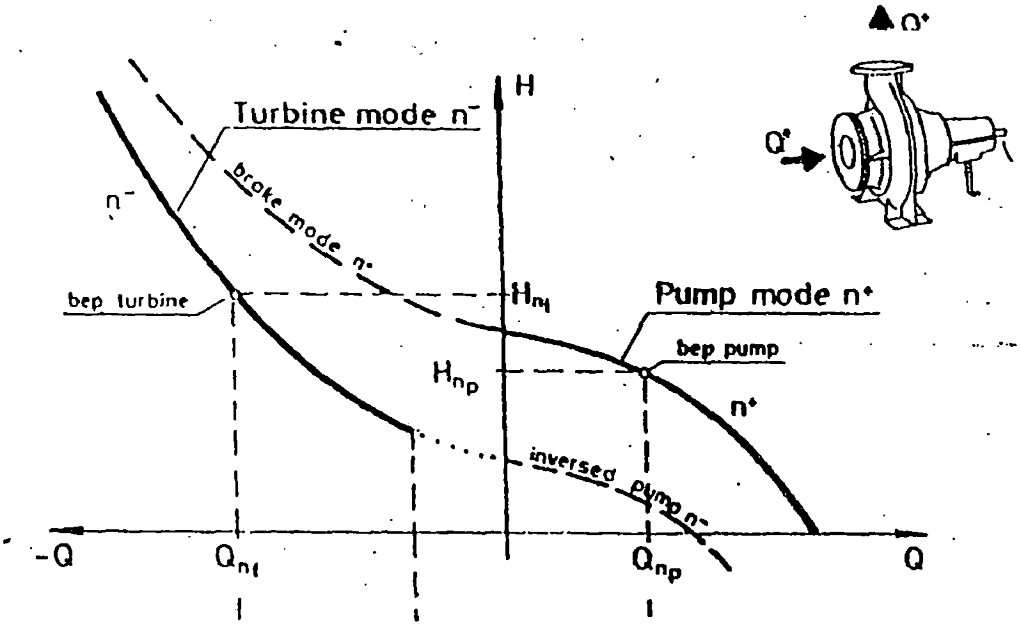


Fig. 3.21 H-Q curve for turbine and pump mode (Chapallaz, J. M.) [2]

Fig. 3.22, 3.232 and 3.24 show h-q characteristics of pump of specific speed 1800(FPS unit), 7500 (FPS unit) and 13500 (FPS unit) respectively in turbine mode obtained from complete characteristics curve in Fig.2.1, 2.2 and 2.3 respectively.

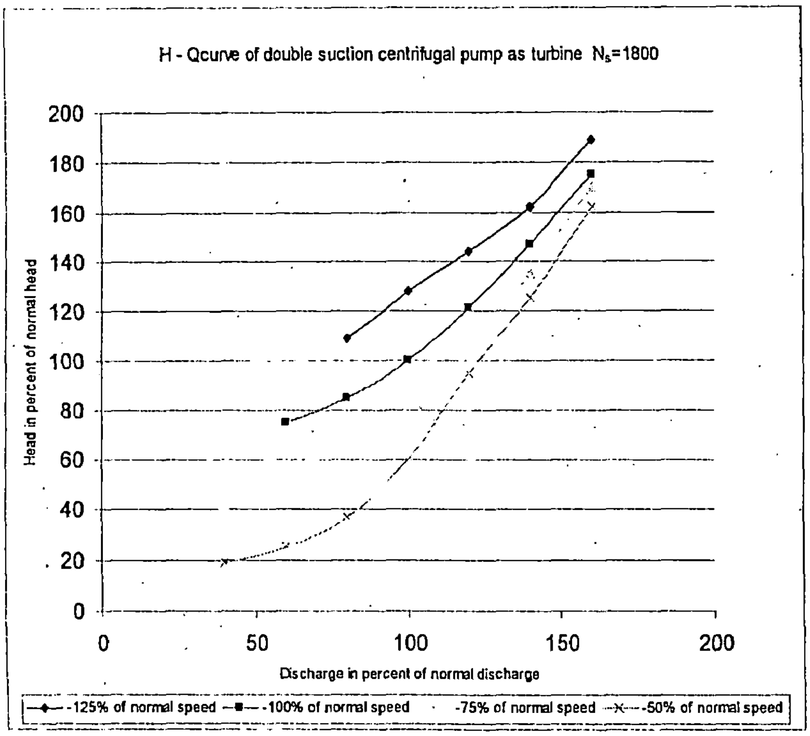


Fig. 3.22 H-Q curve Ns-1800 (FPS unit) of pump as turbine at various speeds (Stepanoff, A. J.) [17]

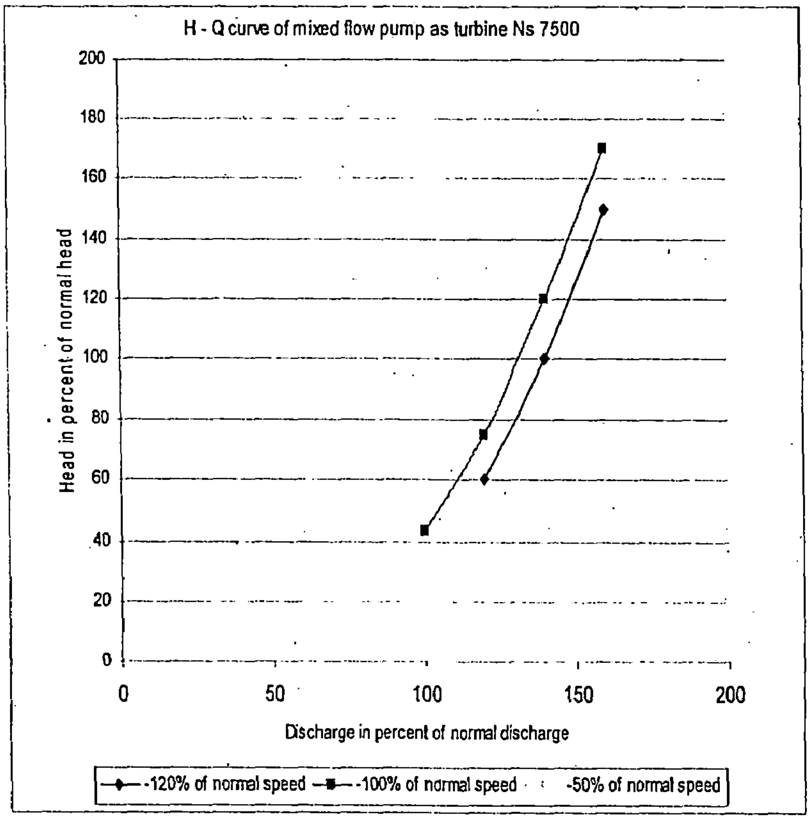


Fig.. 3.23 H-Q curve for Ns-7500 (FPS unit) of pump as turbine at various speeds (Stepanoff, A. J.) [17]

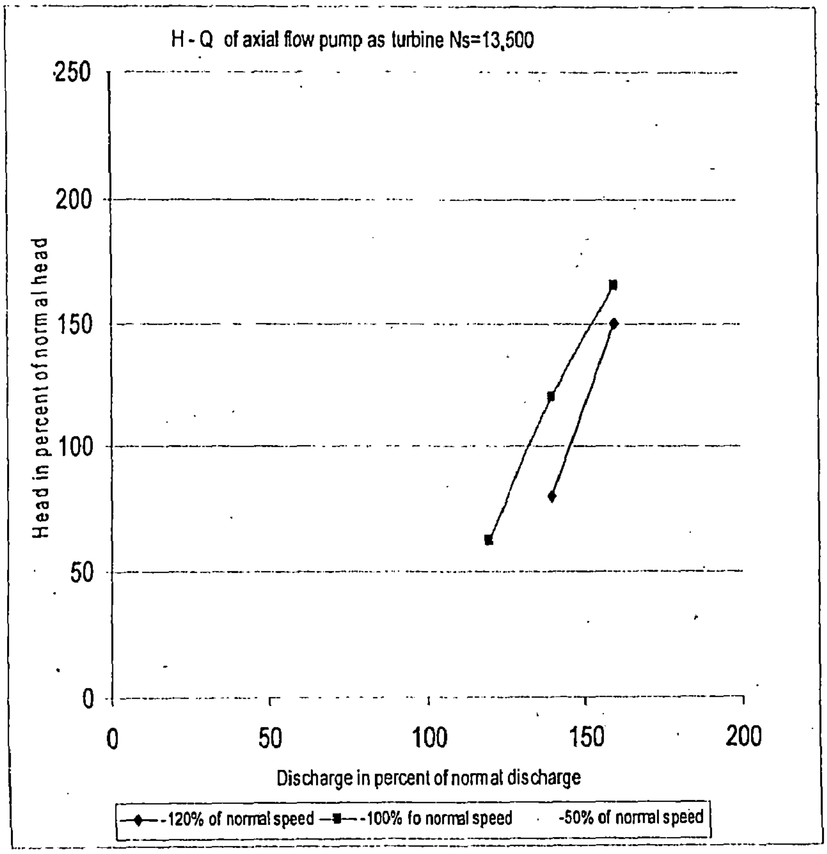


Fig. 3.24 H-Q curve for Ns-13500 (FPS unit) pump as turbine at various speeds (Stepanoff, A. J.) [17]

3.4 REPLICATION OF VALVE CHARACTERISTICS BY TURBINE

Turbines or pump as turbine, as energy recovery device, has high potential where high-pressure liquids are reduced across a pressure-regulating valve or orifice. Study of characteristics of pump for constant speed impeller shows the decrease of head with increase of discharge. In case of valve, pressure developed or head develop by pump in upstream side of valve (due to of closure member) increases with decrease of flow through it and vice-versa. Movement of stem of valve results in alteration of the variable friction head portion of the total system head curve. Flow between maximum and shutoff is obtained when valve stem is in between completely open or closed position respectively. Flow through the valve is regulated by translatory or rotary motion of valve stem. For turbine at particular runner speed, head increases with increase in discharge. It is also seen that when speed of runner of turbine changes, head-discharge curve get shifted. Like valve generated h-q curve, head increase with flow for turbines for different speeds it shifted like h-q valve at different valve closure positions. Fig. 3.25 shows the valve and turbines h-q characteristics for different valve closure positions and turbine speed respectively.

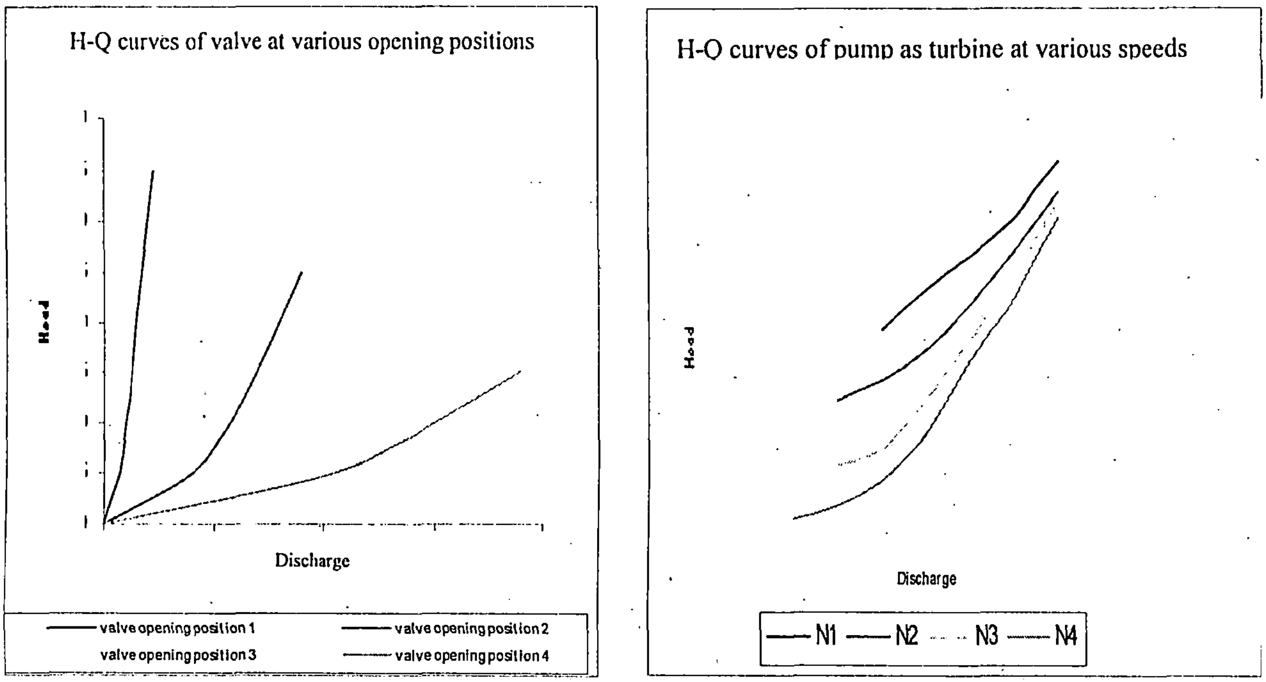


Fig.. 3.25 H-Q of valve at different openings and turbines in various speeds

The comparison between h-q of valve and turbine shows the turbine can replicate valve h-q characteristics at certain extent. Thus, turbines in a large number in parallel if used, can be used as valve in pump testing and energy can be recovered. The range of speed variation is limited from no speed that is when the shaft of turbine is completely locked, to no torque that is, when the turbine is running at run away speed and no power is extracted from it.

Curves in Fig. 3.26, 3.27 and 3.28 respectively show the change in head and discharge that can be achieved by varying speed from no speed to run away speed for pumps operating as turbine of specific speed (FPS unit) of 1800, 7500 and 13500 which is obtained from complete characteristics curve in Fig.2.1, 2.2 and 2.3 respectively.

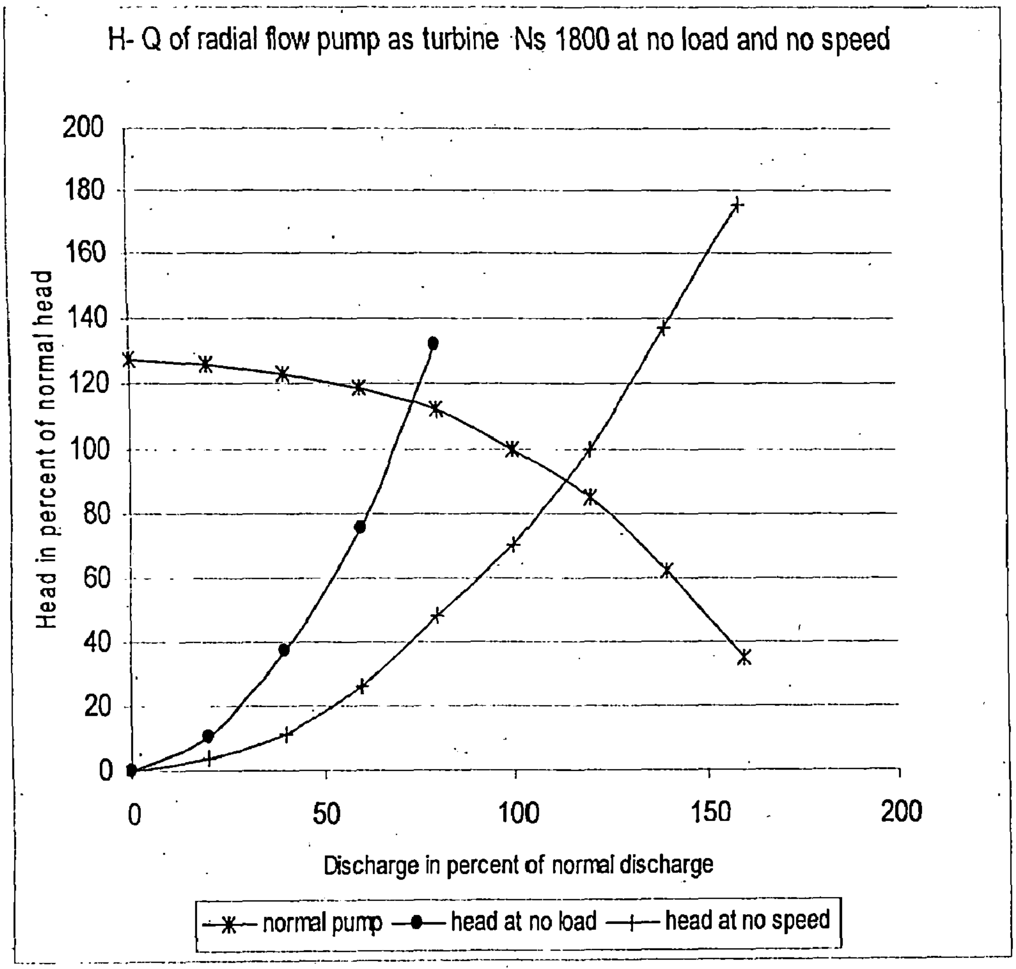


Fig. 3.26 Working range of pump as turbine (Stepanoff, A. J.) [17]

