MHPG Series

Harnessing Water Power on a Small Scale

Volume 11

Manual on Pumps Used as Turbines

J.-M. Chapallaz, P. Eichenberger, G. Fischer

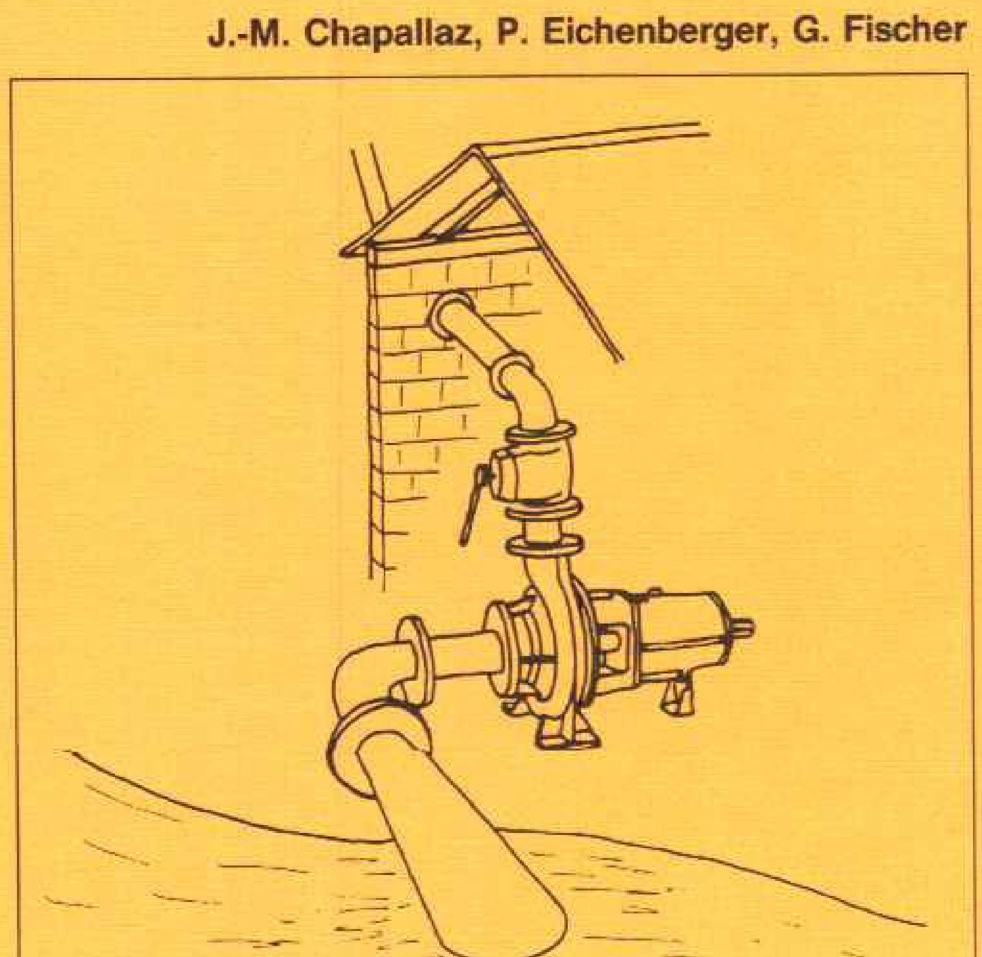

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TABLE OF CONTENTS

APPENDIX A: Essential Theory of Engineering Hydraulics

APPENDIX B: Basic Theory of Hydraulic Machines

APPENDIX C: Pump Designs and Characteristics

LIST of PRINCIPAL SYMBOLS

Symbol	*SI Unit*	*Meaning*
A	m^2	Area
b ,B	m	Width
c	m/s	Absolute flow velocity in pump impeller
d, D	m	Diameter
F	N	Force
g	m/s^2	gravitational acceleration $g = 9.81\ m/s^2$
GD^2	kgm^2	Moment of gyration $GD^2 = 4\ J$
H	m	Head
J	kgm^2	Mass moment of inertia
L	m	Length
M	Nm	Bending moment
n	1/min , rpm	Rotational speed
P	kW	Power, Output
p	N/m^2	Pressure
r	m	Radius
Re	-	Reynolds number
T	Nm	Torque
t	s	Time
T_a	s	Unit acceleration time
T_r	s	Reflection time
u	m/s	Tangential velocity in pump impeller
Q	m^3/s	Flow
v	m/s	Velocity of flow in pipes, valves
w	m/s	Relative velocity in pump impellers
z	m	Distance to reference plane / height
α	°	Flow angle
β	°	Vane angle
η	-	Efficiency
λ	-	penstock friction coefficient
ρ	kg/m^3	Density
ω	rad/s	Angular speed
ζ	-	Loss coefficient (related to the square of flow velocity)
σ	-	Sigma number ; Thoma number

Subscript	*Meaning*
0	Value at specified speed
1	Suction side of pump / PAT impeller
2	Pressure side of pump / PAT impeller
a	Acceleration T_a = Unit Acceleration Time
av	available
h	hydraulic
n	nominal value
p	Pump mode
r	Reflection T_r = Reflection time
req	required
R	Runaway conditions
t	Turbine mode
u	tangential direction
m	meridian direction (perpendicular to cross section)
v	vapour
atm	atmospheric

LIST of ABBREVIATIONS

IGC	Induction Generator Controller
ELC	Electronic Load Controller
IMAG	Induction Motor used as Generator
MHP	Micro-Hydro Power or Micro-Hydropower Plant
NPSH	Net Positive Suction Head
O&M	Operation & Maintenance
PAT	Pump used As Turbine
RPM	Revolutions Per Minute
TREH	Total Required Exhaust Head

0. PREFACE

In the discussion of the energy situation in developing countries and especially in rural areas, it is generally recognized that small hydropower may play a significant role. However, high initial investment costs of small hydropower plants have restricted rapid development of this energy potential in many countries. The use of **standard pumps as turbines (PAT)** may often be an alternative with a considerable economic advantage and might therefore contribute to a broader application of micro-hydropower . This handbook provides a practical method enabling engineers and technicians **to select a PAT** for a specific purpose. Furthermore, it covers all **aspects related to the installation, operation and control of the machine**; this is as important as the pure selection of the machine since a PAT is, in fact, used for a purpose which it was not explicitly designed for.

The handbook has been written to meet the special, but not exclusive, needs of all those engineers and technicians, engaged in **hydropower projects in developing countries**. It is not only intended for the narrow grouping of hydropower specialists but rather for all those faced with the problem of energy production or recovery. The formal hydraulic theory used in the handbook has been simplified in order that the non-specialized mechanical, electrical, civil, rural or agricultural engineer should be able to follow all aspects covered by the handbook and to undertake the necessary computations without difficulty. Moreover, the Appendices A and B provide a short introduction into basic hydraulic engineering and the theory of hydraulic machines which should be studied by those not being familiar with hydropower previous to reading the main text. Worked examples at the end of the handbook and a step-by-step calculation procedure with diagrams developed into working tools are **designed for the practicing engineer** using the handbook as a concise working reference.

S.I. units have been used throughout the book and standard symbols for physical properties employed.

The handbook has been written by a team of mechanical and civil engineers specialized in micro-hydropower development. Apart from the input of their own field of experience, a considerable amount of documents and data on pumps operated as turbines has been gathered and incorporated critically in the manual.

The selection charts presented in the manual are based on more than 80 test results of which about 50 were supplied by Mr. P. A. Dutoit of C.I.M.H., France. He deserves special thanks from the authors for this valuable contribution.

The authors would like to express their indebtedness to Dr. Peter Baz and Mr. Klaus Rudolph (GTZ/GATE) who initiated this project and provided the financial means to establish the manual.

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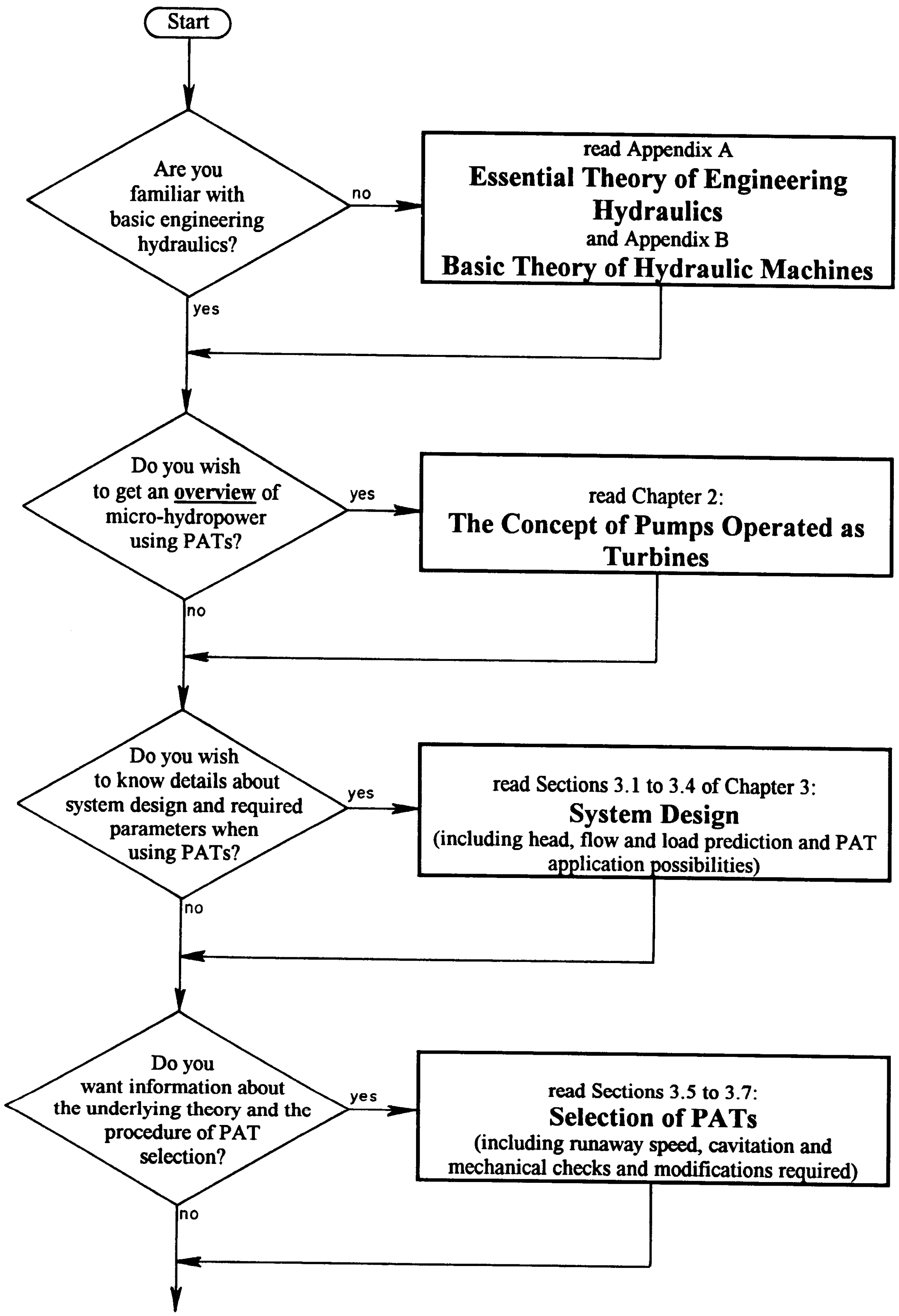
- Mr. Alatorre-Frenk, University of Warwick, UK