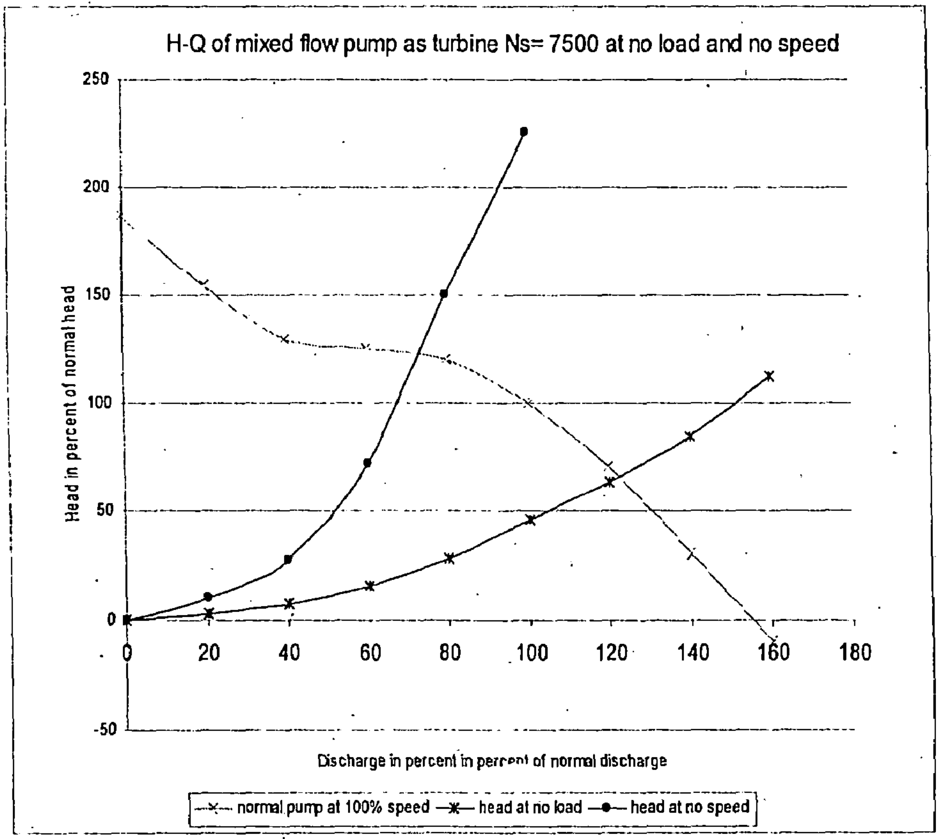


Fig. 3.27 Working range of pump as turbine (Stepanoff, A. J.) [17]

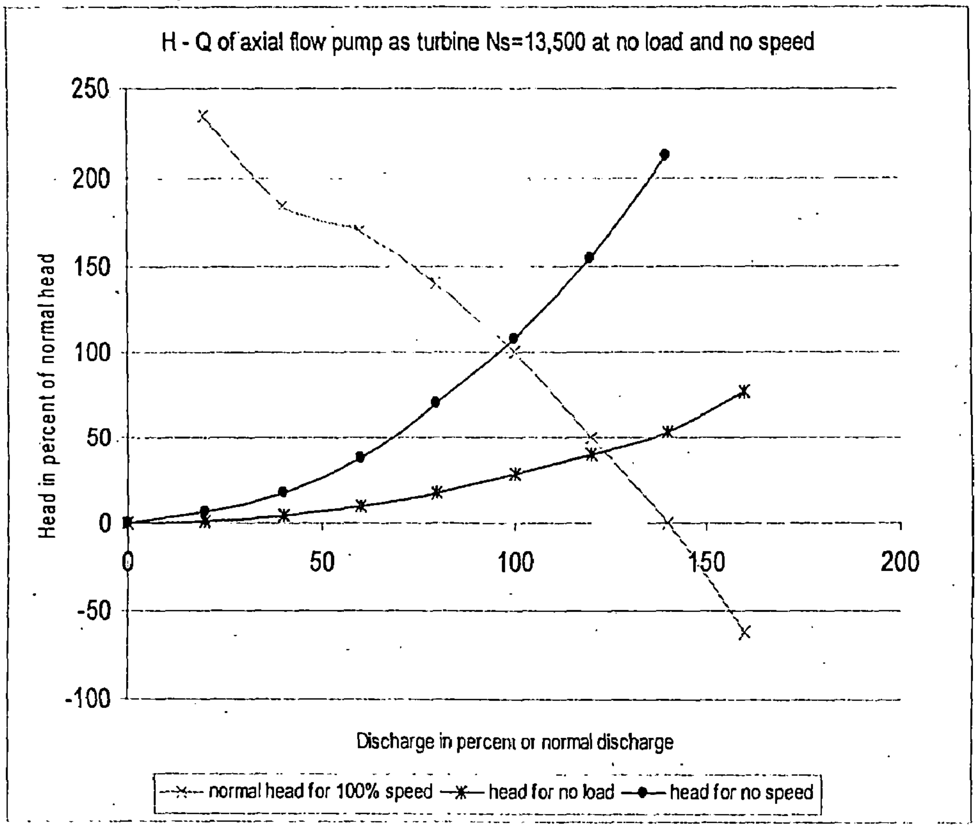


Fig. 3.28 Working range of pump as turbine (Stepanoff, A. J.) [17]

4. EXPERIMENTAL SET-UP AND TEST RESULTS

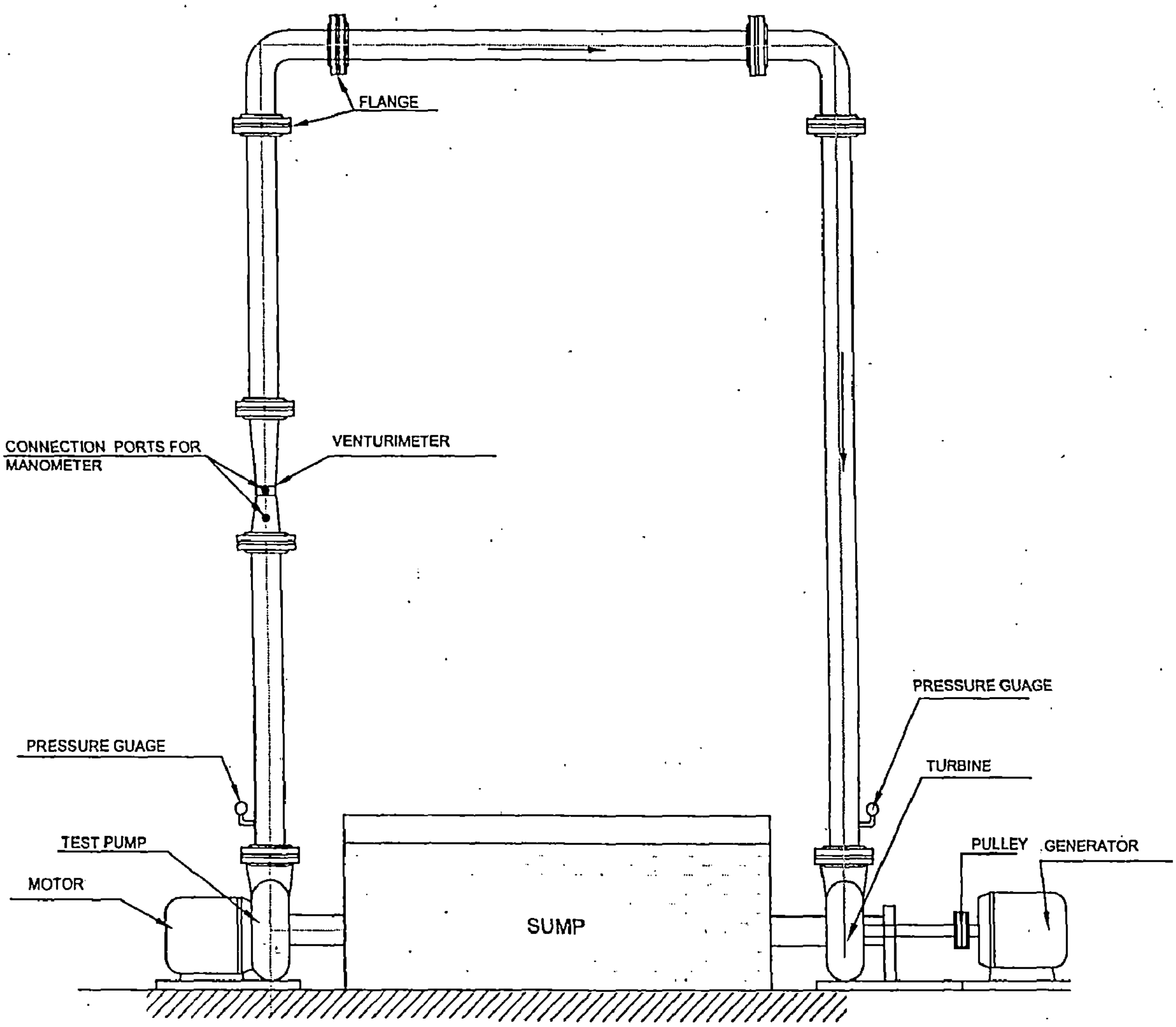
4.1 DESCRIPTION OF EXPERIMENTAL SET-UP

Equipments were set-up with the prime objective of developing h-q curve of a test pump using pump as turbine (PAT) instead of valve and recover possible energy. For that purpose we need to measure head and discharge developed by the test pump, head acting on turbine, power developed by generator at various loads and speeds. Keeping these things in view, the experimental equipments were selected and installed in River Engineering laboratory of Department of Water Resources Development and Management (formerly WRDTC). Fig. 4.1 and 4.2 show the experimental set up. One of the test loops used for testing pump at Kirloskar Brothers Limited, Pune [7] is shown in Appendix-B1. It has flow measuring capacity as high as 50000m^3/hr, head 120 m and have power up supply to 4.5 MW.

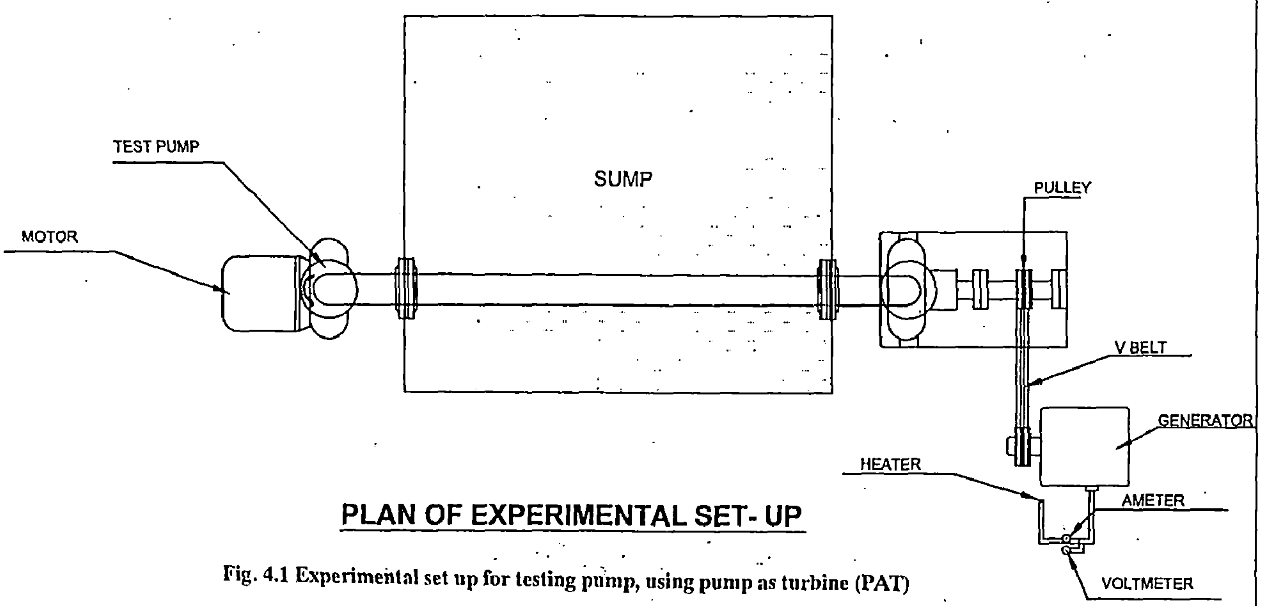
A single stage, single suction centrifugal pump of type NS-8A, 100 mm X 100 mm size, having impeller diameter of 241 mm manufactured by Kirloskar Brothers, Pune was used as test pump. For running the pump, a 5.5 KW, 3 phase, 50 cycles, 400/400 volts, and 1420 rpm motor was coupled to it. The induction motor and pump were mounted on the same base plate. Fig. 4.3 shows two views of KS4 Kirloskar pump used as test pump and Fr. MKH:132, as a prime mover.

Two different capacity single stage, single suction centrifugal pumps, one of type NW4+, 100 mm X 100 mm size, having impeller diameter of 212 mm manufactured by Kirloskar Brothers, Pune and the other of model SN 6996, 80mm X 100mm size, 7.5 hp manufactured by Voltas, India were used as turbines one at a time for testing the test pump. Fig. 4.5 and 4.6 are two views of Kirloskar NW4+ PUMP and Voltas pump respectively used as turbine (PAT) in testing the test pump.

A gate valve of 100 mm diameter was used to for the generation h-q curve of the test

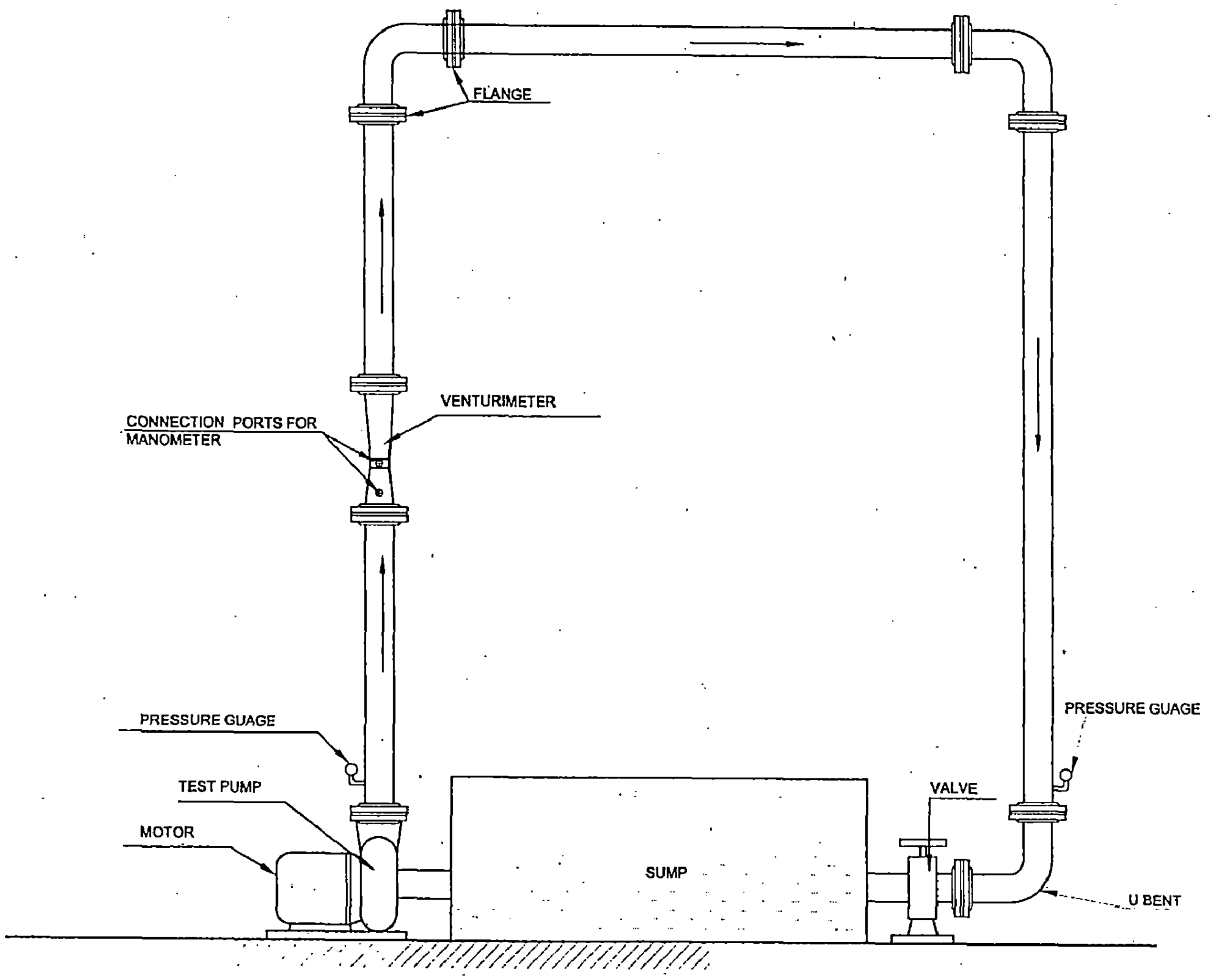


ELEVATION OF EXPERIMENTAL SET- UP FOR FLOW REGULATION USING PUMP AS TURBINE
ALTERNATE :01



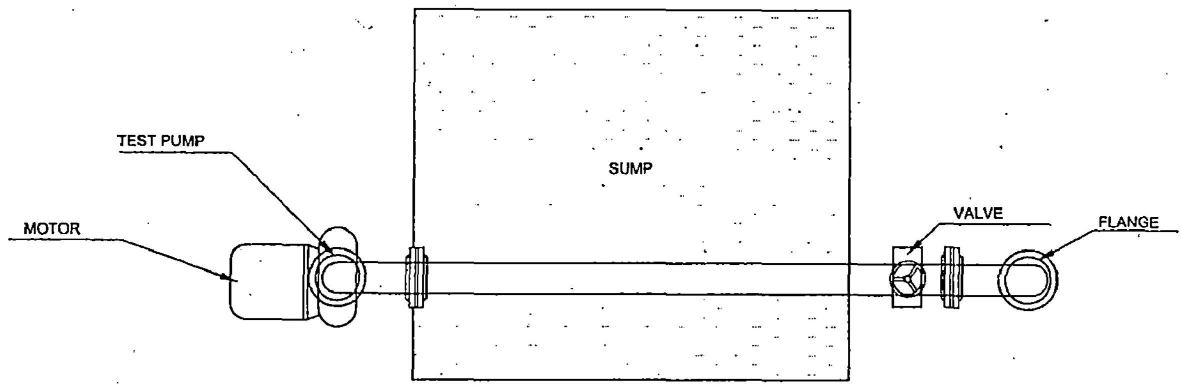
PLAN OF EXPERIMENTAL SET- UP

Fig. 4.1 Experimental set up for testing pump, using pump as turbine (PAT)



ELEVATION OF EXPERIMENTAL SET- UP FOR FLOW REGULATION USING GATE VALVE

ALTERNATE :02



PLAN OF EXPERIMENTAL SET- UP

Fig. 4.2 Experimental set up for testinpump using valve

pump so that it could be compared with the curve that was generated by those turbines (PATs).

Discharge nozzle of the test pump was connected to a turbine (PAT) with pipes, bends and fittings and the turbine was coupled with an alternating current single-phase generator by suitable V-Belt. Pressure gauges of required ranges were fitted for measurement of pressure head developed by test pump and pressure head acting on turbine (PAT). Venturimeter connected to mercury manometer was used for the flow measurement caused by the test pump Fig. 4.4 shows the venturimeter and manometer used in test. Delivery side of turbine (PAT) was connected to sump tank through pipes. Other equipments include are, a voltmeter for measurement of voltage generated by generator, an ammeter for measurement of current flowing through electric loads, a tachometer for measurement of rotational speed of generator, and room heaters for utilizing the electric power generated during tests. Specifications of test set are as follows

4.1.1 Specification of Pump and Prime Mover (Motor)

a. Test Pump

Type	Centrifugal
Make	Kirloskar, India (Kirloskar Brothers, Pune)
Model	KS4
Size	100 mm X 100mm
Impeller diameter	241mm
Horse power	6.3
Design head	15.9 m
Capacity	26 lps
Efficiency	77%
Speed	1420

Ranges:

Capacity	31.2 to 17.5 lps
Head	13 to 17 m

b. Prime Mover (Motor)

Type	Induction
Make	Kirloskar, India (Kirloskar brothers, Pune)
Model	Fr.MKH:132
Power rating	5.5 KW
Voltage	415 Volt
Current	11 Amper
Cycles	50 cycle per second
Speed	1420 rpm
Connection	Delta
Insulation type	B

4.1.2 Specification of Pump Used as Turbine

a. Pump one (PAT 1)

Type	Centrifugal
Make	Kirloskar, India (Kirloskar Brothers, Pune)
Model	NW4+
Size	100mm X 100mm
Impeller diameter	212mm
Horse power	5
Design head	13 m
Capacity	22 lps
Efficiency	74 %
Speed	1450 rpm
Ranges:	
Capacity	26 to 19 lps
Head	11.5 to 13.5 m

b. Pump two (PAT 2)

Type	Centrifugal
Make	Voltas, India
Model	S N 6996
Size	80 mm X 100 mm

Horse power	7.5
Speed	1440 rpm

4.1.3 Specification of Generator

a. Generator

Type	Alternating current, single phase
Make	Kirloskar, India
M. C. No.	80/ H 28 A - 14
Power rating	3.5 KW
Volt amperes	3.5 KVA
Power Factor	1
Voltage	220 Volt
Current	15.9 Ampere
Cycles	50 cycle per second
Speed	1500 rpm
Excitation:	200 volt, 0.6 ampere

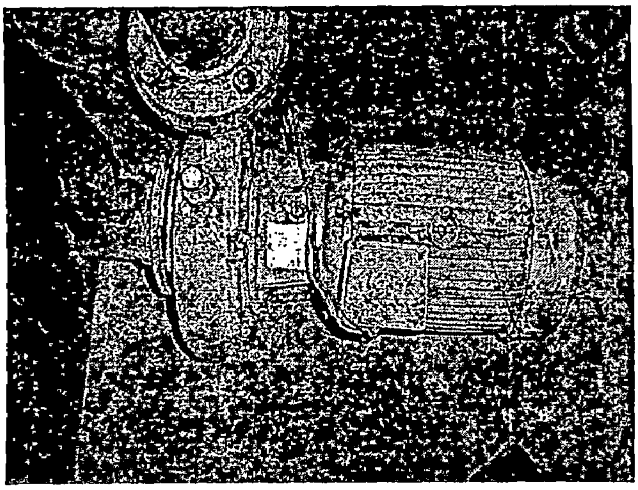


Fig. no. 4.3 KS4 Kirloskar centrifugal pump with Fr.MKH:132 induction motor

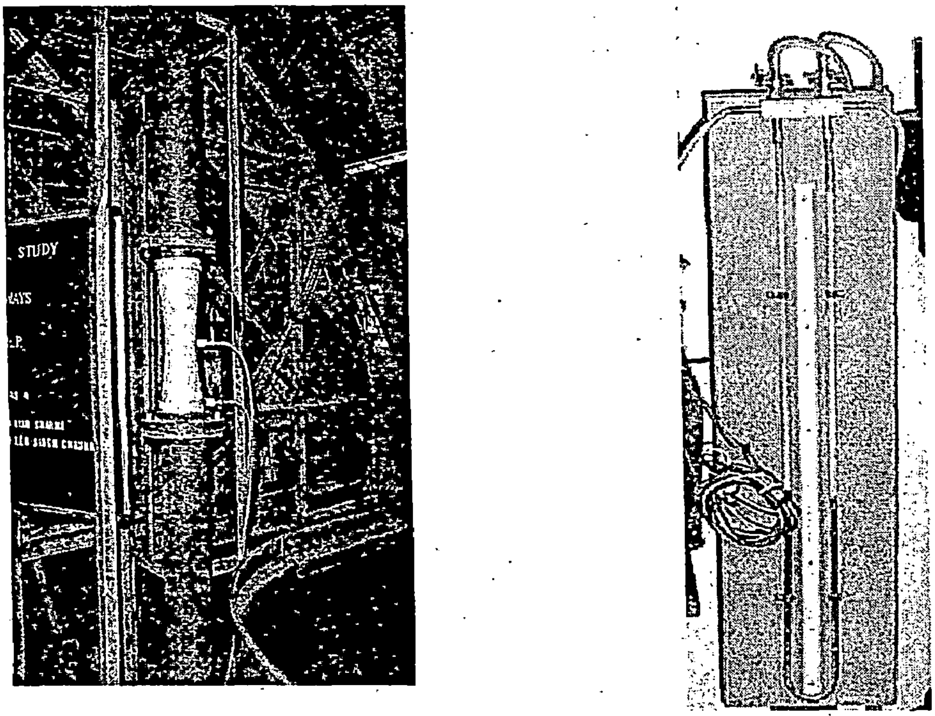


Fig. no. 4.4 Venturimeter and mercury manometer

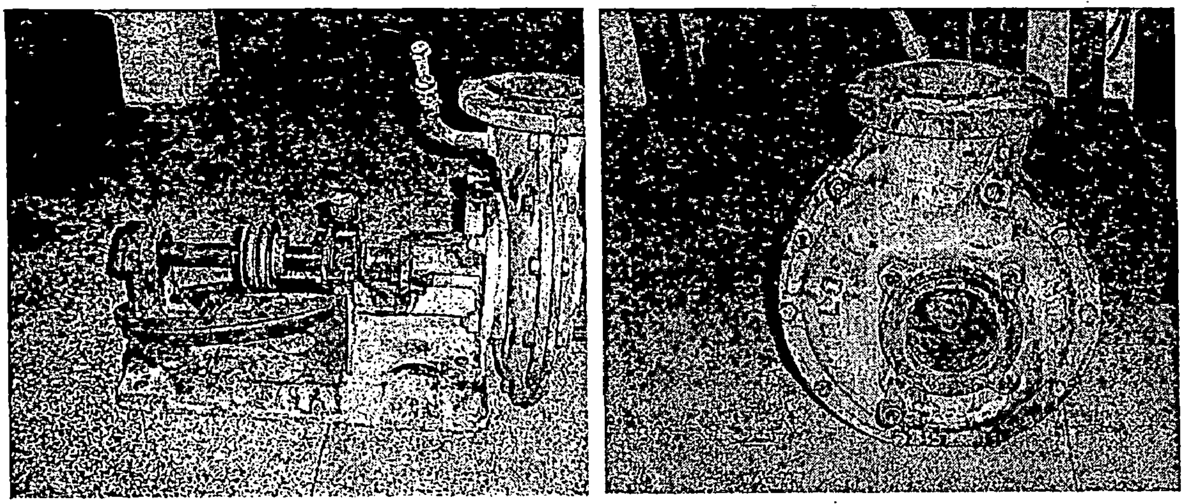


Fig. 4.5 NW4+ Kirloskar centrifugal pump (used as turbine)

4.2 EXPERIMENTAL PROCEDURE

For generating h-q curve of test pump using turbines (PATs), the experimental set up was completed as shown in Fig. 4.1 (first with turbine (PAT 1)). With the experimental set-up complete, the sump was filled with water up to the level so that during operation, the discharge pipe was immersed in it. The test pump was on and

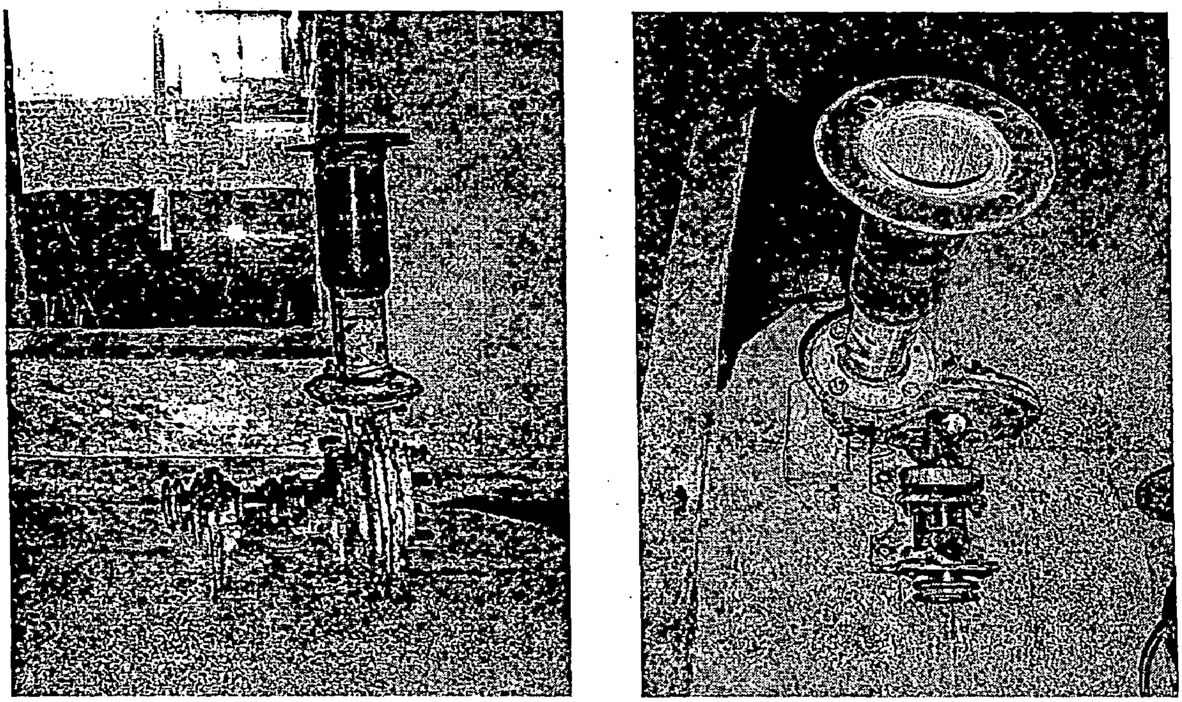


Fig. no. 4.6 Voltas centrifugal pump (used as turbine)

was allowed to run for sometime (without taking any readings) it to stabilize. Air bubbles inside the manometer tube were removed by opening knobs, on the top of manometer tube, that allowed water to pass through and air to escape. After the test pump was stabilized, the rpm of the generator was measures, followed by measurement of voltage generated by generator and current flowing to load. The pressure head difference across the throat and the inlet of venturimeter was noted, as difference in level of mercury in limbs of manometer tube, for flow measurement. Four different reading as said above were taken when the generator was loaded with 0 KW, 1 KW, 2 KW and 3 KW heaters respectively. Fig. 4.7 shows the experiment performed on the test pump (KS4) with NW4+ pump as turbine.

After the readings were note for finding out h-q curve of test pump by the first turbine (PAT 1), the turbine was removed and replaced by second pump to operate as turbine (PAT 2). The same procedures of the experiment were followed and readings were noted for study and analysis. Fig. 4.8 shows the experiment performed on test pump (KS4) with SN6996 pump as turbine.

After that, completion of experiment with the second turbine (PAT 2), it was removed and replaced by valve for getting h-q of test pump. The procedures of experiment were similar to that with turbine except for no reading for voltage and current. The experiment started with valve fully closed followed by gradual opening of valve to full opening position. For each valve stem position, pressure gauge reading on test pump and head difference of mercury level in two limbs of manometer was measured and noted. Fig. 4.9 shows the test pump being tested with gate valve.

4.3 EXPERIMENTAL DATA AND BASIS FOR CALCULATION

4.3.1 Venturimeter Flow Calculation:

Difference of level of fluid flowing through venturimeter throat and inlet h (m)

$$h = y * \left(\frac{\gamma}{\gamma_w} - 1 \right) \quad \text{.................................4.1}$$

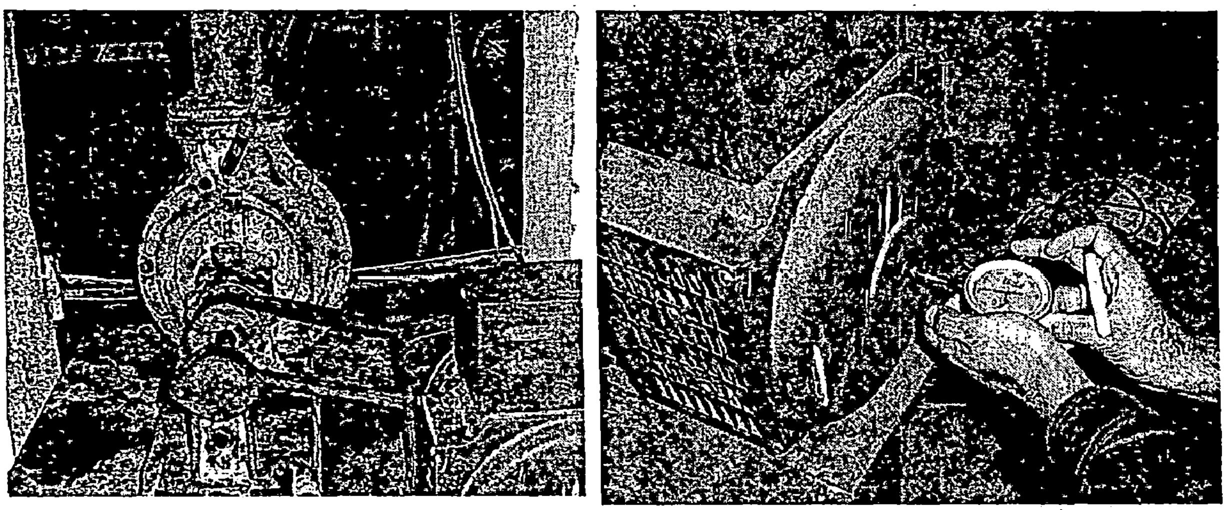


Fig. 4.7 Kirloskar pump used as turbine for flow regulation and generator driving

where,

* is symbol indicating multiplication

y is difference of level of mercury in manometer limbs (m).