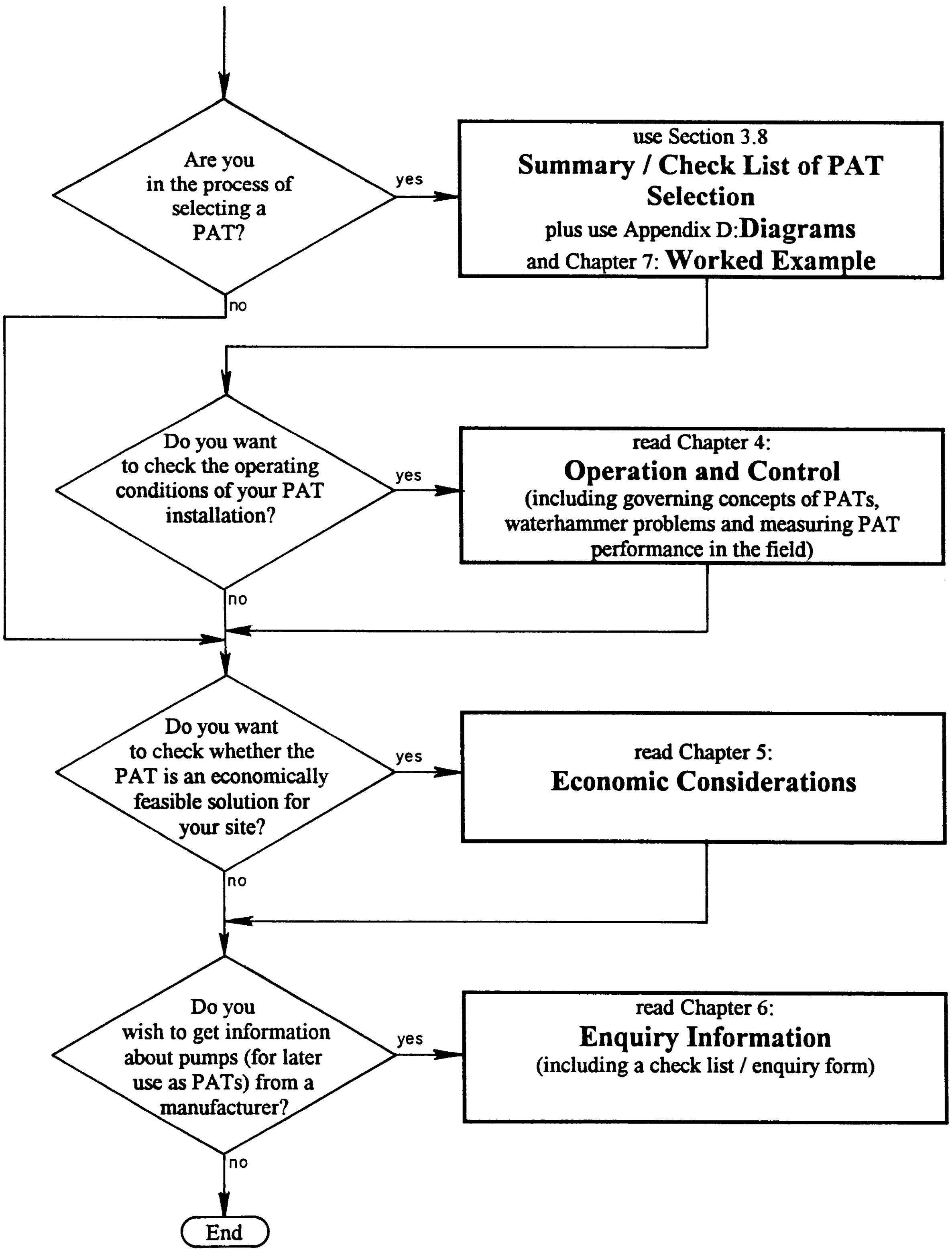
- *Dipl.-Ing.* W.-P. Strate, University of Hannover, Germany

- Messrs. Alex Arter and Jorge Senn, SKAT, Switzerland

HOW TO USE THE MANUAL

FIGURE 0.1 : Guide through the handbook




Are you in the process of selecting a PAT?
yes
use Section 3.8
Summary / Check List of PAT Selection
plus use Appendix D: Diagrams
and Chapter 7: Worked Example
no
Do you want to check the operating conditions of your PAT installation?
yes
read Chapter 4:
Operation and Control
(including governing concepts of PATs, waterhammer problems and measuring PAT performance in the field)
no
Do you want to check whether the PAT is an economically feasible solution for your site?
yes
read Chapter 5:
Economic Considerations
no
Do you wish to get information about pumps (for later use as PATs) from a manufacturer?
yes
read Chapter 6:
Enquiry Information
(including a check list / enquiry form)
no
End

1. INTRODUCTION AND SUMMARY

1.1 Why Using a Pump as a Turbine

A pump may have the following advantages over a conventional turbine:

- Economic advantages

  Purpose-built turbines are more expensive than standard pumps of comparable size because:

  - turbine manufacturers are few

  - market for turbines is small compared to the market for pumps (no series production of turbines)

- Availability

  The availability of pumps and their spare parts is far better than that of turbines, especially in developing countries

- Construction

  Standard pumps are simple and sturdy and do not require highly qualified mechanics for maintenance; this makes standard pumps used as turbines more appropriate for developing countries than sophisticated turbines

  (details see section 2.3)

1.2 Fields of Application of Pumps Used as Turbines (PATs)

Two broad categories of practical applications of pumps used as turbines may be distinguished:

- <u>**Power generation**</u> as the primary objective of an installation

  - Pump-turbines have been used for several decades in large pumped storage schemes with power ratings of several megawatts. Pump-turbines are specifically designed to operate in both modes, pumping water into an elevated storage lake overnight at low tariff electricity and, during the day, generating peak demand electricity through the same machine operating in turbine mode.

  - Standard pumps not intentionally designed to operate as turbines are now more and more used in micro-hydropower (MHP) schemes (<u>**5 to 500 kW**</u>) due to their advantages mentioned above. Applications range from direct drive of machinery in agro-processing factories and small industries (flour mills, oil expellers, rice hullers, saw mills, wood and metal workshops) to electricity generation both in stand-alone and grid-linked stations. The focus of this handbook will mainly be on this category of applications.

- <u>**Hydraulic power recovery**</u> as a by-product of an installation which was not designed to generate energy in the first place

  A common feature of these installations is the need to reduce pressure of a fluid at the end of a process or before further application or treatment of the fluid. The conventional approach would dissipate this surplus pressure in any kind of throttling device (valve, orifice, stilling basin). Using a reverse running pump as a hydraulic power recovery turbine (HPRT) recovers this energy at almost no costs thereby improving overall plant efficiency. Interest on this kind of application has grown recently mainly as a result of increasing energy prices and energy conservation awareness.

Examples of fields of application:

- **Irrigation** schemes usually involve the distribution of water to different topographical levels. For reasons of costs and complexity of systems, the total discharge required is usually pumped (or conveyed by gravity) to the highest point of the scheme and from there distributed to lower levels requiring dissipation of energy. By using a PAT instead of pressure reducing valves or stilling basins, significant amounts of energy can be recovered.

- **water supply and sewage disposal systems** in mountainous regions may involve large differences in elevation between the source and the end-user or the treatment works. These hydraulic pressure heads might be exploited for energy recovery purposes without any negative effects on the operation of the systems.

- **industry:** In many process applications, a fluid must be kept at a high pressure during the actual process but must be let down afterwards; this provides a vast field of applications for HPRT:

- Petro-chemical processes (e.g. discharge of hydrocracker residuals)

- Reverse osmosis in desalination plants (pressing sea-water through a semi-permeable membrane to obtain potable water)

- Gas scrubbing (unwanted gases in gas mixtures are dissolved in a fluid at high pressure)

- Mining [to provide supportable working conditions in deep mines (1500 to 3000m below surface level) cooling water is delivered into the mine thereby attaining high pressures which have to be released]

1.3 The Difference between Pump- and Turbine-Mode Performance

Rotational fluid machines are completely reversible and a pump can run effectively as a turbine. However, performance in both modes are not identical although the theory of ideal fluids would predict the same . Without exception, the optimum flow and head in the turbine mode is greater than in pumping mode. The main reason for this difference is related to the hydraulic losses of the machine:

- When operating at the best efficiency point (optimum flow pattern through the machine at nominal speed) in the pump mode, the output head is (compared to the ideal conditions according to theory) **reduced** by the hydraulic losses (friction and volumetric losses).
- When operating at the same speed in turbine mode, the pressure head required on the machine to operate at the best efficiency point must be **increased** by the hydraulic losses as compared to the ideal fluid. Thus, the head in pumping mode differs from the turbine head about twice the hydraulic losses ($H_p \approx H_t * \eta^2$).

This general law is not entirely true. The energy transfer between the fluid and the impeller is mainly determined by the shape (angle) of the vanes at the **impeller outlet in the pump mode.** When flow is reversed in the **turbine mode, it is the shape of the pump casing (volute)** which determines the energy transfer. Different pump designs and manufacturing details will therefore affect performance: machines may have similar performance in pump mode (similar impeller) but will not necessarily yield the same performance in the turbine mode.

1.4 Selecting a PAT

Since the application of pumps used as turbines is not yet as widespread as the use of the same machine in its normal operation, performance curves are usually only available for the pump mode. Methods to predict turbine mode performance from pump mode data have therefore been developed. Due to the effect of the different pump design details mentioned above, neither of the methods developed so far are 100 % reliable; errors from predicted to actual flow and head in turbine mode may be as high as 10 % and more. The only way to obtain reliable results is to measure the performance of the machine in its desired sense of operation, which would, however, offset a large part of the low-cost advantage of the PAT and is therefore not feasible for micro-hydropower applications.

This handbook presents a PAT selection method based on two parameters of the pump mode performance; i.e. best efficiency and specific speed. While the latter 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. However, uncertainties in the prediction of the turbine mode performance using pump characteristic curves still remain.

Due to the inaccuracy of the selection methods, the layout point of the PAT must be chosen in such a way that a deviating performance of the installed machine does not have negative effects on the operation of a plant or that performance could be corrected. This may be achieved by selecting a PAT which will operate slightly in the overload range in turbine mode (beyond its best efficiency point).