γ is specific weight of mercury (N/m^3)

γ_w is specific weight of water (N/m^3)

$$h = y * \left(\frac{13.6}{1} - 1 \right)$$

$$h = y * (13.6 - 1)$$

$$h = y * 12.6$$

Fig. 4.8 Voltas pump used as turbine for flow regulation and generator driving

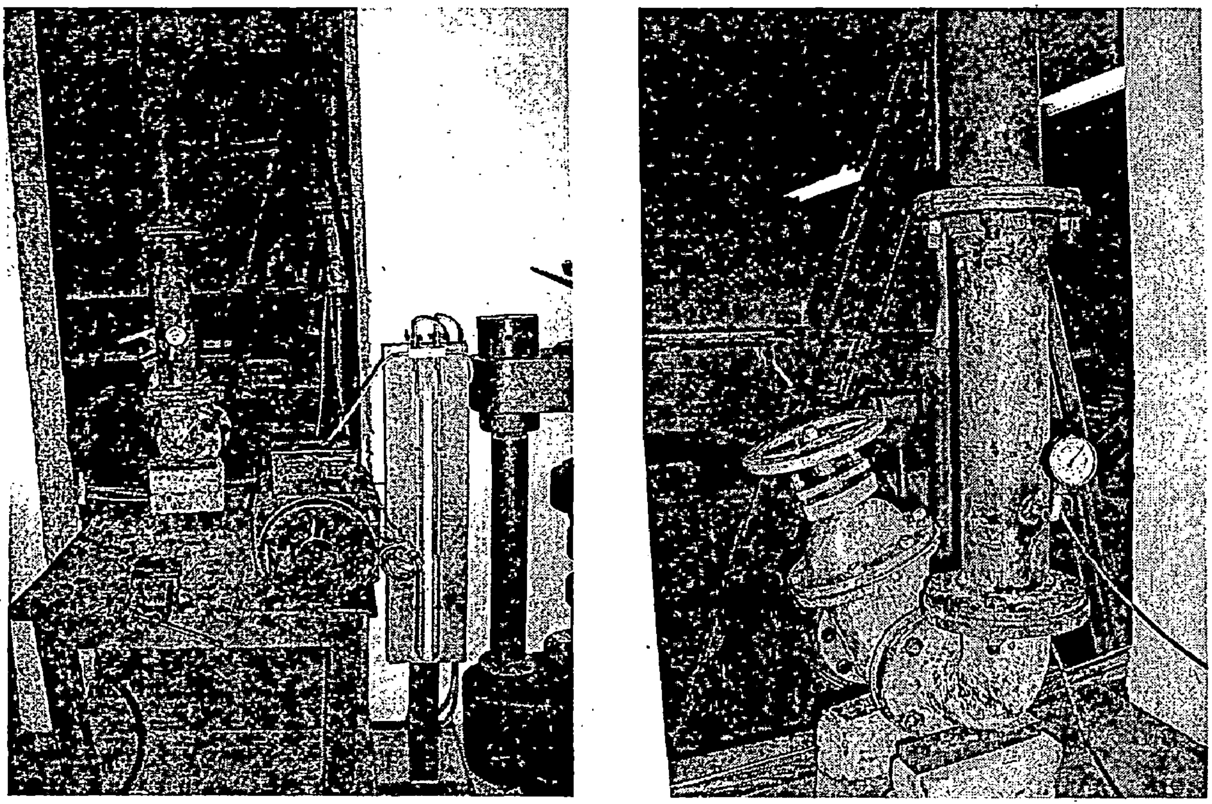


Fig. 4.9Gate valve used for pump testing

We have,

Discharge $Q = \left\{C_d * a_1 * a_2 * (2 * g * h)^{1/2}\right\} / \left(a_1^2 - a_2^2\right)^{1/2}$ 4.2

Q is discharge (m^3/s)

Where,

$a_1 = \pi/4*(0.1)^2 = 7.854*10^{-3}$ (area of inlet of venturimeter m^2)

$= 0.007854\ m^2$

$a_2 = \pi/4*(0.06)^2 = 2.827*10^{-3}$ (area of throat of venturimeter m^2)

$= 0.002827\ m^2$

$C_d = 0.98$ (coefficient of discharge)

$$Q = \left\{0.98*7.827*2.827*10^{-6}*(2*9.81*12.6*y)^{1/2}\right\} / \left\{(7.854*10^{-3})^2 - (2.827*10^{-3})^2\right\}^{1/2}$$

$$= 0.04466*(y)^{1/2}\ m^3/s.$$

4.3.2 Calculation of Power Input to Turbine (P_h)

$P_h = \rho g H Q$ 4.3

ρ is density of fluid used (1000kg/m^3)

g is acceleration due to gravity (9.81 m/ sec^2)

H is head acting on turbine (m)

Q is discharge through turbine (m^3/sec.)

4.3.3 Calculation of Power Generated by Generator (P_g)

$Power(P_g) = Voltage(V) * current(I) * powerfactor(\cos\phi)$

$P_g = V * I * \cos\phi$ 4.4

4.3.4 Calculation of Power Out put of Turbine (P_t)

Power produced by Turbine= Power out put of generator + Generator losses (10% of generator out put) + losses in belt drive (3% of generator out put)

$P_t = P_g + \frac{P_g}{0.9} + 0.03 * P_g$ 4.5

4.3.5 Calculation of Efficiency (η)

$$Efficiency(\eta) = \frac{outputofturbine}{inputtoturbine} \qquad \text{.......4.6}$$

$$\eta = \frac{P_t}{P_h}$$

4.4 TEST RESULTS

Measurements of the tests with PAT 1, PAT 2 and valve are presented in Table 4.1, 4.2 and 4.3 respectively.

Table 4.1 Measurements of Experiment done with PAT 1

S. No.	rpm generator	Load (KW)	Pressure head Developed by Test Pump	Pressure head on Turbine	Venturimetere reading (cm) left arm	Venturimetere reading (cm) right arm	Differential mercury manometer head (cm)	Voltage (V)	Current (I)
1	1320	0	17.6	15.8	12.8	26	13.2	170	0
2	1200	1	16.5	14.6	9.6	28.9	19.3	155	2.4
3	1100	2	15.8	13.8	8.7	29.9	21.2	140	4.2
4	1000	3	15.4	13.3	7.8	30.6	22.8	125	5.8

Table 4.2 Measurements of Experiment done with PAT 2

S. No.	rpm generator	Load (KW)	Pressure head Developed by Test Pump	Pressure head on Turbine	Venturimetere reading (cm) left arm	Venturimetere reading (cm) right arm	Differential mercury manometer head (cm)	Voltage (V)	Current (I)
1	1320	0	19	17.4	35	40.5	5.5	170	0
2	1210	1	18.8	17.2	34.1	41.6	7.5	155	2.4
3	1105	2	18.5	16.7	32.9	42.7	9.8	140	4.2
4	1010	3	18.2	16	32.3	43.5	11.2	125	5.8

Table 4.3 Measurements of Experiment done with valve

S. No.	Venturimeter reading (cm) left arm	Venturimeter reading (cm) right arm	Differential mercury manometer head (cm)	Head developed in meter (m)
1	30.4	30.4	0	19
2	30.5	30.3	0.2	19.2
3	30.6	30.3	0.3	19.2
4	30.6	30.2	0.4	19.3
5	31.6	29.3	2.3	19.3
6	36.4	24.4	12	17.9
7	43	17.8	25.2	14.6
8	48.4	12.4	36	11.8
9	51.6	9	42.6	9.8
10	53.7	6.8	46.9	8.6
11	55	5.5	49.5	7.7
12	56	4.5	51.5	6.5
13	56.9	3.4	53.5	6
14	57.3	3.4	53.9	5.7
15	57.5	3.2	54.3	5.6

Using values form Tables (4.6 & 4.3), Fig. 4.10 is drawn, using Table (4.1, 4.2 ,4.4 and 4.5) Fig. 4.11 is drawn and Fig. 4.10, and 4.11 is superimposed to get Fig. 4.12.

H-Q curve of NS-8A by valve

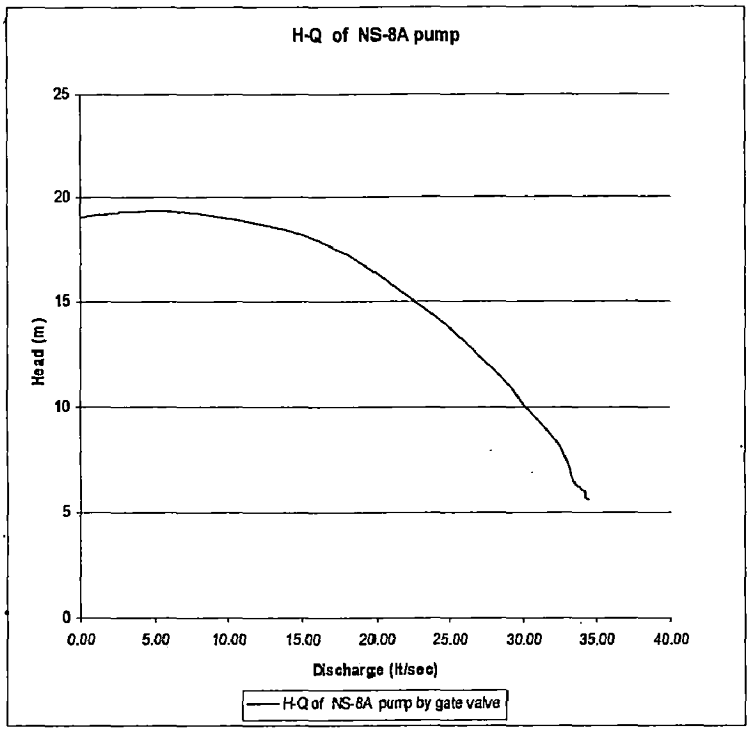


Fig. 4.10 H-Q curve of test pump by valve

H-Q curve of test pump produced by PATs

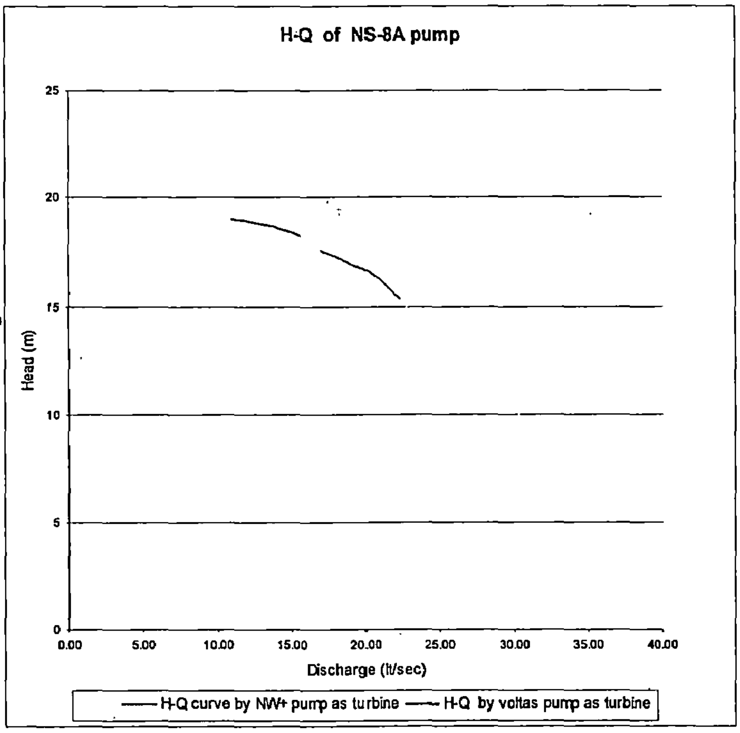


Fig. 4.11 H-Q curve of test pump by 2 turbines

Using equations 4.1, 4.2, 4.3, 4.5 and 4.6 discharge, power acting on turbine, power produced by generator, power produced by turbine and efficiency were respectively obtained and are tabulated in their respective tables of PATs and valve.

Table 4.4 Calculated values of discharge, power and efficiency (PAT 1)

S. No	Differential water head (m)	Discharge (lt/s)	Vel. head (m)	Power by generator (W)	Generator losses (10% generator o/p) (W)	Belt loss @3% of generator o/p (W)	o/p turbine (W)	i/p turbine (W)	efficiency
1	1.6632	16.96	0.2	0	-	-	0	0	0
2	2.4318	20.5	0.4	372	413.3	11.16	424	3007	0.14
3	2.6712	21.5	0.4	588	653.3	17.64	671	2991	0.22
4	2.8728	22.3	0.4	725	805.6	21.75	827	2999	0.28

Table 4.5 Calculated values of discharge, power and efficiency (PAT 2)

S. No.	Differential water head (m)	Discharge (lt/s)	Vel. head (m)	Power by generator (W)	Generator losses (10% generator o/p) (W)	Belt loss @3% of generator o/p (W)	o/p turbine (W)	i/p turbine (W)	efficiency
1	0.693	10.95	0.2	0	0	0	0	1895	0
2	0.945	12.8	0.4	372	413.3	11.16	424	2201	0.19
3	1.2348	14.61	0.3	588	653.3	17.64	671	2455	0.27
4	1.4112	15.62	0.5	725	805.6	21.75	827	2527	0.33

Table 4.6 Calculated values of head and discharge used by valve

S. No.	Differential water head (m)	Discharge (m^3/sec)	Discharge (lt/sec)
1	0	0.000	0.00
2	0.0252	0.002	2.09
3	0.0378	0.003	2.56
4	0.0504	0.003	2.95
5	0.2898	0.007	7.08
6	1.512	0.016	16.17
7	3.1752	0.023	23.44
8	4.536	0.028	28.01
9	5.3676	0.030	30.47
10	5.9094	0.032	31.97
11	6.237	0.033	32.85
12	6.489	0.034	33.51
13	6.741	0.034	34.15
14	6.7914	0.034	34.28
15	6.8418	0.034	34.40

H-Q curve of test pump produced by valve superimposed on H-Q produced by PATs

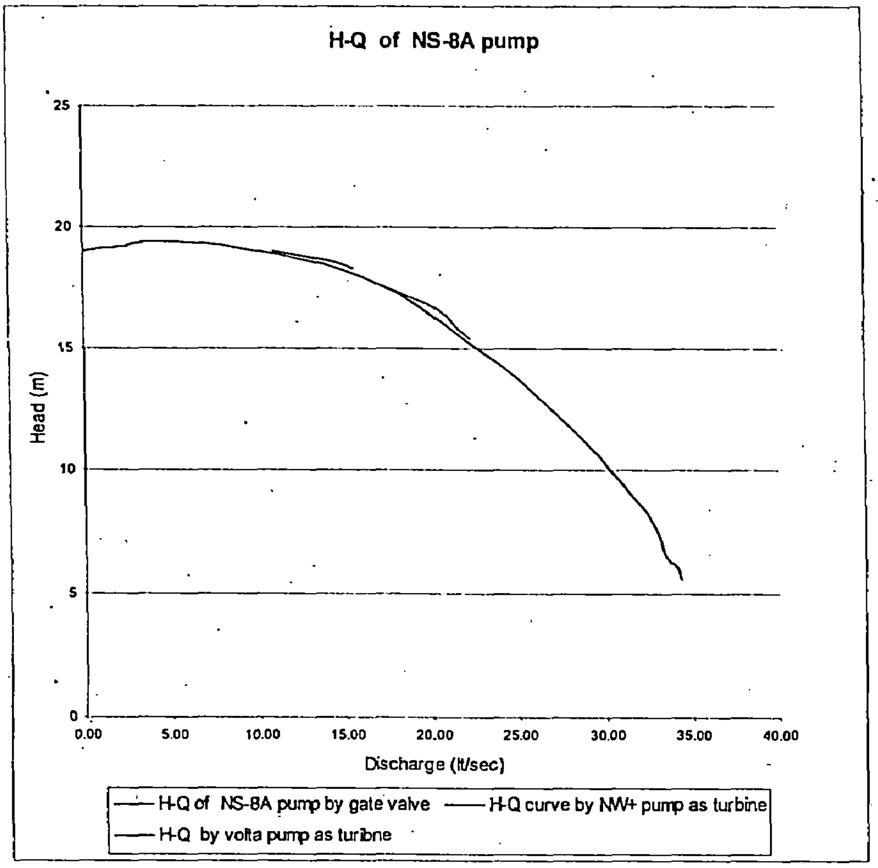


Fig. 4.12 H-Q curve of test pump by gate valve and 2 turbines superimposed

5. DISCUSSION ON TEST RESULTS & ITS IMPLICATIONS

5.1 DISCUSSION ON TEST RESULTS

It is evident form Fig. 4.10, 4.11 and 4.12 in the chapter- 4 that varying the speed and capacity of turbines (PATs) can regulate flow of test pump and a portion of h-q curve can be generated. The variation of head with flow follows the head-discharge characteristics as developed by valve during the pump test; however the h-q generated by turbines (PATs) in combination is not sufficient to represent the entire h-q curve as produced by valve.

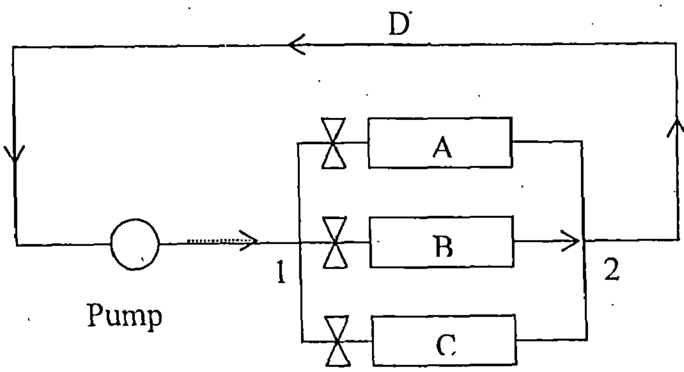
When valve is fully open, the resistance to the flow is the least and hence the flow is the maximum. When valve is closed slightly, it creates some resistance and this develops pressure upstream and hence adds resistance on the test pump. Thus the task of the valve is to add resistance to flow by increasing upstream pressure that acts on pump and maintain pressure difference so that sufficient pressure exists at downstream of the valve to push the flow. By further closing the valve, upstream pressure on the test pump is increased and pressure loss in the valve is also increases to maintain nearly original total pressure downstream of the valve.