1.5 Design Changes

In most instances, no design changes or modifications need to be made for a pump operating as a turbine provided that selection has taken into account the higher operating head and power output of the machine in turbine mode and consequently, nominal turbine speed has been taken well below maximum permissible pump speed. However, a design review is also required to check any adverse effects occurring from the reverse rotation in turbine mode. Thus, a design review includes items such as:

- Checking that threaded shaft components cannot loosen
- Evaluation of the bearing design
- Checking shaft stress due to increased power output
- Checking effects on shaft seals due to increased pressure

1.6 Type of Pump to Be Used and Efficiency in Turbine Mode

Virtually any type of pump may be used as turbine. However, the main advantage of a PAT, i.e. lower costs than a conventional turbine, is very pronounced for standard centrifugal and mixed flow pumps whereas axial flow pumps are less advantageous in that respect; furthermore, data on their turbine mode performance are limited and therefore add additional uncertainty to their application as a PAT. The vast field of different pump designs and power ranges provides a suitable PAT for almost any application with heads from about 10 m up to several hundred meters (with multi-staging). Large flows may be accommodated with double-flow pumps. Even submersible pumps may be used as PATs which, when integrated in the water course or pipe system, are completely hidden away underground, an important factor for the conservation of the environment.

Efficiencies of pumps used as turbines may be the same as in pump mode but are more often several percent (3 - 5%) lower.

1.7 Operation and Control - Limits of Application

Although the PATs cover a wide range of the small hydro-power domain, they cannot replace conventional turbines. Since PATs have no hydraulic control device such as guide vanes, they are usually unsuitable to accommodate variable flow conditions. Throttling flow by means of a control valve in the penstock is inefficient and only applicable over a small range. Part flow conditions will therefore remain the domain of purpose-built turbines.

The lack of a hydraulic control device of a PAT has long been seen as a disadvantage also in terms of constancy of PAT speed under variable load. Grid-linked electricity generation or direct drive of machinery are either constant load applications or do not require precise speed control. These applications are therefore very suitable for PATs. Stand alone electricity generation on the other hand requires some form of governing to keep voltage and frequency within acceptable limits under changing load. The use of PATs in free-standing electricity generation is, however, not excluded due to the recent development of electronic load controllers which provide effective governing in conjunction with both induction and synchronous generators. Electronic load controllers keep the load on the PAT constant by switching in ballast loads whenever the electricity demand of the consumers drops. Dissipating generated energy instead of reducing flow at times of low power demand (as has been done for decades with conventional turbines and mechanical governors) is only an acceptable solution when maximum water saving is not required (e.g. installations without storage pond).

Just as with a conventional turbine, a number of operating conditions must be checked also when using a PAT. The location of the PAT above the tail water level determines the minimum backpressure on the pump which must be higher than the total required exhaust head (TREH) of the machine to avoid cavitation. Tests have shown that TREH in turbine mode is usually somewhat lower than the equivalent value in pump mode, i.e. the net positive suction head (NPSH).
Another condition to be checked is the runaway speed, which is the maximum speed at which the PAT would operate if no load were applied and it has no direct analogy in the pump mode. It has been found that runaway speeds for PATs are between 120 and 160 % of nominal speed.
A PAT might accelerate to runaway speed after full load rejection, i.e after a failure of the transmission belt, grid failure or idling processing machines coupled to the PAT. It must therefore be checked whether all rotating elements including generator or machinery can stand this overspeed. Due to the relatively low inertia of the rotating parts of a PAT, the machine might reach runaway speed in a very short time which might induce pressure surges in the penstock. Analysis of this transient behaviour must prove that the structural integrity of the installation as a whole is ensured (waterhammer, transient speed rise).

1.8 Economic Considerations

In some applications, the price advantage of a PAT might partly be offset by a lower annual energy production due to a lack of hydraulic control and lower part flow efficiency of the PAT. Simple comparison of the purchasing price of the PAT and the corresponding conventional turbine is not sufficient to determine the most economic machine. To arrive at a more reliable statement of the economic viability of either the PAT or the conventional turbine, the total project costs and benefits must be identified and compared with each other. This might include lower O&M costs due to e.g. the simple design of a PAT which does not require specialized mechanics for maintenance and repair.

2. THE CONCEPT OF PUMPS OPERATED AS TURBINES

2.1 General

The principal decision whether to use a conventional turbine or a pump operated as a turbine (PAT) at a given MHP site is governed by the following parameters:

PLANNING

Availability of know-how

Availability of system components

ENGINEERING DESIGN

Site parameters (head, flow,)

Stream flow characteristics (seasonal flow variations)

Feasibility of implementation

Power (output) requirements - load curves - Station factor

OPERATION AND MAINTENANCE (0 & M)

Regulation of speed and/or flow

Maintenance skills available

Reliability of system components

Availability of spare parts

ECONOMICS

Initial investment costs

O & M costs

Energy production - Annual Returns

Net benefit - Capital recovery

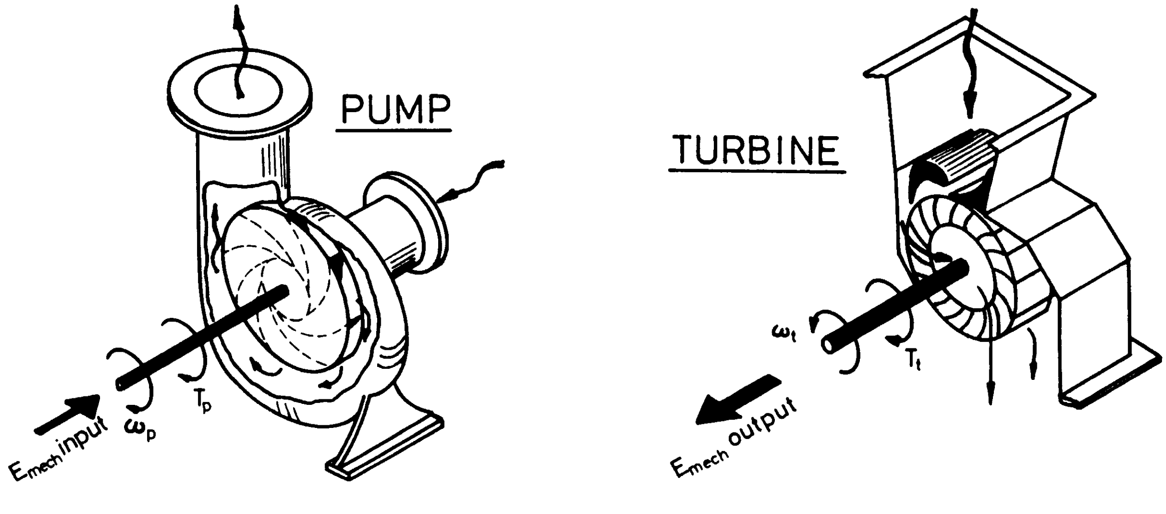
FIGURE 2. 1 :
Parameters influencing the choice between a conventional turbine and a pump operated as a turbine (PAT)

Using a pump as a turbine (PAT) in an MHP is not merely a replacement of a conventional turbine by another machine which might be a little less expensive and sophisticated. As can be seen from Figure 2.1 above, nearly all aspects of a MHP, from design and implementation to O & M, are influenced by the choice of the machine.

This chapter works out the main differences between a pump and a turbine, presents some basic pump designs and indicates where a PAT can have distinct advantages over a conventional turbine.

2.2 The difference between pumps and turbines

The basic hydraulic theory is the same for both machines. However, the behaviour of real fluid flow including friction and turbulence results in different rules for the design of pumps and turbines.



	Turbine	**Pump**
Flow of energy	Input of hydraulic energy (water under pressure) Output of mechanical energy (torque on the shaft)	Input of mechanical energy (torque on the shaft) Output of hydraulic energy (water under pressure)
Hydraulic pressure head	Available turbine head decreases with increasing flow (friction losses)	Total dynamic head to be generated by the pump increases with increasing flow
Direction of rotation	Turbine runner rotates in the OPPOSITE direction of a pump impeller	
Direction of torque	In both modes the same direction	

(for more details on engineering hydraulics and hydraulic machines see Appendices A and B)

FIGURE 2.2 :
Main difference of fluid and energy flow in pumps and turbines

<u>Some particular differences</u>

a) Operating conditions:
<u>PUMPS</u> are usually operated with constant speed, head and flow. A pump is therefore designed for one particular point of operation (duty point) and does not require a regulating device (guide vane). Ideally, the duty point coincides with the maximum efficiency of the pump.

<u>TURBINES</u> operate under variable head and flow conditions. In an MHP, flow must be adjustable to either accommodate to seasonal variations of the available water or to adjust power output according to the demand of the consumers. Adjustable guide vanes and/or runner blades (or nozzles controlled by a streamlined valve) regulate the flow (see Figure 2.3).

b) Hydraulic design:
In a **PUMP**, kinetic energy imparted to the fluid must be converted into pressure energy ; i.e. flow must be decelerated as it passes through the impeller and the volute casing. Decelerated flow is generally very sensitive to separation and the formation of eddies. To avoid these, impeller passages are made of long smooth channels with gradually increasing cross-sectional area.
Friction losses through these long passages are relatively high (see Figure 2.3).
On the other hand, 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 (see also section 3.5.2).

c) Location of the machine - Cavitation
The physical location of the **PUMP** in relation to the water level of the sump from which water is being pumped is critical. If it is too high, cavitation may occur. In this context, suction pipe design is very important since friction losses in the suction pipe reduce pressure at the pump inlet and increase the tendency to cavitation.
TURBINES are less sensitive to cavitation since friction losses in the draft tube increase the backpressure on the turbine (see also Appendix A and section 3.6.2).

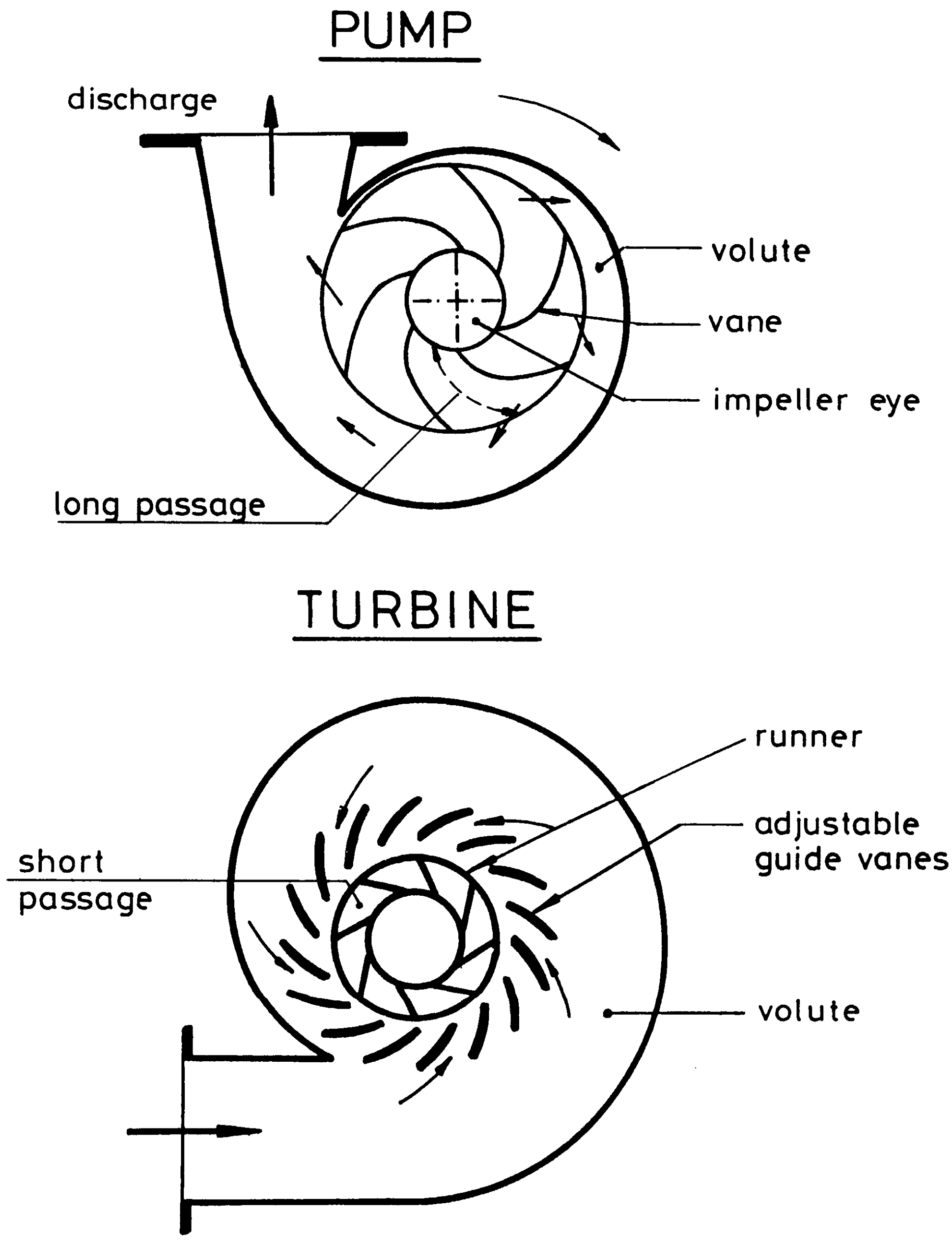


FIGURE 2.3 :
Comparison of flow conditions in a pump impeller and a turbine runner