With a pump used as turbine instead of valve, the h-q curve of turbine (PAT) at certain speed intersects the h-q curve of the test pump. The point of intersection is known as operating point. When the turbine (PAT) is further loaded, the speed get reduced and the process is like opening valve (for radial flow) and other operating point is obtained for that speed. Similarly when load is removed from turbine (PAT), its speed increases and the process is like closing valve further and other point of operation is obtained. With further loading and unloading more numbers of operating points can be obtained. Joining these points a portion of h-q curve of test pump can be generated. With two pumps used as turbines and loading those to run at various speeds, during experiment, a portion of h-q curve of test pump is developed thus the turbines (PATs) doing valve duty for certain flow range.

For generation of continuous h-q curve of test pump, more pumps of the same or different capacity can be used as turbines in parallel. An analogy can be drawn from a network of pipes use for water distribution and the same principle could be used for our works.

5.2 BRANCH-LINE PUMPING SYSTEM

When liquid leave from centrifugal pump/s through a pipe and divide into a network of pipes it flows dividing into network of pipes. The total flow depends on the combined system resistance. Fig. 5.1 shows the closed loop pump system with three branch lines and no elevation difference.



A, B and C are system total head curve for three branch lines

Fig. 5.1 Closed-loop pump system with branch lines (Karassik) [6]

The system total head curves for each branch lines and header are independent of each other and leveled A, B, C, and D. Fig. 5.2 shows system-head curves for pump and branch lines with all valve open. These curves are constructed for several flow rates by adding the frictional resistances of the pipes, fittings, and head losses through the equipment serviced from junction point 1 to 2. Curves A, B, C, and D therefore represent the variation in system resistance versus flow through each branch and header. If the valves are fully open in all branches the total system resistance, total pump flow, and individual branch flows are found by the following method. It is noted that

- The total flow must be equal to the sum of the branch flows,

- The head loss or pressure drop across each branch from junction 1 to junction 2 is identical and
- The flow divides to produce these identical head losses.

Therefore, at several head points, flow though each branch is added together and a curve A+B+C is obtained. Header D is in series with A + B + C, and their system heads are added together for several flow conditions to obtain curve (A + B + C) + D. Let curve E be the head-capacity characteristic of a centrifugal pump, point X represents the pump flow since at this point the system total head and pump total head are equal. Point Y_{1-2} represents the total head across junction points 1 and 2 and this head determines the flow through each branch; consequently points *a*, *b*, and *c* gives individual branch flows as shown in Fig. 5.2.

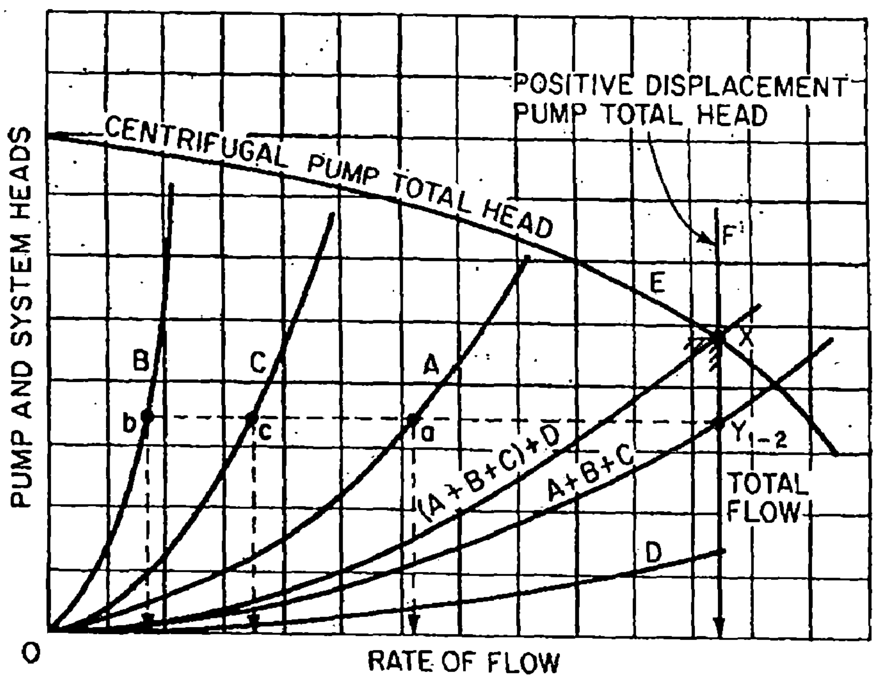


Fig. 5.2 System-head curves for pump and branch lines with all valves open (Karassik) [6]

If valve A is opened and valve B and C are closed, the pump flow and branch ‘A’ flow are the same and the total flow is less than the condition for all valves open at point X as a result of an increase in system head. Fig. 5.3 shows the construction of the curves required to determine pump flow point X’. It is also shown in the figure that the system head curve for different combinations of open valves A, B, and C and the resulting flow caused by a pump having characteristic curve E. For these various valve combinations the head differential across the junction point is found by subtracting the head of curve D from the system total head for the condition investigated, to obtain point Y'_{1-2}. The intersection of a horizontal line through point Y'_{1-2} and the individual branch curves gives the branch flow.

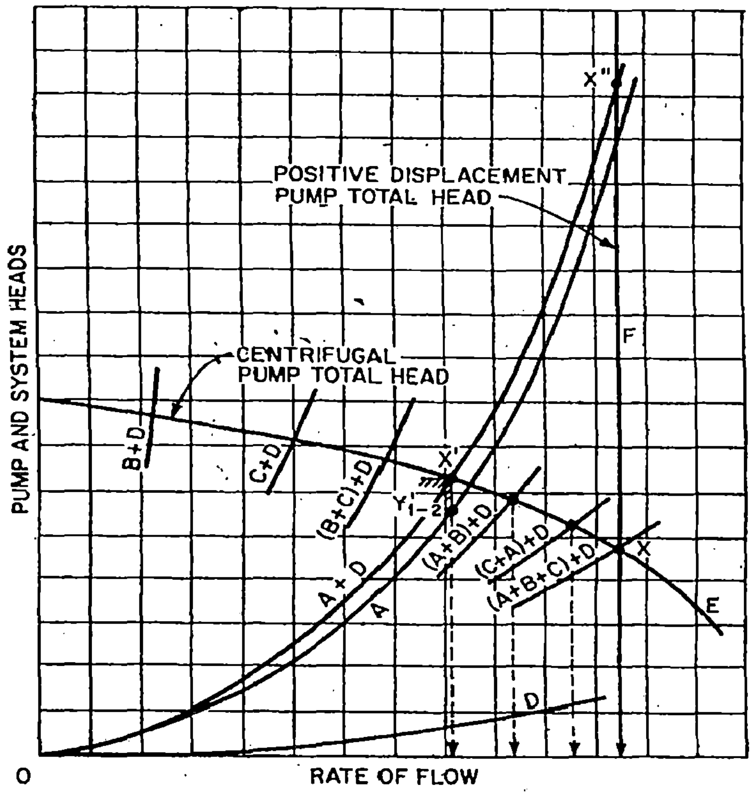


Fig. 5.3 System-head curves for pump and branch line with different combination of open valves (Karassik) [6]

5.3 CONCLUSION FROM TEST RESULTS AND FLOW THROUGH BRANCH PIPE

The turbine (PAT) head-discharge characteristic is similar to the system head curve of the piping system; the same principle would be used for determination of operating point of turbine (PAT) thus head and discharge point of test pump at that operating point. Turbines (PAT) whose head-capacity curve is known are connected in parallel and finally to the test pump. As in piping system, as per the number of turbines (PATs) that would be operated; at several head points, flow through each turbine (PAT) is added together to get the system total h-q curve for that numbers of turbines (PATs) at that speed. The point of intersection of head-discharge of the test pump and the system gives the point of operation. Discharge at this operating point is measured by flow meter and dynamic head is measured by algebraically adding pressure head, velocity head, and datum head. Loading and unloading system results in speed change of turbines (PATs) and more points of operation can be obtained but, as shown by experiment, these points are close to each other and only small portion of h-q curve could be produced. The h-q curve of the turbines (PATs) for various speeds can be determined by using the eq.2.15. Closing or opening of one or more turbines (PATs) in recovery system can shift the operating point farther. Opening of turbines (PATs) cause increase capacity of the system and is similar to opening a branch pipe in pipeline network. As total discharges get increased, this results in decrease in dynamic head acting on system by test pump. Though total discharge is increased but discharge for individual turbines (PATs) get reduced as the increased discharge is to be shared by more number of turbines (PATs). Shutting down of turbines (PATs) in system results in lowering the total capacity of the system. This is like closing branches in pipe network and this result in increased total dynamic head acting on system by test pump. Though total discharge get decreased, but it is shared by lesser number of turbines (PATs) which cause increase in discharge for individual turbines (PATs). Thus by opening and closing of turbines (PATs), in system, which is running at certain speed, the h-q turbine (PAT) at that speed is followed and various point of operation for system and thus the h-q curve for the test pump is determined while power being recovered. Fig. 5.4 is a typical arrangement for energy recovery system for recovering energy during pump-testing.

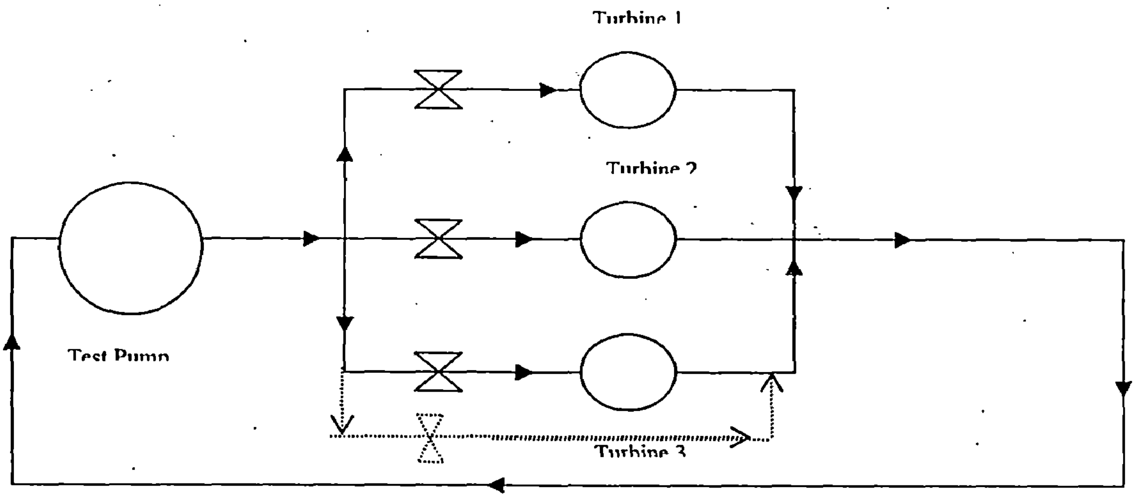


Fig. 5.4 A typical arrangement for testing pump using turbines in parallel

5.4 DESIGN OF ENERGY RECOVERY SYSTEM FOR 24UPH3 PUMP

Fig. 5.5 shows head-discharge and power–discharge characteristics curve of Kirloskar make 24UPH3 pump. The curve is produced by valve in different closure positions; the pump has 750 mm impeller diameter and it runs at 725 rpm in normal operating condition.

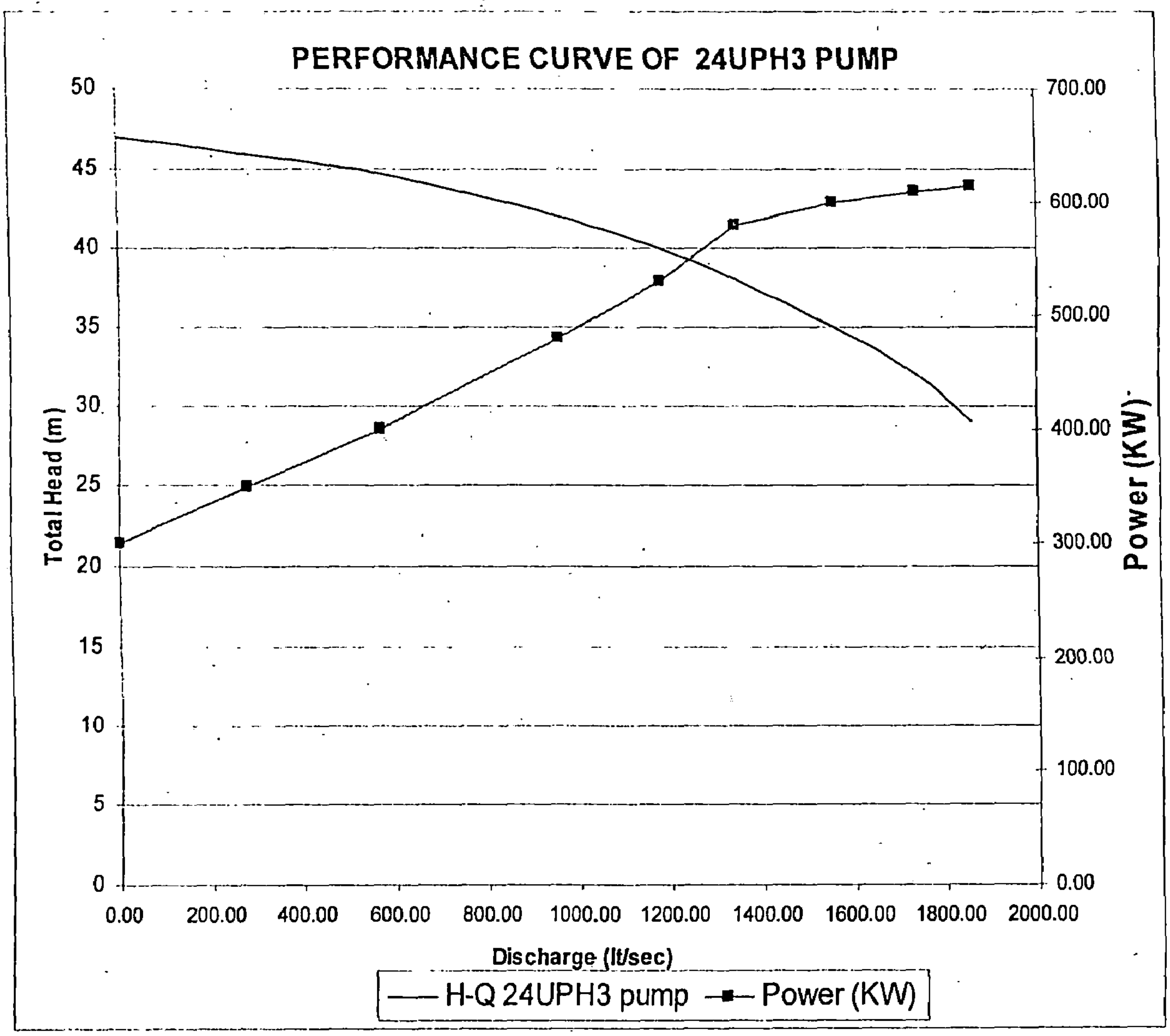


Fig. 5.5 Performance curve of 24UPH3 pump at 725 rpm

Let the curve be generated by joining minimum of 7 points, these points are generated by using a valve and are points of operation at different valve closure positions. The h-q curves of a valve for those points are obtained by using relation 5.1. Table 5.1 shows the different values of resistance coefficient (K) of valve at different closure positions.

$H = KQ^2$ 5.1

where, H is head loss (m) and Q is flow (m^3/s) through valve

when valve is fully open.

$Q_1 = 1.85 m^3/s$, $H_1 = 29$ m

thus, $K_1 = 8.5$

at partial valve openings, values of Ks are

Table 5.1 Values of resistance coefficient at different valve closure positions

K_1	K_2	K_3	K_4	K_5	K_6
8.5	12.45	19.5	35.5	78	322

Using values of K_s, h-q curves at various valve openings are obtained.

H-Q curves for 24UPH pump and valve at different opening positions is in Fig. 5.6

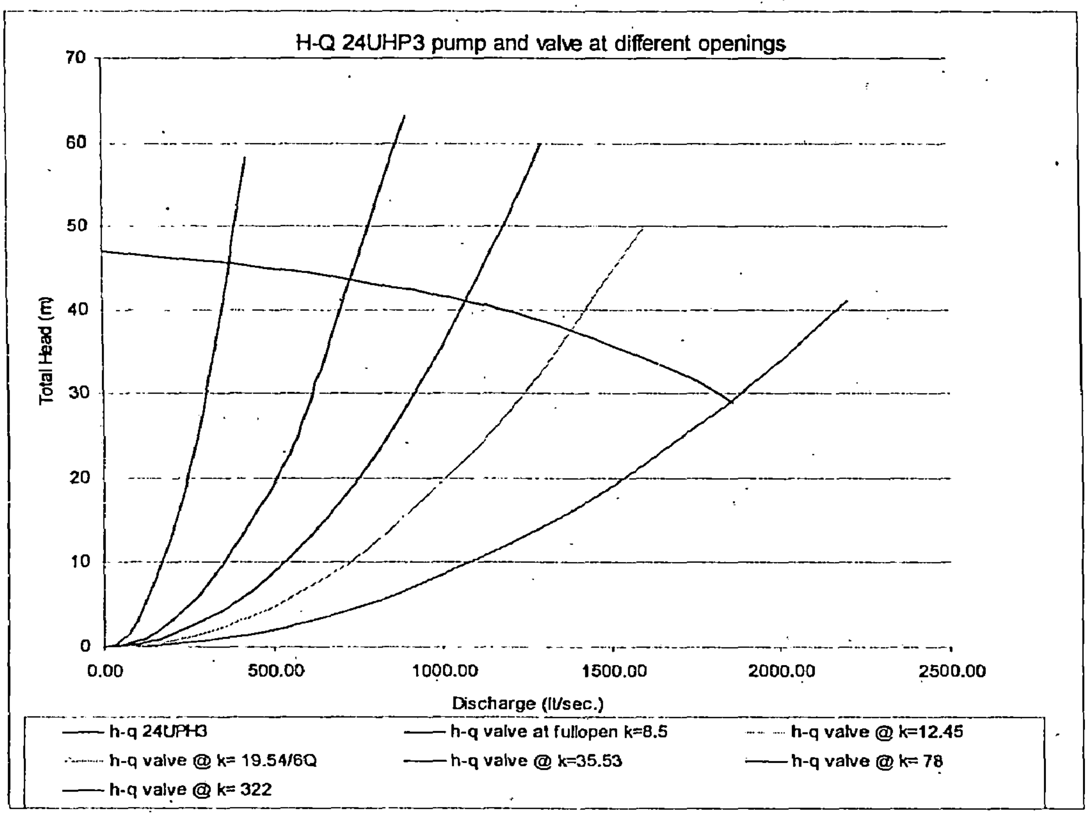


Fig. 5.6 H-Q curves of test pump and valve at different closure positions.

A turbine (PAT) whose h-q characteristic matches with that of valve when its *K* value is 332 is to be selected. The turbine (PAT) that matches the condition is of CORNELL PUMP CO. of model 10TR1-4 TURBINE. When 6 of these turbines (PATs) are run in parallel to test the 24UPH3 pump, the system will have h-q point that has discharge 1860lt/sec at 29 meter head, which is maximum discharge at that head the test pump can produce.

Fig. 5.7 is the performance curves of CORNELL PUMP CO. of model 10TR1-4 TURBINE in turbine mode.

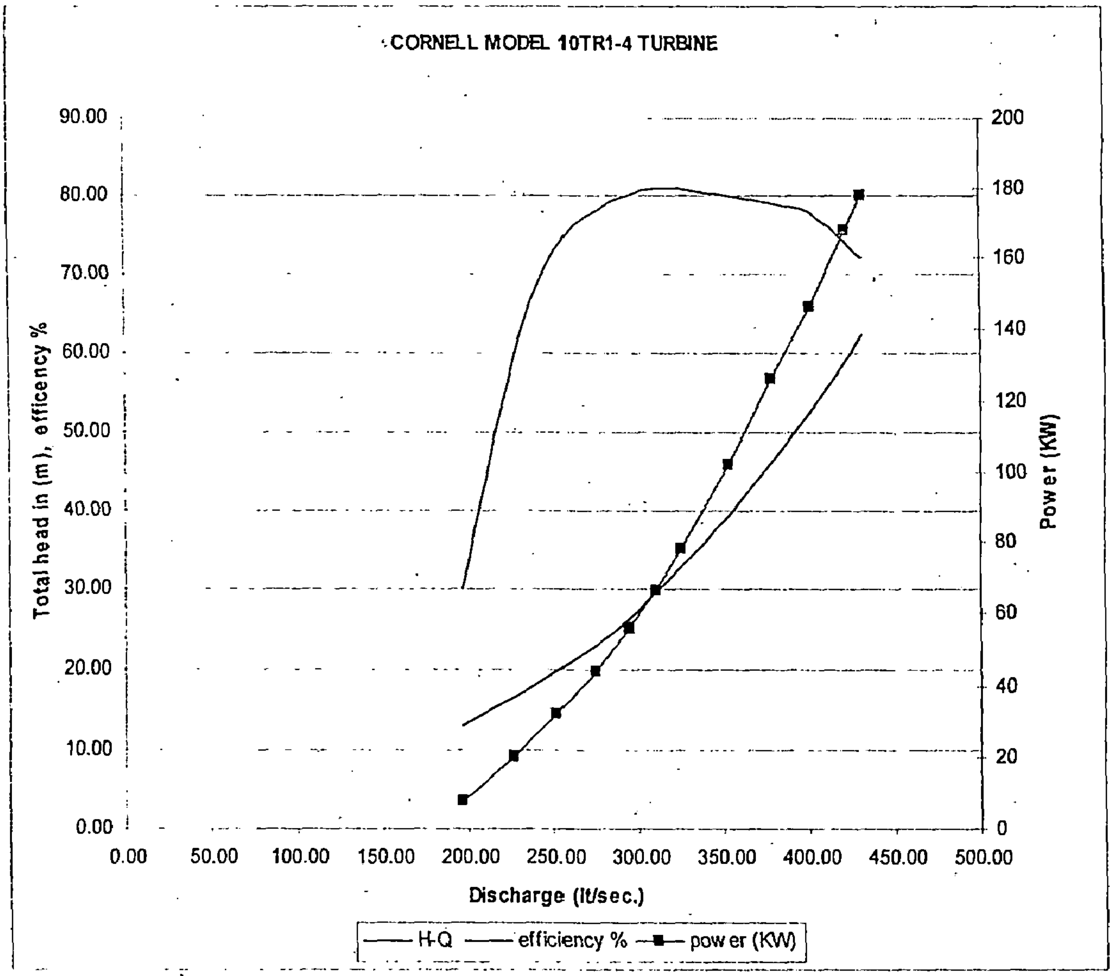


Fig. 5.7 Performance curve of CORNELL model 10TR1-4 pump as turbine (Chappel, J. R.) [3]

Fig. 5.8 is the h-q curve of recovery system (six parallel turbines (PATs) in operation) and of test pump. It is superimposed with the valve h-q curve at fully opened condition. When turbines (PATs) in system operate at normal (1800 rpm) speed, operating point is the intersection of h-q pump and system. The flow distribution for turbines (PATs) of system at normal speed is shown in Fig. 5.9. It is worth noting the remarkable collapsibility of the h-q valve with the h-q curve of recovery system. Thus the h-q turbines (PATs) in combination replicate the valve characteristics.

Superimposed h-q curve (test pump, valve at full opening and system)

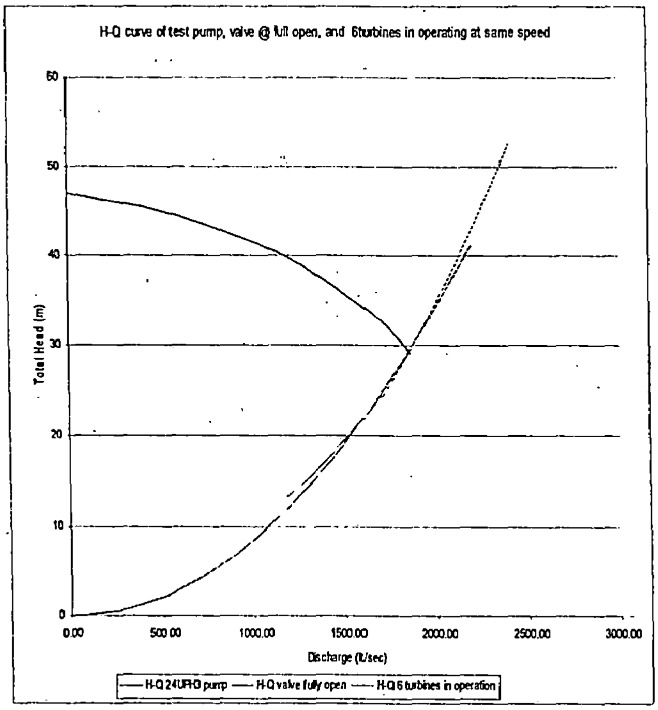


Fig. 5.8 H-Q curve of system (6turbines operating) and valve at full open position

Superimposed h-q curve (test pump and system) and flow distribution for turbines as QT_n (n=6)

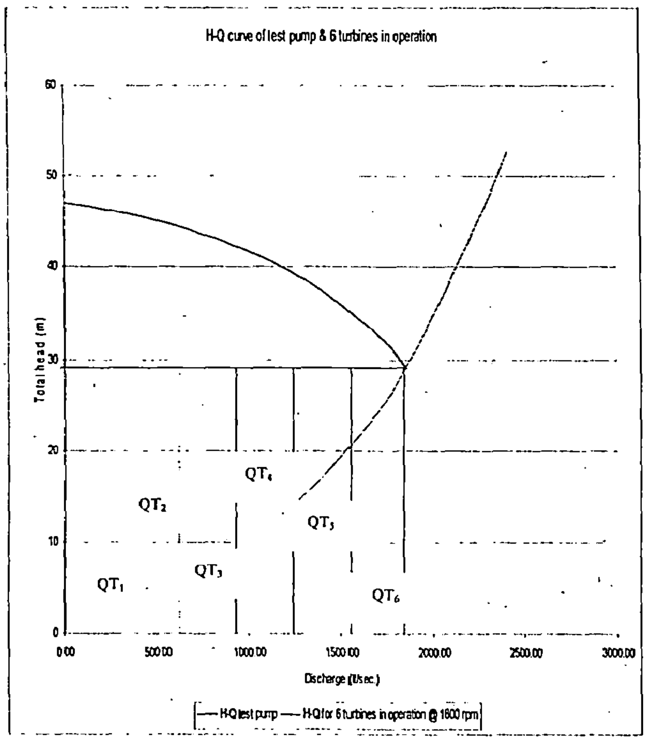


Fig. 5.9 H-Q curve of recovery system (6 turbines operating) and flow distribution in turbines

Power produced by a turbine (PAT) at that flow and head (310 lt/s. and 29 m) is obtained form the performance curve of the turbine from Fig. 5.7 and is 67 kW at 81% efficiency. Thus the total power produced by the system is 6 X 67 = 402 kW.

When one of the turbines (PATs) in system is closed and the remaining turbines (PAT) are operated, at the normal speed, the recovery system has different operating point. The h-q curve followed is shown in Fig. 5.10 and the curve when superimposed with the h-q valve at that operating point almost coincides with that of valve replicating valve characteristics. The flow distribution for five turbines (PATs) is shown in Fig. 5.11.

Power produced by a turbine (PAT) at flow rate of 326 lt/s. and 33 m head is obtained from the performance curve of the turbine in Fig. 5.7 and is 78 kW at 81% efficiency. Thus the total power produced by five turbines (PATs) or by system is 5 X 78 = 390 kW.

Superimposed h-q curve (test pump, valve at partial opening and system)

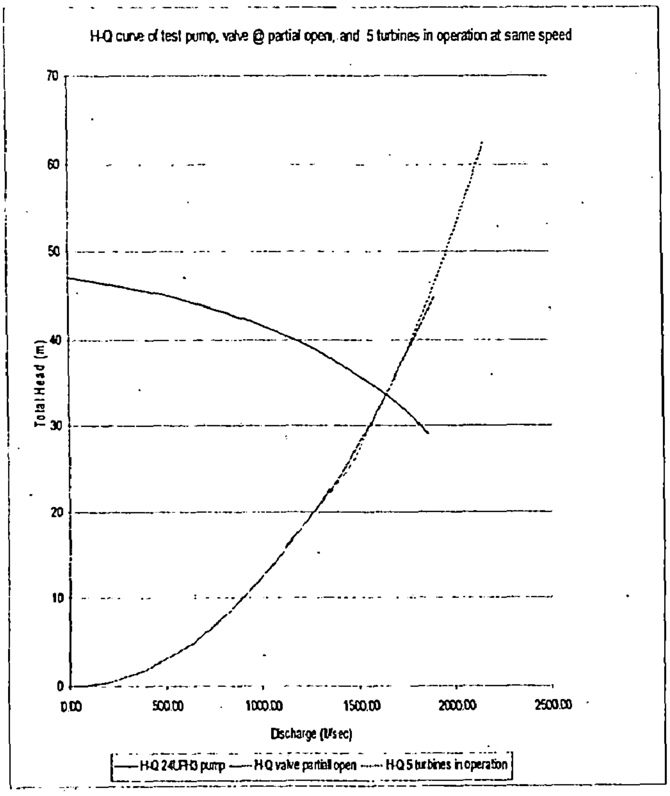


Fig. 5.10 H-Q of system (5turbines operating) and valve at partial open position

Superimposed h-q curve (test pump and system) and flow distribution for turbines as QT_n (n=5)

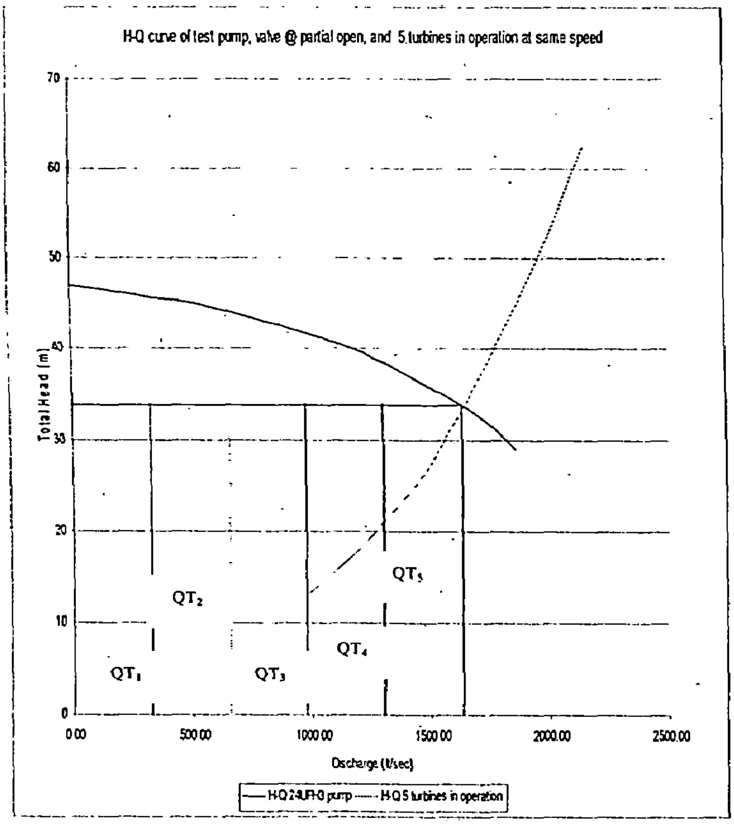


Fig. 5.11 H-Q curve of recovery system (5 turbines operating) and flow distribution in turbines

When two turbines (PATs) are shut down and the remaining turbines are operated in the normal speed, the h-q curve of the system that will be followed is shown in Fig. 5.12. When h-q curve of valve is drawn from the operating point of the recovery system, it remarkably collapses with the h-q curve of the system. The turbines (PATs) in combination replicate valve characteristics at that operating point too. Fig. 5.13 shows the flow distribution for four turbines (PATs) in operation in normal speed. It is also seen that for individual turbines (PAT), head acting on it and discharge both are increasing following the turbine (PAT) h-q characteristics at that speed.

Power produced by a turbine (PAT) at 337.5 lt/s. flow rate and 37.5 m head is obtained form the performance curve of the turbine (PAT) from Fig. 5.7 and is found to be 87 kW at 80%efficiency. Thus the total power produced is 4 X 87= 348 kW.