2.3 A Pump Used as a Turbine

2.3.1 General

When operating a pump as a turbine, the flow pattern through the machine is similar to the flow conditions in a turbine. Thus, the advantageous features of turbines (i.e. accelerated flow and operating conditions which are less subjected to cavitation) also occur for PATs. On the other hand, the pump impeller with its long passages is basically not designed to run in reverse (increased friction loss) ; that is why a PAT may be slightly less efficient than a well-designed turbine.
Generally, it has been observed that efficiencies which can be expected from a particular pump in turbine mode are equal or slightly less than those in pump mode.

The comparison of the operating conditions of pumps and turbines in the previous section has indicated the major drawback of a PAT : because of the absence of a flow regulation device, PATs **require a fairly constant stream flow** throughout the year. When the available streamflow drops during the dry season, turbine flow must be throttled in order to avoid that the forebay is emptied and air is sucked in. Throttling discharge (e.g. by a control valve) results in a considerable drop of efficiency since, firstly, the pump spiral casing is not designed for flows deviating from design flow and, secondly, the throttling valve dissipates energy, i.e it reduces the net head on the PAT.

The main advantages and disadvantages of pumps used as turbines (PAT) are summarized hereafter.

2.3.2 Advantages

- **costs :** the investment costs of PATs may be less than 50% of those of a comparable turbine (especially for small units below 50 kW). This might be an important issue for projects with limited budgets and loan possibilities

- **construction:** the absence of a flow control device, usually felt as a drawback, is at the same time an advantage since the pump construction is usually simple and sturdy

- **availability:** due to their widespread application (irrigation, industry, water supply), standard pumps are readily available (short delivery times) and manufacturers and their representatives are world-wide present

- **spare parts:** spare parts are readily available since major pump manufacturers offer after- sales services almost throughout the world

- **maintenance:** no special equipment and skills required

2.3.3 Disadvantages

- **no hydraulic control device:** therefore, a control valve must be incorporated in the penstock line (additional costs) to start and stop the PAT. If the valve is used to accommodate to seasonal variations of flow, the hydraulic losses of the installation will increase sharply

- **peak efficiency :** efficiencies of PATs are inferior to sophisticated turbines of the medium to high output range which reach over 90 %. PATs reach efficiencies comparable if not superior to locally manufactured Cross Flow or Pelton turbines.

- **<u>lower efficiency at part load:</u>** a conventional turbine has an effective hydraulic control (adjustable guide vanes, nozzles or runner blades) to adjust the machine to the available flow or the required output. If PATs are operated at other than the design flow, i.e. below their best efficiency point (bep), a relatively rapid drop of efficiency will occur (in addition to the hydraulic losses incurred by the flow regulation - see first point above).

The disadvantages of PATs can be reduced to a minimum if the PAT is very carefully selected and only applied where justified. Poor performance due to an inappropriately selected machine or application will lead to a reduction of annual returns generated by the MHP. Summed up over the entire lifetime of the machine, this reduced output might by far offset the cost advantage of the PAT (lower investment costs) in comparison to a conventional turbine.

2.4 Application Particulars of PATs

2.4.1 Size of the Machine

The advantages of PATs over conventional turbines may be significant for micro hydro up to 500 kW. For larger outputs, pumps are no longer standard, mass-produced units but are purpose-built machines like conventional turbines. Thus, the cost advantage of PATs is reduced. Furthermore, it might be worthwhile in larger hydropower installations to spend more on a purpose-built, more efficient turbine since the bigger the installation the smaller the share of the hydraulic equipment as compared to total investment costs. Figure 2.4 below shows this tendency (note, the percentages indicated are only samples and might vary considerably from plant to plant).

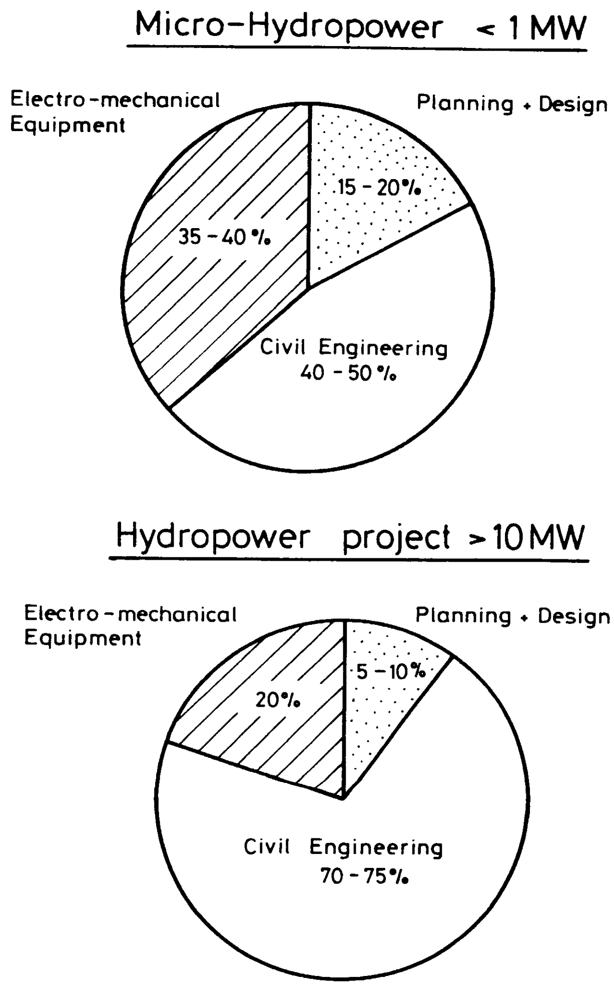


FIGURE 2.4 :
Distribution of investment costs in hydropower installations of different size (samples only)

2.4.2 Pump Design

The many diverse applications of pumps has resulted in a large variety of different pump designs. For relatively clean water as is usually available at MHPs, the classic spiral casing design has proven to be an efficient and reliable machine. It is this type of pump, available throughout the world, which is also most favoured for an application as a PAT. The simple and sturdy construction and also relatively low price (thanks to standardization) makes this type of pump fit perfectly well into the low-cost philosophy of PATs. The following figure shows a single-stage spiral casing pump of the classic design.

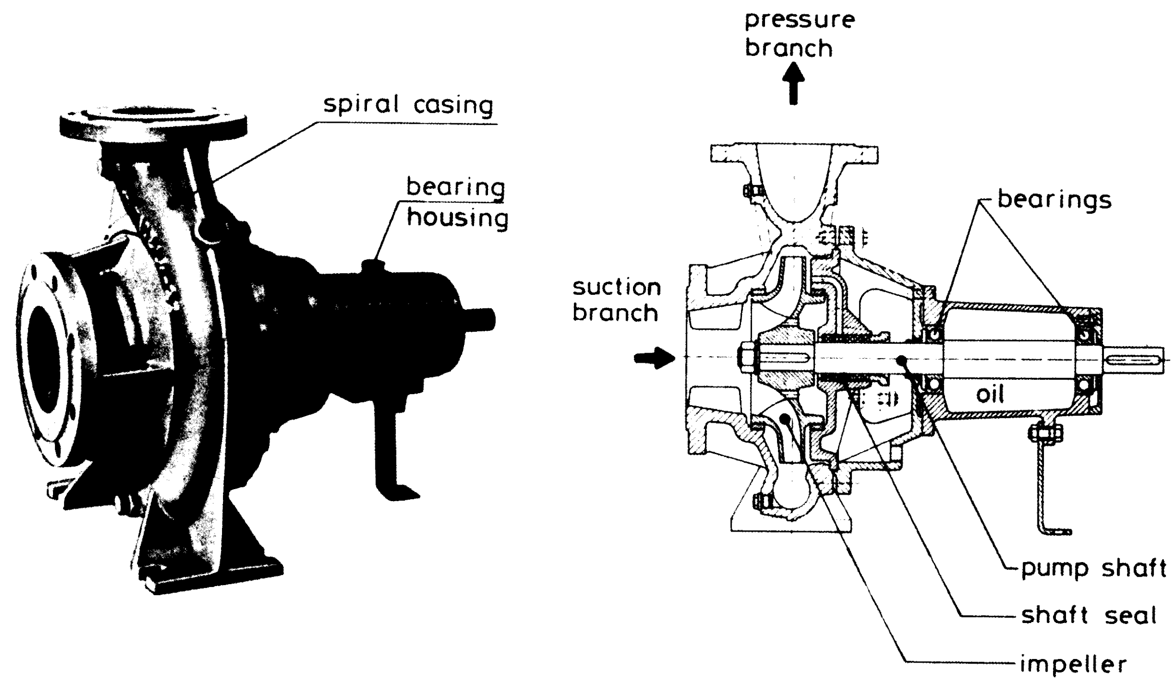


FIGURE 2.5 :
Single-stage spiral casing pump/PAT of the classic design

The basic classification of pumps may be given according to their flow pattern. For rotodynamic machines, the pump industry uses the concept of specific speed to describe the type of pump in more general terms (see also Appendix B).

$$\boxed{n_q = n\frac{\sqrt{Q}}{H^{3/4}}}$$

where n = proposed pump speed (RPM)
Q = pump discharge (m3/s)
H = pump head increase per stage (m)

The following main pump designs can be distinguished:

<u>radial flow (centrifugal) pumps</u> - low specific speed n_q = 10 to 50

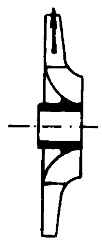
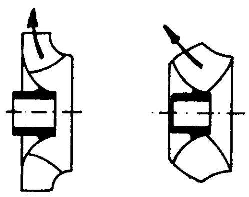
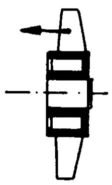
<u>mixed flow pumps</u> - medium specific speed n_q = 50 to 150

<u>axial flow pumps</u> - high specific speed n_q = 135 to 320

As a general rule, axial flow pumps are usually selected for pumping large volumes of water against relatively low heads (1 to 15 m) whereas mixed flow pumps are used for intermediate lifts (6 to 30 m) over a wide range of flow rates.