Superimposed h-q curve (test pump, valve at partial opening and system)

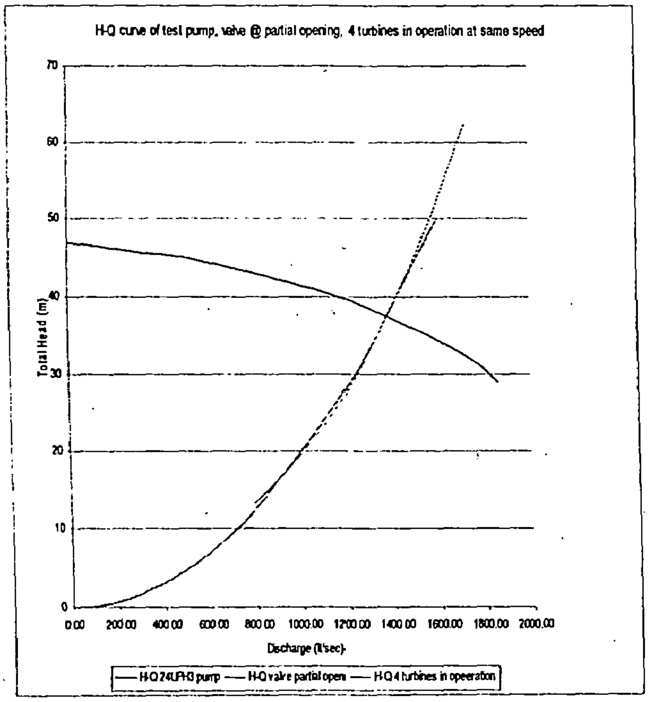


Fig. 5.12 H-Q of system (4 turbines operating) and valve at partial open position

Superimposed h-q curve (test pump and system) and flow distribution for turbines as QT_n (n=4)

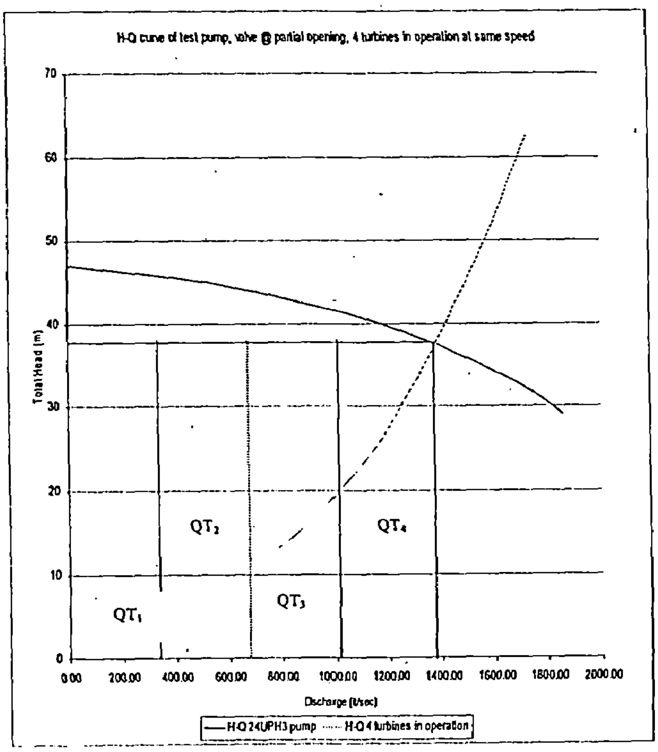


Fig. 5.13 H-Q curve of recovery system (4 turbines operating) and flow distribution in turbines

When three turbines (PATs) are closed and the remaining turbines (PATs) are operated at the normal speed, the recovery system has another operating point. The h-q curve that the system follow is in Fig. 5.14.The curve when superimposed with the h-q curve of valve at that operating point, the recovery systems' h-q curve almost coincides with valve thus replicating valve characteristics at that operating point. The flow distribution for 3 turbines (PATs) system is shown in Fig. 5.15.

Superimposed h-q curve (test pump, valve at partial opening and system)

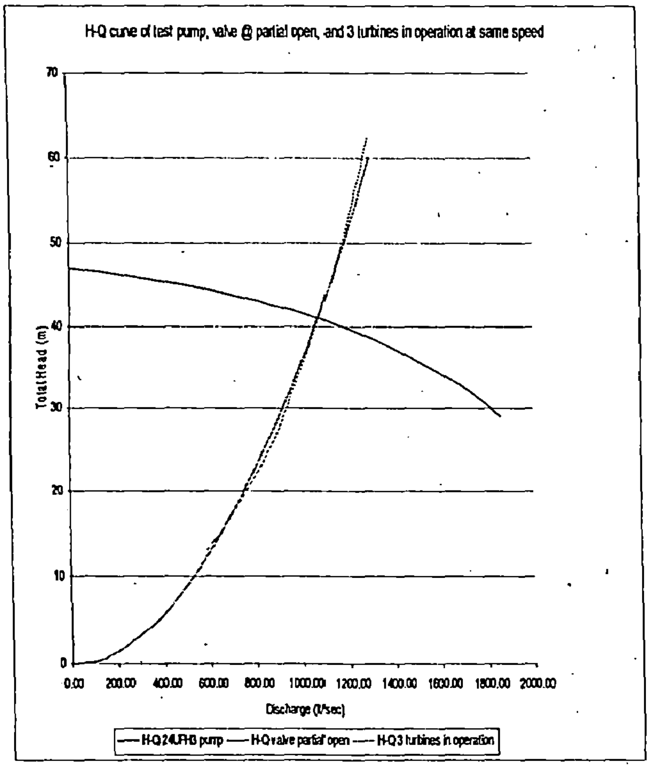


Fig. 5.14. H-Q of system (3 turbines operating) and valve at partial open position

Power produced by a turbine (PAT) at 355 lt/sec. discharge and 41 m head is obtained from the performance curve of the turbine (PAT) in Fig. 5.7 and is 102 kW at 80% efficiency. The total power produced by 3 turbines (PATs) is 3 X 102= 306 kW.

When four turbines (PATs) are shut down and the remaining are operated in the normal speed, the system h-q curve that will be followed is shown in Fig. 5.16. When h-q curve for valve is drawn form that operating point of the recovery system, a remarkable collapsibility with the h-q curves of recovery system is observed. The turbines in combinations replicate valve characteristics at that operating point too. Fig. 5.17 shows the flow distribution for two turbines (PATs) in recovery system. It is also seen that for individual turbines (PAT), both head acting on it and discharge are increasing following the turbine (PAT) h-q.

Superimposed h-q curve (test pump and system) and flow distribution for turbines as QT_n (n=3)

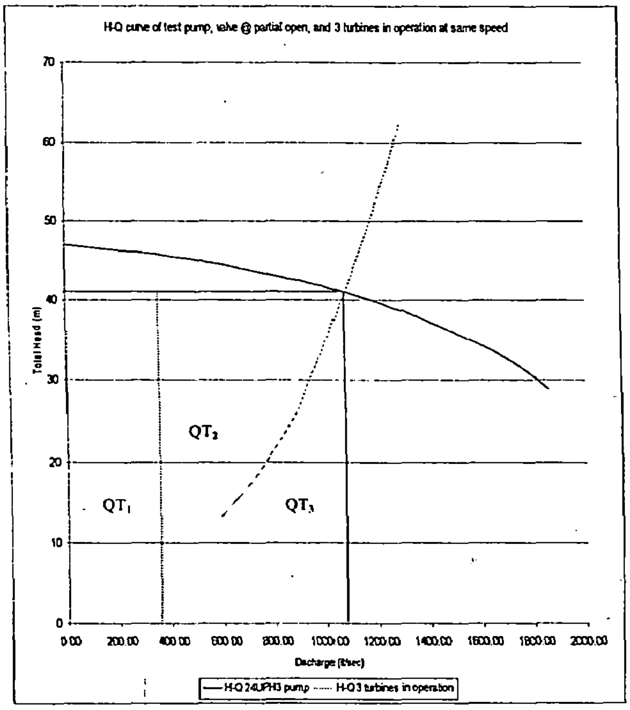


Fig. 5.15 H-Q curve of recovery system (3 turbines operating) and flow distribution in turbines

Superimposed h-q curve (test pump, valve at partial opening and system)

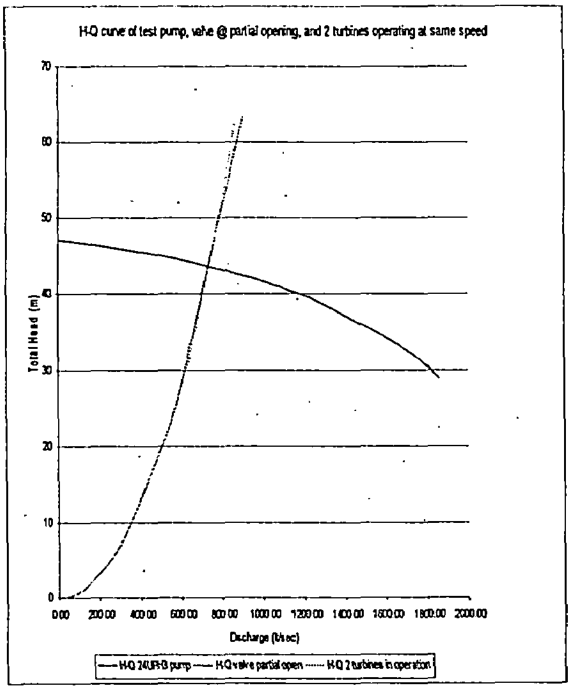


Fig. 5.16 H-Q of system (2 turbines operating) and valve at partial open position

Superimposed h-q curve (test pump and 2turbines) and flow distribution for turbines as QT_n (n=2)

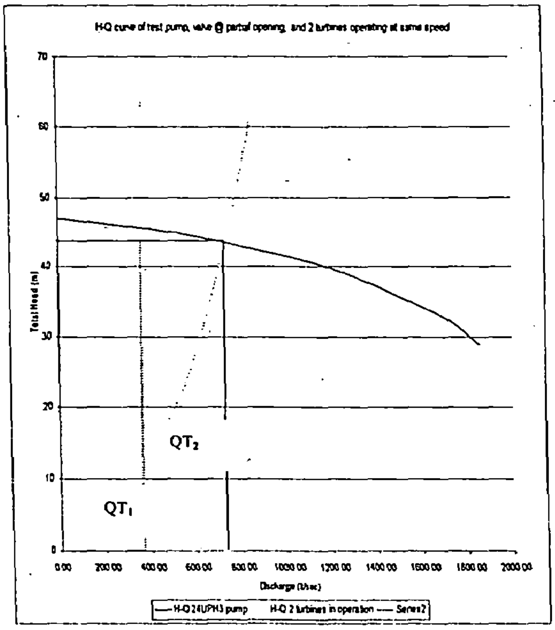


Fig. 5.17 H-Q curve of recovery system (2 turbines operating) and flow distribution in turbines

Power produced by a turbine (PAT) at 362.5 lt/s. discharge and 44 m head is obtained form the performance curve of the turbine (PAT) from Fig. 5.7 and is found to be 112 KW at 79% efficiency. Thus the total power produced by recovery system is 2 X 112= 224 kW.

When five turbines (PATs) are closed and the remaining are operated at the normal speed, the recovery system has another operating point. The h-q curve is shown in Fig. 5.18 and the curve when superimposed with the h-q valve at that operating point, it almost coincides, replicating valve characteristics. The flow distribution for one turbine (PAT) in recovery system is shown in Fig. 5.19.

Power produced by a turbine(PAT) at flow rate of 378.5 lt/s. and 46 m head is obtained form the performance curve of the turbine (PAT) from Fig. 5.7 and is found to be 137 kW at 78% efficiency. Thus the total power produced by the system is 1 X 137= 137 kW.

During recovery process, a few near by points from the normal h-q point (obtained form normal speed) can be obtained by varying speed. Variation of speed is possible

Superimposed h-q curve (test pump, valve at partial opening and system)

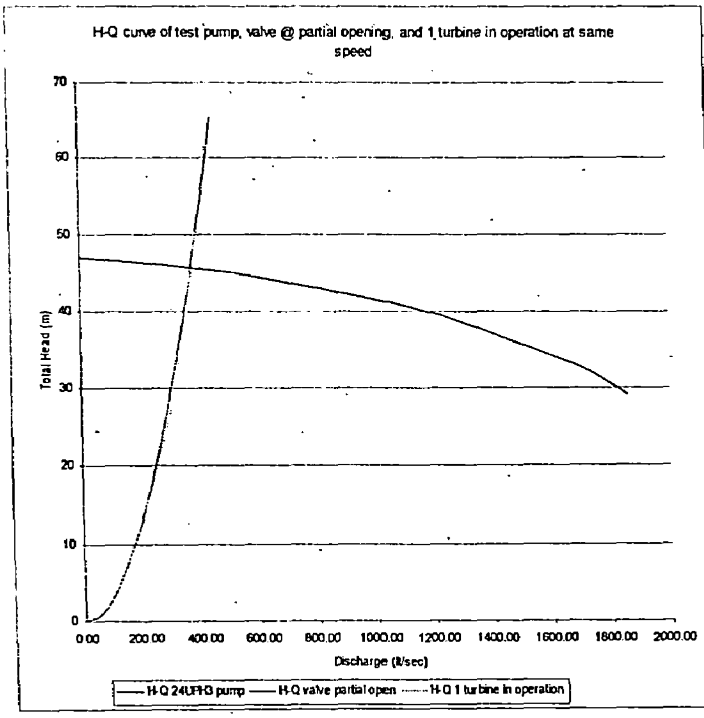


Fig. 5.18 H-Q of system (1turbine operating) and valve at partial open position

Superimposed h-q curve (test pump and system) and flow distribution for turbines as QT_n (n=1)

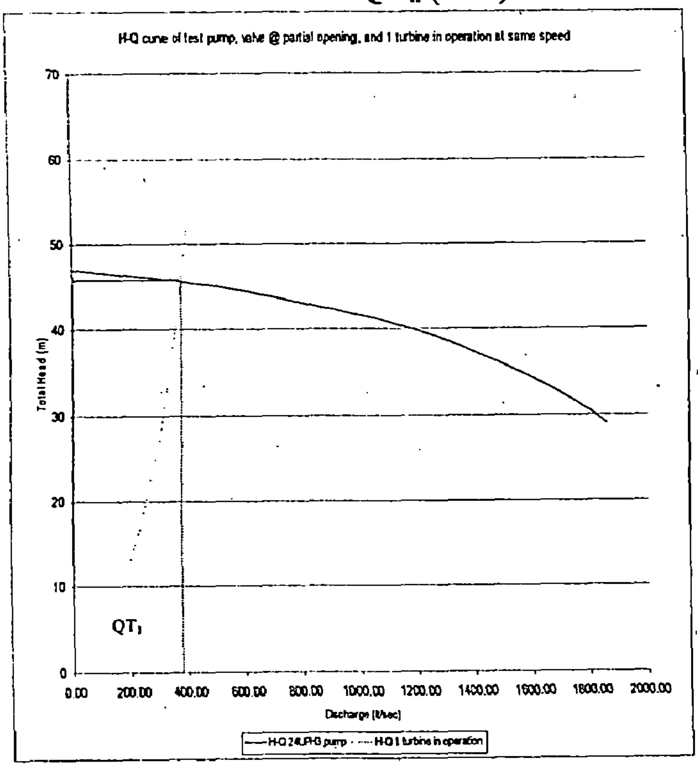


Fig. 5.19 H-Q curve of recovery system (1 turbine operating) and flow distribution in turbine

by loading and unloading the system. The change in speed of system cause h-q curve to shift from the normal h-q to either left or right depending upon increase or decrease of speed.

When all turbines (PATs) are closed, a complete shut off is created and head produced by the test pump is maximum and is called shut of head. The h-q curve of test pump is obtained by joining these operating points starting from 6 turbines (PATs) in operation to shut off point. Power- discharge and efficiency-discharge curve is obtained by measuring the power consumed by test pump and energy utilized with respect to supplied to test pump in building head and discharge respectively at different operating points. Fig. 5.20 shows a typical energy recovery system that test pumps and recovers a portion of energy used during testing.

5.5 EFFICIENCY OF ENERGY RECOVERY

With above combination of turbines (PATs), energy is recovered at various operating points which vary from lower side of 0 kW when all turbines (PATs) are closed to as high as 402 kW when all turbines (PATs) are operated. Table 5.2 shows the energy recovered and efficiency of the recovery system at different combination of turbines (PATs) operation.

Table 5.2

Power out-put of energy recovery system and efficiency of energy recovery at various combinations of turbines (PATs) operation

Operating Point (normal speed)	0 Turbine Working	1 Turbine Working	2 Turbines Working	3 Turbines Working	4 Turbines Working	5 Turbines Working	6 Turbines Working
	1	2	3	4	5	6	7
Power Input (kW)	300	350	420	500	600	610	615
Power Output (kW)	0	137	224	306	348	390	402
Efficiency in %	0.00	39.14	53.33	61.20	58.00	63.93	65.37

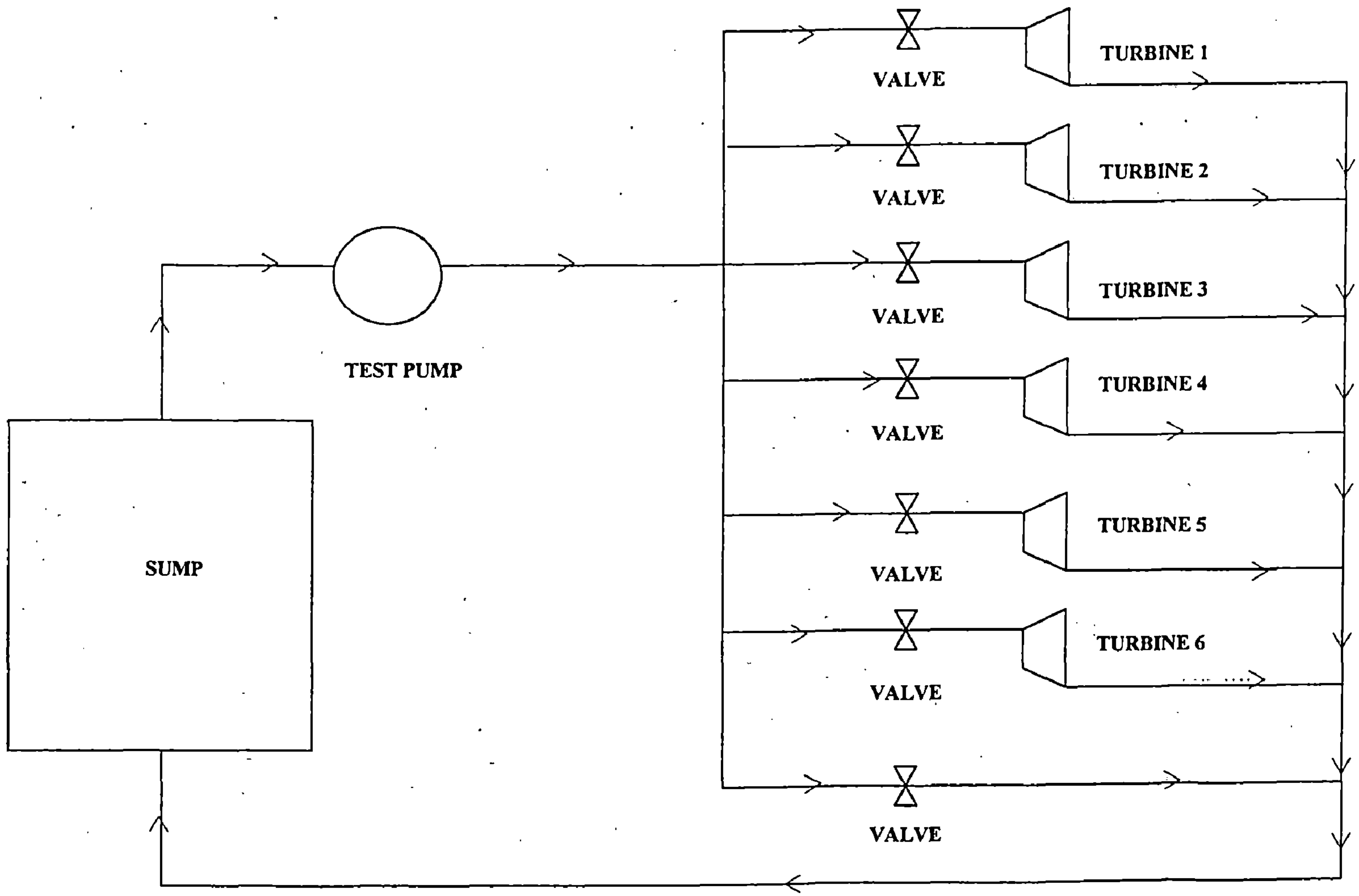


Fig. 5.20 Energy recovery system that tests 24UPH3 test pumps

6. CONCLUSIONS AND RECOMMENDATIONS

6.1 CONCLUSIONS

High-energy consumption during pump test results in added cost to buyer. Further, energy being used in testing is also wasted. Energy recovery during pump test reduces the energy demand, which can contribute to the reduction of cost of testing, and production cost of pump finally.