Centrifugal pumps can cover the range from low to high lifts but with only low to moderate capacity. Their primary advantage is their suction capability ; i. e. the pump can be placed above the water source and the impeller need not be submerged to pump the fluid .

Table 2.6 Overview of the basic impeller pump designs

specific speed	*impeller*	*maximum head (single stage)*	*remarks*
radial flow nq = 10 - 50 (low specific speed)		about 200 m	pump impeller designed for high pressure but relatively small flows
mixed flow nq = 50 - 150 (medium specific speed)		about 36 m	medium flow at medium head
axial flow nq = 135 - 320 (high specific speed)		about 15 m	high flows at low head

The application range of these pumps can be widened by using multistaged pumps, i.e. the impellers are placed one after the other (in series) in a single unit. Each impeller picks up the flow from the previous one and boosts up the pressure thus making it possible to lift the water to higher elevations.
In a double (or multi-) flow pump (also called double entry pump), the impellers are arranged in parallel thus doubling the capacity of the pump unit while maintaining the same head as the individual impeller (single flow pump).
More details on existing pump designs including submersible pumps are presented in Appendix C.

2.4.3 Range of Application of Turbines and PATs

Figure 2. 7 and 2. 8 represent the general field of application of different types of turbines and PATs available for micro-hydropower schemes .

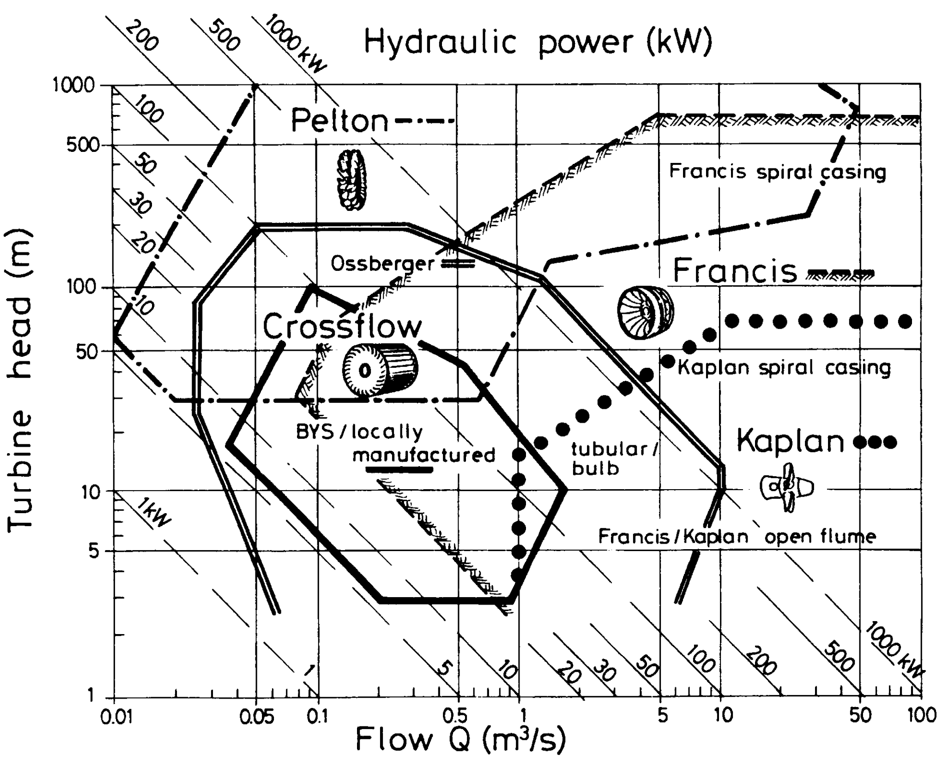


FIGURE 2.7 :
General range of application of different turbine types

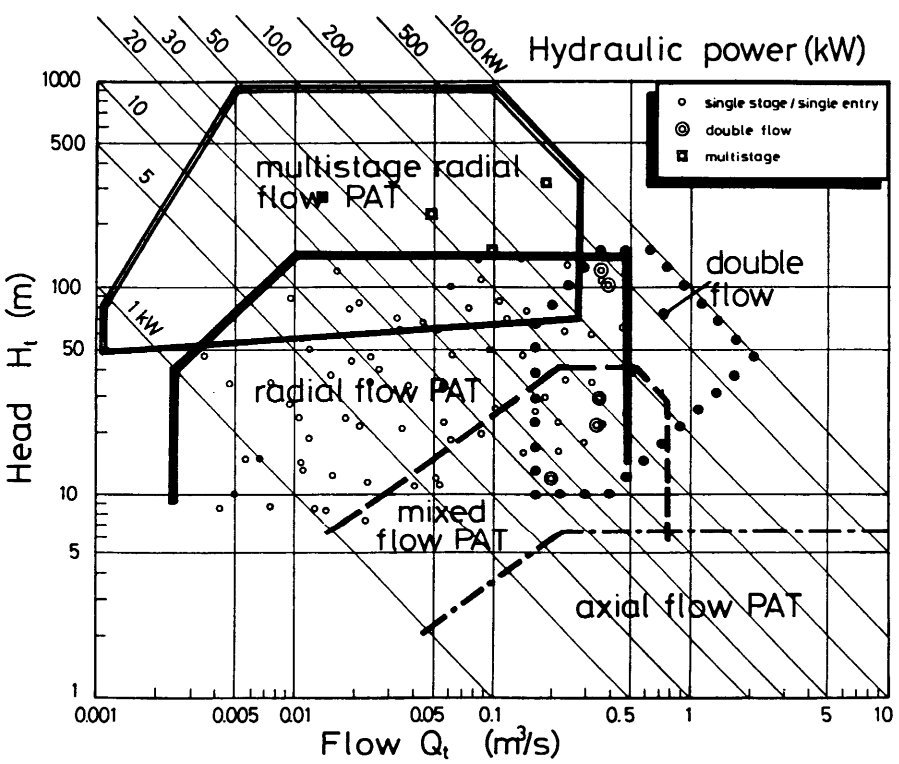


FIGURE 2.8 :
General range of application of different PAT types

2.5 Conditions and limits of the application of PATs

- Efficient operation of PATs require **constancy of both flow and load conditions** (see section 2.3) due to the lack of hydraulic control devices of PATs

- Over a limited range, flow may be (hand-)**regulated by a control valve**; this is a frequently adopted solution for direct drive of machinery or grid-linked electricity generation where speed variations are usually not a problem. However, this method is inefficient since the valve reduces not only flow, it also dissipates a considerable amount of pressure energy. The power output will therefore drop sharply .

- Variations of flow can also be accommodated by using several **PATs in parallel** and switching them on and off according to the available flow. However, the cost advantage of such an installation compared to a conventional turbine with flow regulation will be reduced.

- Although load might be constant during normal operation, it can happen that direct-driven machinery or grid-linked generators become unloaded. The equipment (PAT and machinery) will rapidly accelerate to **runaway speed**. Therefore, it must be ensured that all machines can stand the specific runaway speed of the installation. Overspeed protection (overspeed trip and automatic shut-down devices) could be considered but is seldom justified in an MHP installation for reasons of costs.

- For stand-alone electricity generation where the load on the generator usually varies considerably over the day, PATs can also be used but preferably in conjunction with **electronic load controllers** which keep the load on the PAT constant by switching in ballast loads whenever the electricity demand drops. Dissipating generated energy in the ballast load instead of reducing flow at times of low power demand is only an acceptable solution if maximum water saving is not needed (always sufficient streamflow; no storage basin available).

3. SYSTEM DESIGN

3.1 Introduction

This chapter is concerned with the design of an MHP using a pump as turbine (PAT). Basically, the design procedure is similar to that of a conventional turbine. But given the conditions and limits of a PAT as outlined in the previous chapter, a slightly different approach will be required for PATs.
An overview of possible micro-hydro applications (direct drive of machinery, electricity generation and combinations) in conjunction with power demand and available flow leads to the determination of the design flow of the PAT.
The selection of a pump yielding the desired flow and head in the turbine mode is the main issue of this chapter and probably of the manual as a whole.
Finally, the last two sections of this chapter deal with design particulars such as runaway speed, cavitation and draft tube design, and the necessary modifications and reviews of design due to the reverse operation of a PAT.

3.2 Layout of an MHP using a PAT

3.2.1 System Components of an MHP using a PAT

The main elements of an MHP are the following (see Figure 3.1):

- the **intake** where water of a river or a spring is diverted from its natural course; the intake may also draw water from a lake or a storage pond/basin. In some cases the intake contains control or regulating devices or structures such as dams or gates
- in the **settling basin or sand trap**, sediments (sand and/or silt) are separated from the water. If not eliminated, sediments can accumulate in the water course of the MHP reducing flow or the particles might cause rapid wear on the turbine
- the **head race channel or conduit** leads the water to the head of the penstock
- the **forebay** or surge tank is a chamber with a free water surface at the head of the penstock. It serves several functions:
    - settling of particles which may have passed through the sand trap
    - catching floating debris by means of a trashrack
    - protecting head race channel or conduit from excessive changes in water level caused by flow variations in the turbine or PAT
- the **penstock** is a closed conduit supplying water under pressure to the turbine or PAT
- the **turbine or PAT** converts the fluid energy into mechanical energy on the shaft which can be used in various ways (see section 3.2.2 below)

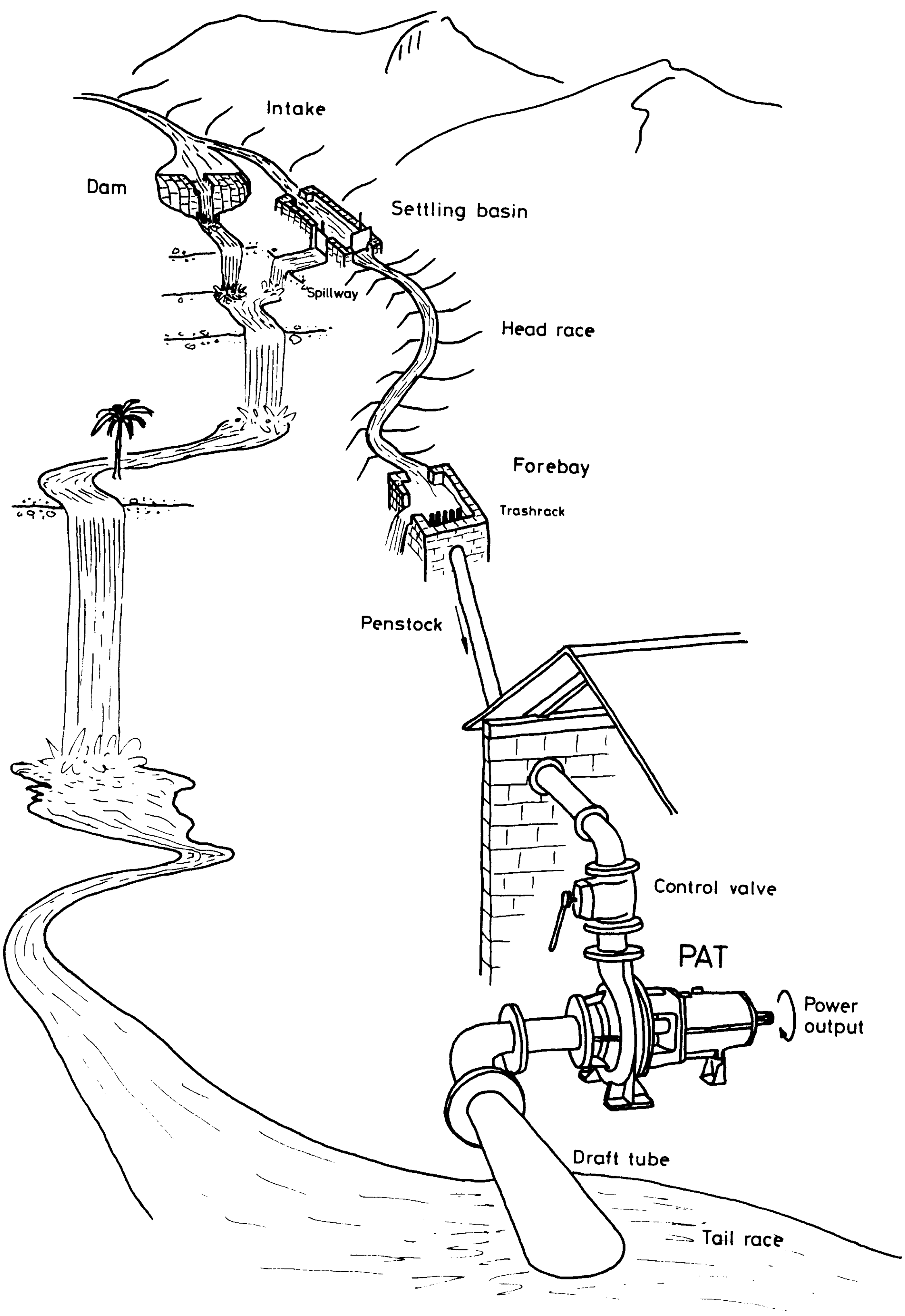


FIGURE 3.1 :
Main elements of an MHP using a PAT

3.2.2 Overview of possible micro hydropower applications

The use of a PAT in an MHP scheme does not restrict the type of hydropower consumers. Direct drive of machinery, electricity generation (in parallel to a large grid or isolated) or combinations of these are possible just as with a conventional turbine. The only difference is that a PAT cannot make use of the available water as efficiently as a turbine due to its lack of hydraulic controls. The following figure gives an overview of the various applications of hydro-power dealt with in this manual.

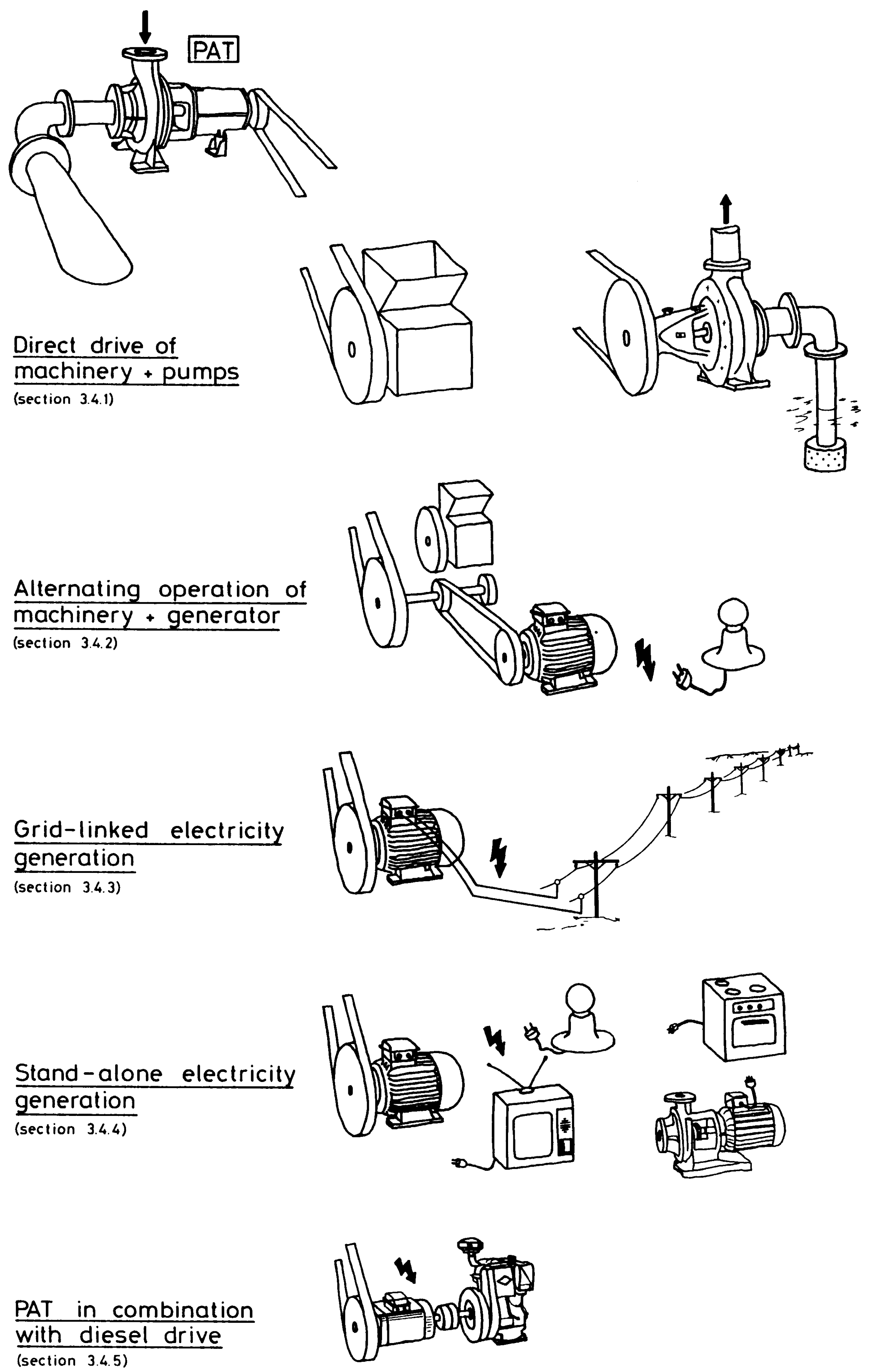


FIGURE 3.2 :
Overview of possible hydropower applications driven by a PAT

3.3 Design Parameters

3.3.1 Flow

As was described earlier, a PAT cannot operate efficiently at flows below design discharge. It is therefore very important to know actual streamflows over the year which will enable the designer to determine the nominal flow of the PAT.

Unfortunately, streamflow data over several years are seldom available for MHP schemes and there might not be sufficient time to carry out a complete measuring programme before the implementation of the scheme; however, a few measurements of flow (especially during the driest season) must be done in any case. Appropriate flow measuring methods are for example the weir (see Appendix H) or the salt gulp method. These measurements in combination with estimates of mean and minimum flows using rainfall data and catchment area can be worked out into a flow-duration curve (see for example *Allen R. Inversin:* Micro-Hydropower Sourcebook).

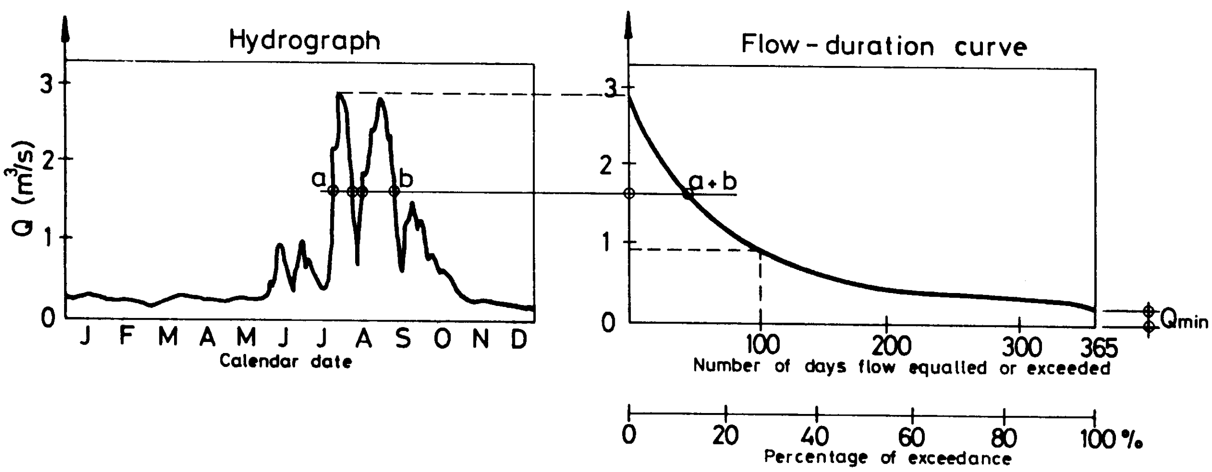


FIGURE 3.3 :
Hydrograph and corresponding flow-duration curve of a potential river for an MHP

Using the flow-duration curve, we can determine the flow which can be used and the power to be generated with a given PAT.

- If power should be available all year round, the design flow of the PAT must not exceed the minimum streamflow (this is the flow exceeded during 100 % of time = 365 days in the flow-duration curve); in practice, the annual operating time of a PAT will be less than 100 % since it may have to be stopped during maintenance and repair (after failure of components) or during exceptional floods of the river . Furthermore, a minimum flow must be left in the river to meet environmental needs (see Figure 3.4 a).