The results obtained during experiment have proved that a turbine, speed when varied, can regulate flow but has limited range. It is also seen that regulation range varies with capacity of turbine. High capacity turbines can regulate flow in higher discharge range and low capacity in lower discharge range. Turbines operations by arranging in parallel have higher capacity and have operating point at high discharge range. Shutting down these turbines progressively causes reduction in capacity and head acting on it get increased. Performance curve for the test pump can be drawn using turbines in parallel and by progressively closing them.

The recovery system produces maximum power when all turbines are in operation. Although power produced by individual turbine go on increasing with closure of turbines but the total power produced by the system get reduced than when all turbines are in operation. The increase in power produced by the individual turbine is due to increase in both discharge and head acting on it. Thus, when the system is designed, care should be taken such that the system has maximum efficiency at total maximum discharge and low head. Hence the turbines selected should have maximum efficiency at the lowest discharge and head available to it at that discharge so that the total power produced is the highest.

If turbines of different capacities are used in parallel, with different operation combination more number of operation points can be obtained which helps in obtaining smoother characteristic curve.

6.2 RECOMMENDATIONS

- The experimental-set up could not be arranged to test the energy recovery system using parallel combination of pump as turbine. Experiment is recommended to be conduct for finding out the how closely the results match with the theoretical results.
- Performance characteristics curves of pump as turbine are generally not available; more of such curves can be developed by conducting experiments and used them for designing energy recovery system for various test pumps efficiently.

References:

1. Antwerpen Van, H. J.and Greyvenstein, G. P., "*Use of Turbines for Simultaneous Pressure Regulation and Recovery in Secondary cooling Water System in Deep Mines*" Energy conversion and Management, 7 June 2004
2. Chapallaz, J. M., Eichenberger, P., Fischer, G, "*Manual on Pumps Used as Turbines*" Deutsches Zentrum Fur Entwicklungstechnolgien (GTZ), 1992
3. Chappell J. R., Hickman W. W. and Seegmiller D. S. "*Pump as Turbine Experience Profile*", U. S. Department of Energy, 1982
4. Daugherty, R. L., "*Hydraulic Turbines with a Chapter on Centrifugal Pumps*", Mc.Graw-Hill Book Company, Inc. New York, 1920
5. Jain, V. K., "*Pump Theory and Practice*", Galgotia Booksource, New Delhi, 1987
6. Karassik, I. J., KRutzsch, W. C., Fraser, W. H. and Messina, J. P. "*Pump Handbook*", Mc Graw Hill, Inc, USA, 1976
7. Kirloskar Brothers Limited, "Pump Test House Hydraulic Research Centre" Kirloskar Brothers Limited, Pune, India, 2003
8. Kittredge, C. P. and Thoma, D., "*Centrifugal Pumps Operated Under Abnormal Conditions*", A.S.M.E, J . Eng. Power, 1991
9. Kittredge, C. P., "*Centrifugal Pumps Used as Hydraulic turbines*", A.S.M.E., January 1961
10. Knapp, R. T., Pasadena Calif, " *Complete Characteristics of Centrifugal Pumps and Their Use in the Prediction of Transient Behavior*", A.S.M.E., November 1937
11. Mayo, H. A., Whippen, W.G. "*Small scale hydro/ centrifugal pumps as Turbines*" Allis-Chalmers Corporation – York, PA, USA, 1981
12. Mayo, H. A, "*Small hydro-Using Pumps as turbines*", Pennsylvania Electric Association, Industrial and commercial Conference Harrisburg, Pennsylvania, USA , May 1982
13. Mehmet Kanoglu, "*Cryogenic turbine Efficiencies*" Exerg, International Journal, 1(3) (2001)
14. Pacific Pumps, "*Hydraulic Power Recovery turbines (HPRT)*", Pacific Pumps Division, Huntington Park, California, 1979

15. Saini, R. P. and Ahmad, N, "*Method for Selection of Pumps Used in Turbine Mode*", First International conferernce on renewal Energy-Small Hydro, Hyderabad, India 3-7 February 1997,

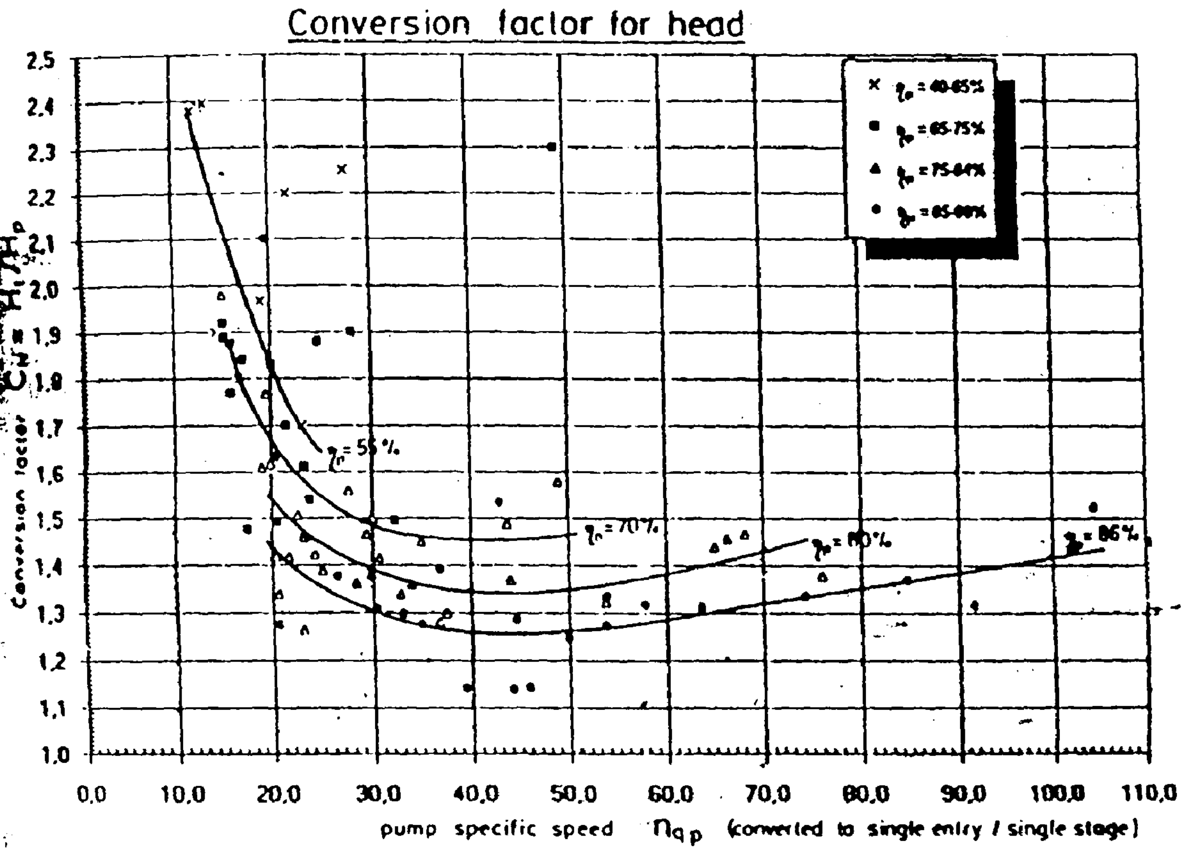
16. Singh, Punit, Kshirsagar, J. T., Nestman, Franz, "*Experiment and Numerical Studies on a Pump as Turbine*" The 7th Asian International Conference on Fluid Mechanery, Fukuoka, Japan October 7-10, 2003

17. Stepanoff, A. J., "*Centrifugal and axial Flow Pumps, Theory, Design and application*" John Wiley & Sons, INC. New York 1957

18. Warring, R. H., "*Hand book of valves, piping and pipelines*" Gulf Publishing Company, Huston, Texas, 1982

19. Zappe,R. W, "*Valve Selection Handbook*", third Edition, Gulf Publishing Company, Huston, 2003

Appendix-A1



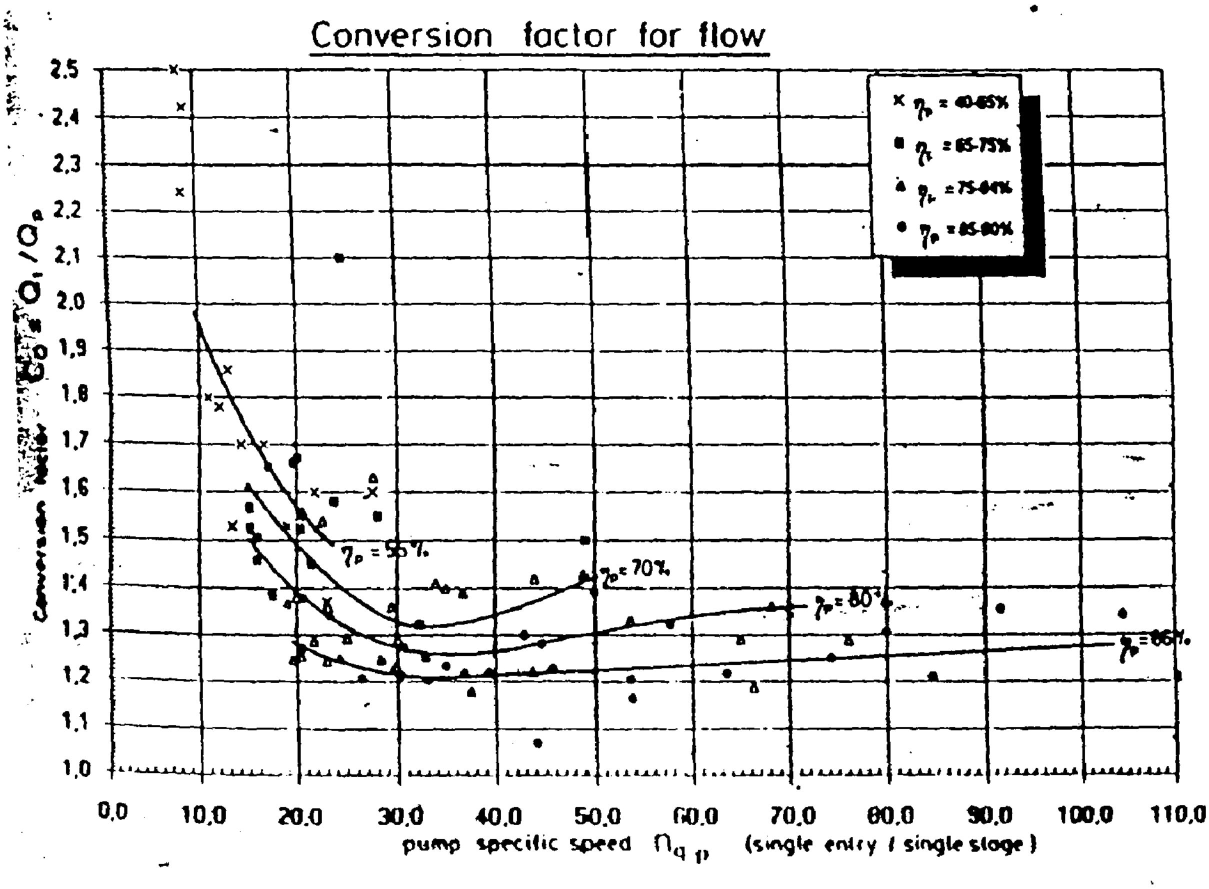


Fig. A1 The turbine mode performance (conversion factors related to pump best efficiency) in function of pump specific speed and maximum pump efficiency (Chapallaz) [2]

Appendix- B1

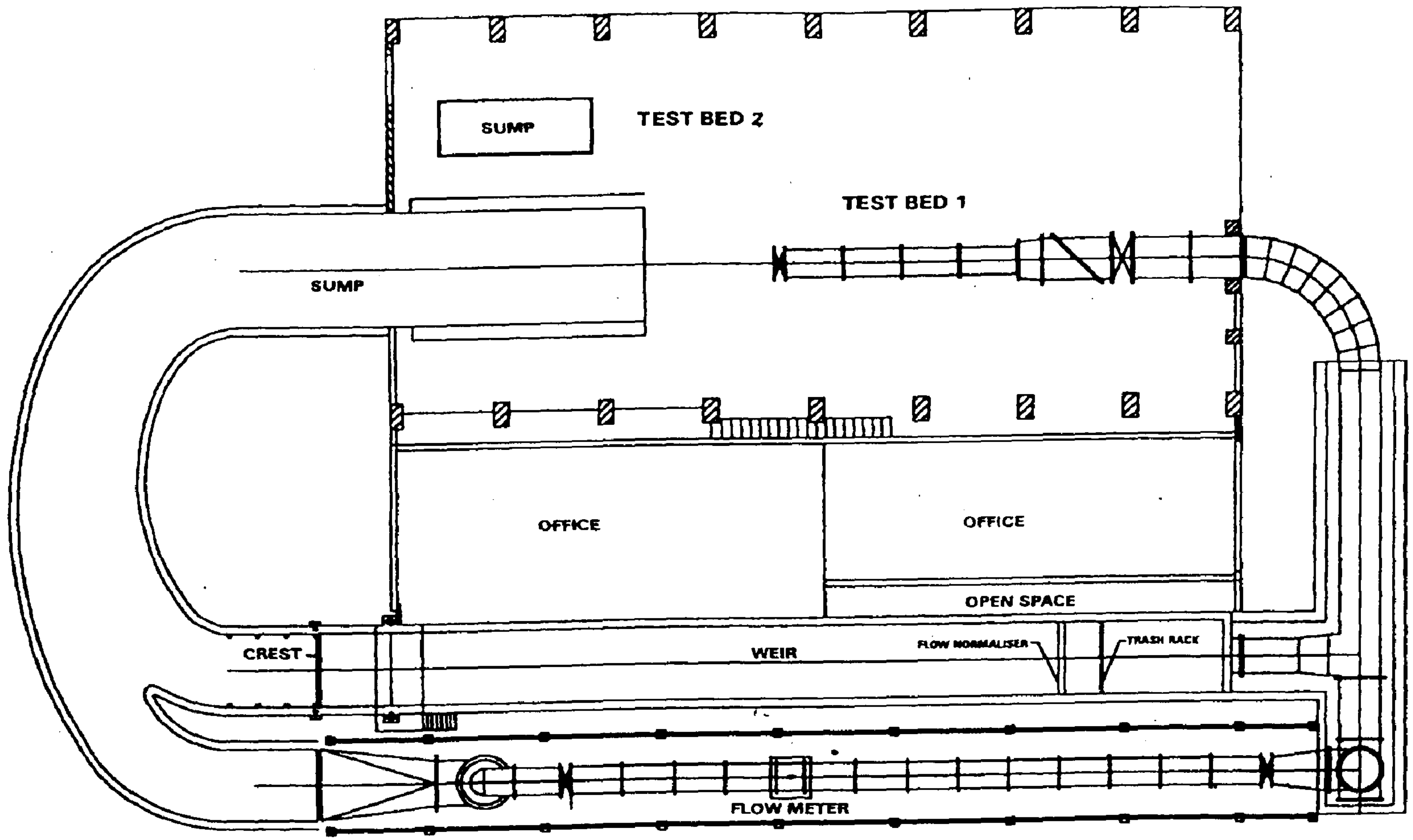


Fig. B2 Layout of large test bed (Kirloskar Brothers Limited,Pune) [7]