- If the power required by the consumers is higher than can be supplied by the PAT at minimum streamflow, then the layout flow of the PAT could be increased; however, PAT operation under part load and flow (during periods of streamflows inferior to the design flow) are inefficient if not impossible at all. (see Figure 3.4 b)

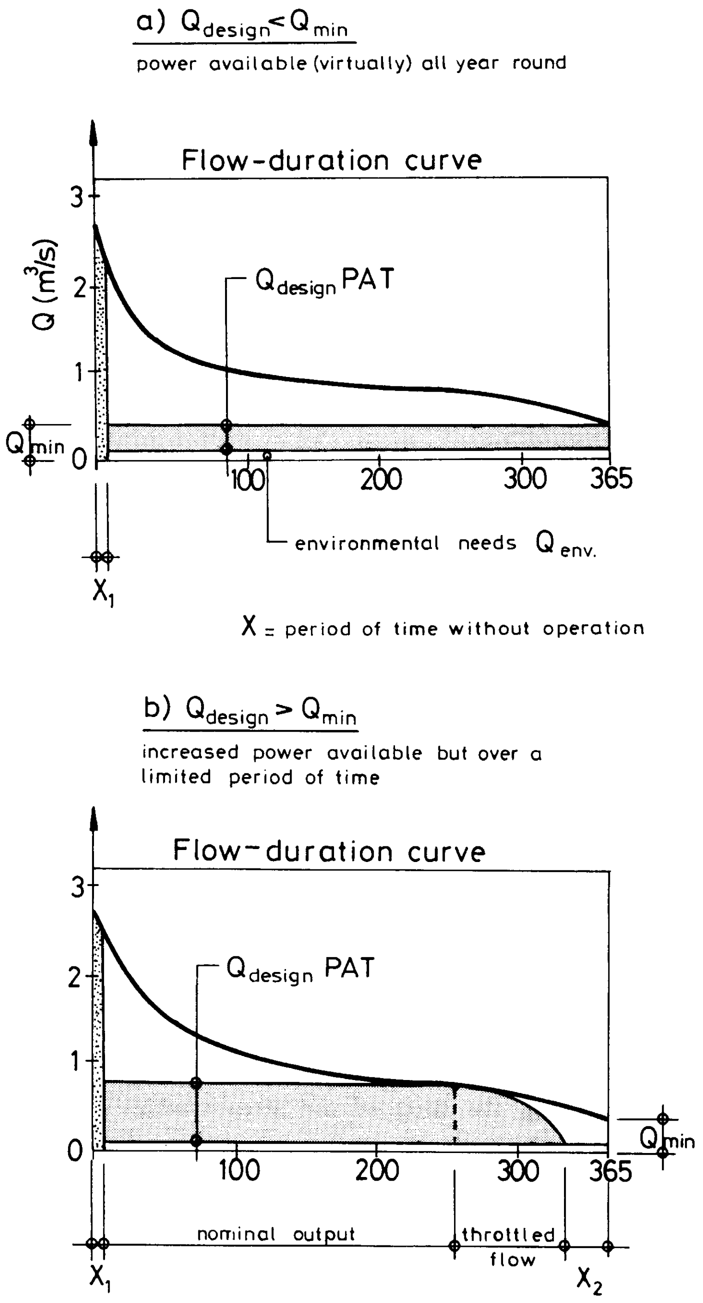


FIGURE 3.4 :
Basic options of design flows for PATs

3.3.2 Head

Head is the pressure available at the PAT expressed in meter water column. It must be distinguished between gross head, which is the difference of elevation between the water surface of the forebay and the tail race (see Figure 3.1 above) and net head, which is the actual pressure available at the turbine. To obtain net head, allowances must be made for losses in the penstock and draft tube (for details see Appendix A).

Gross head can be determined by a topographical survey using levels and tape measures. If no suitable maps of the area are available from a land surveying office, the designer should carry out a site survey yielding maps with contour lines and profiles similar to the ones shown in Figure 3.5. Of special importance are the flood or high water marks of the stream, i.e. the highest water level reached by the stream during flood. These marks will influence the layout of the intake and will determine the lowest possible location of the power house. On the basis of these maps, the basic layout of the MHP can be developed containing penstock alignment , location of power-house, intake and sand trap.

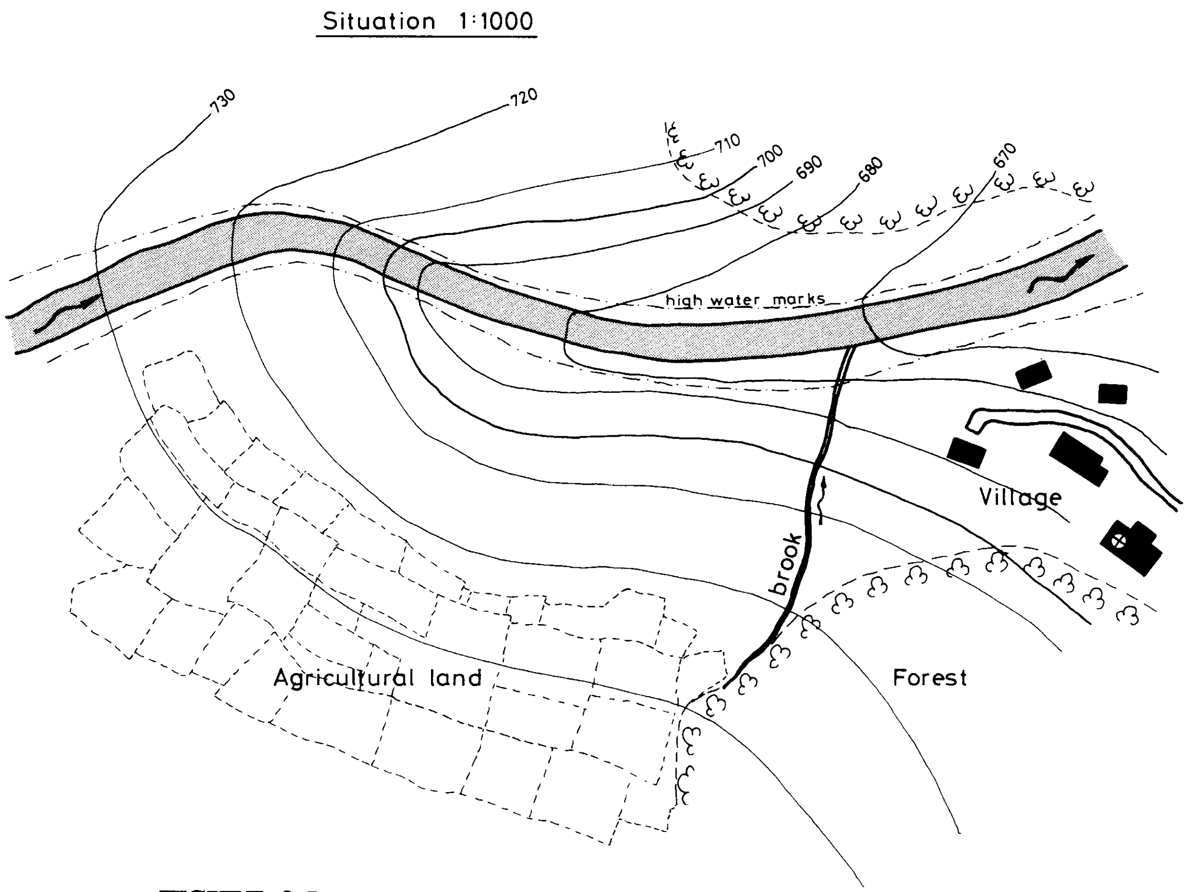


FIGURE 3.5 :
Contour map showing the drop of the stream which could be exploited for hydropower generation

3.3.3 Load

Load is defined as the power demand of the consumers connected to the PAT. These can be direct-driven machines or a generator or a combination of both. Predicting the load of MHP plants is especially important for isolated operation.
The maximum load of **direct-driven machines and pumps** can be calculated by adding up the nominal power of all those machines which will be operated simultaneously.
Predicting the **load of a generator** is more complicated since appliances are numerous and their time of operation cannot normally be influenced by the operator of the PAT. The designer of an MHP must therefore develop a load curve which shows the total power requirements over the day or longer (seasonal, yearly).

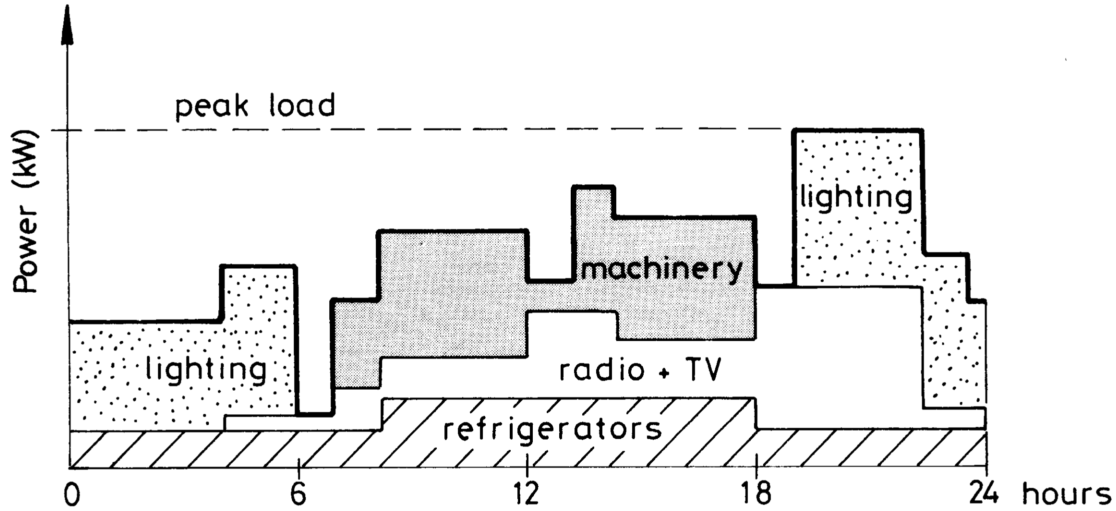


FIGURE 3.6 :
Daily load curve of a planned MHP

An increase of power demand during the lifetime of the plant can usually be expected. If the available flow is sufficient, the load curve should be extrapolated to account for additional consumers joining the scheme in later years, i.e., nominal power output of the PAT will be higher than maximum demand during the first years of operation. However, designer and plant managers should adopt a clear policy regarding the number of consumers allowed to join the MHP scheme since overloading a PAT usually ends up in a breakdown of the whole isolated network.

3.4 Matching Hydropower Potential and Power Demand - Design Flow for Various Hydropower Applications

3.4.1 Direct Drive of Machinery

The use of PATs to drive exclusively machinery of agro-processing factories (flour mills, oil expellers, rice hullers) and small industries (saw mills, paper making, wood and metal workshops) can be a very economic solution. Flat or V-belts are used to adapt PAT output speed to the speed requirements of the machines. The load of such machinery depends on its operation. If the machine is not under load, the installation will accelerate to runaway speed. Therefore, the structural integrity of PAT, gearing and machinery must also be ensured for runaway speed (see also section 3.6.1).

The design flow of the PAT is determined by the power requirement of the driven machine (see Figure 3.7a) . If the streamflow drops below these requirements during the dry season, a storage pond, collecting water during the night , could provide the necessary buffer to operate the PAT at full power output during the working hours (see Figure 3.7b).

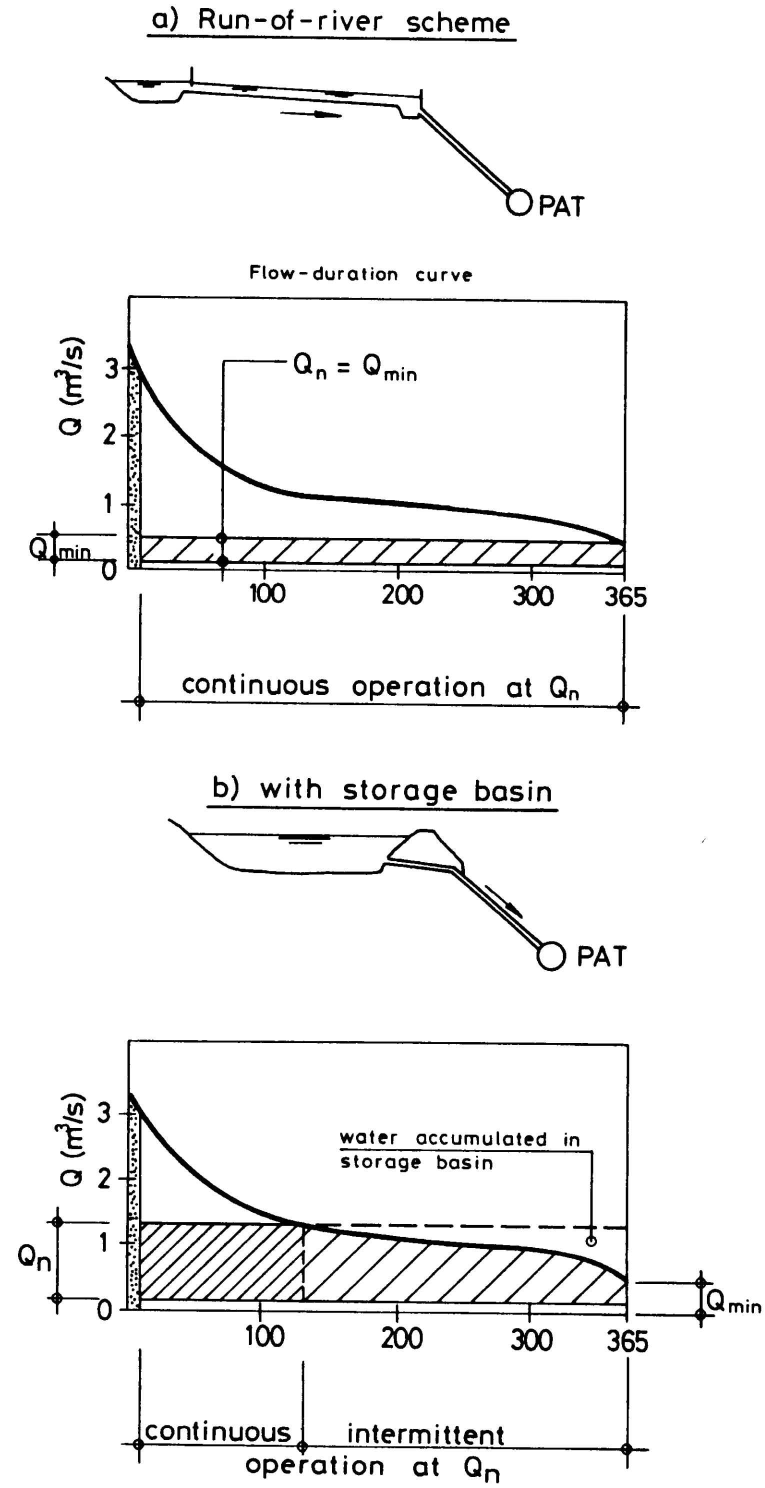


FIGURE 3.7 :
Direct drive of machinery using a PAT

3.4.2 Alternating Operation of Machinery and Generator

The machinery (workshops, agro-processing, etc.)mentioned under 3.4.1 are usually not operated during the night. The available flow could therefore be used to generate electricity for illumination, cooking, radio or TV sets during the evening hours.

The PAT operates at design flow and full power output just as when driving the machinery during the day. Nominal power of the generator must therefore correspond to maximum PAT output to avoid overspeed (and overvoltage). Regulating PAT output by closing the control valve and installing a smaller-rated

generator is not recommended. The control valve could accidently be left fully open in generator operation exposing the generator to full PAT output which could cause damage to the electrical equipment.
Since the power demand does not normally meet the output of the generator, a load controller should be installed. As the name implies, a load controller regulates the generator output by switching in ballast loads whenever the electricity demand of the appliances drops. It does, however, not regulate PAT and turbine flow.

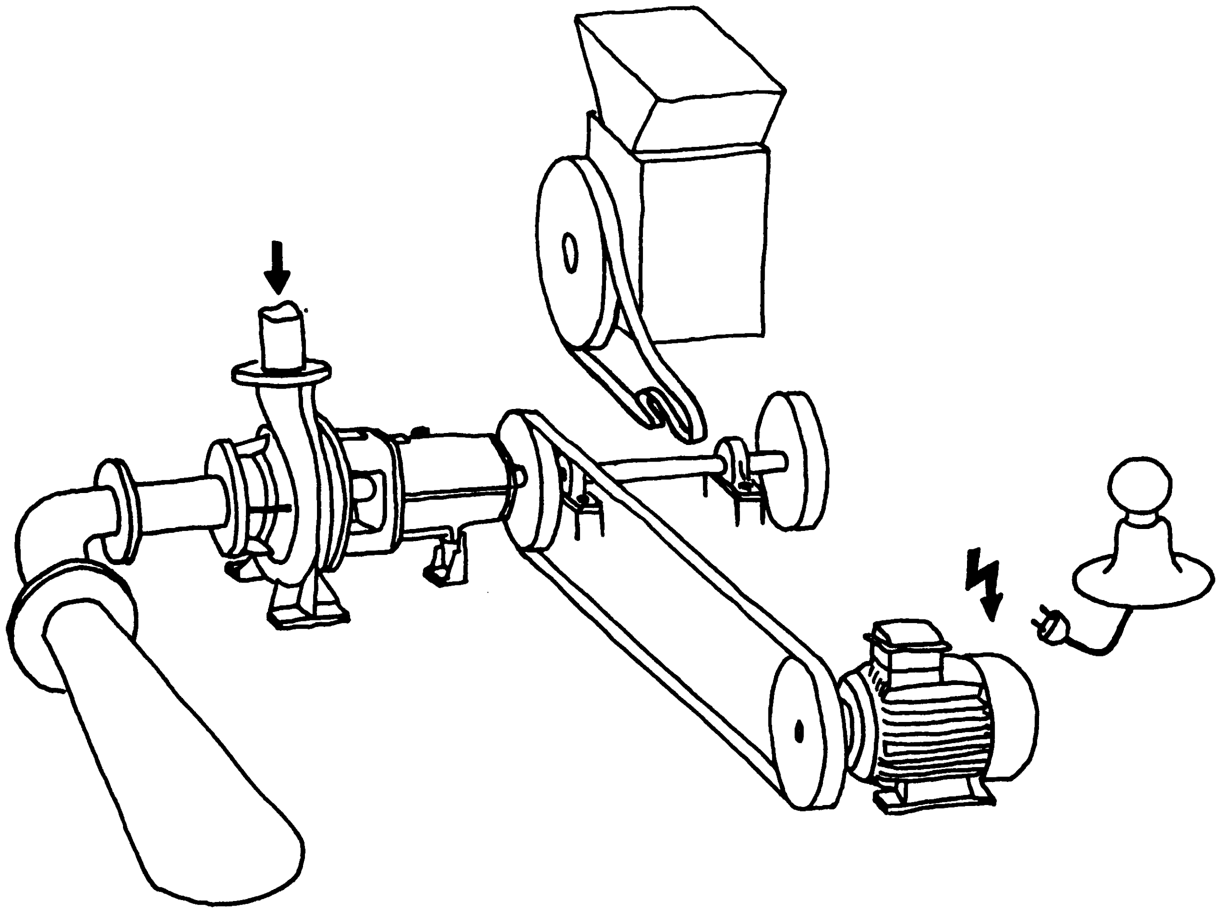
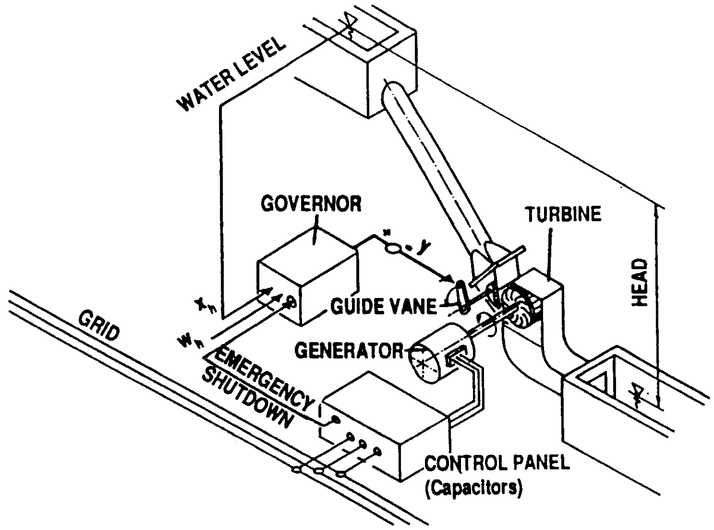


FIGURE 3.8 :
Alternating operation of machinery and a generator for domestic electricity supply

3.4.3 Electricity Generation in parallel to a national or regional grid

Electricity generation in parallel to a large grid eliminates the need for load control. Full generator output can be supplied to the grid at all times using the grid as a virtual storage; i.e. any over-production is delivered to the grid, while in case of peak demand of the local consumers, the grid may feed electricity back to support the generator of the MHP.
Such a system can make full use of the hydropower potential provided that the turbine can be adapted to variations of streamflow. This can best be achieved by a conventional turbine with a flow control governor acting on the turbine guide vane or nozzle. Turbine opening is continuously adjusted to maintain a constant water level in the forebay, hence all available water (except during flood) could be used for energy production.
It has been found that the most economic layout point of the turbine is somewhere at a streamflow which is exceeded during 100 days per year (100-day-rule).



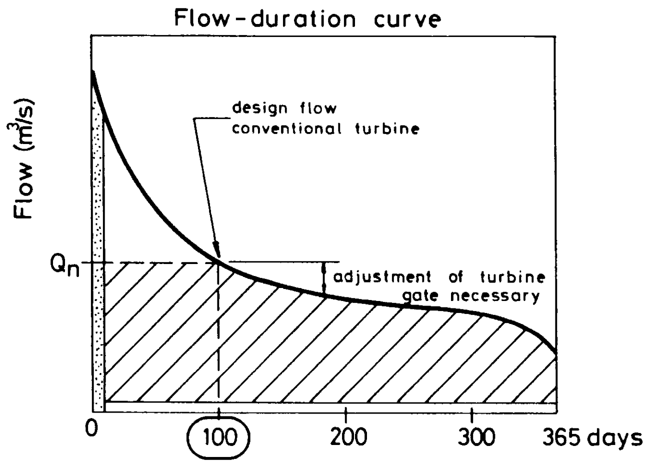


FIGURE 3.9 :
Maximum electricity generation of a grid-linked MHP with a conventional turbine and water level control

Using a PAT for grid-linked electricity generation might not look as favourable since it cannot be adjusted efficiently to varying flow conditions. In some cases, it can nevertheless be a technically and economically feasible solution for electricity generation in parallel to a large grid:

a) MHP sites with **fairly constant streamflows** during most time of the year (as frequently encountered in tropical countries with no distinct seasons) a PAT using a nominal flow equal to minimum flow is a simple but still relatively efficient solution. The additional energy production of a flow regulated turbine is small and might not make up for the lower investment costs of the PAT (see Figure 3.10a).

b) In conjunction with a **storage basin**, the PAT can yield an equally high energy production as a flow regulated turbine. The intermittent operation of the PAT during streamflows below PAT nominal flow does not affect the local consumers since they can also be supplied by the national/regional grid (see Figure 3.10 b). However, storage basins require in many cases relatively large (and expensive) civil works which is not necessarily in accordance with the cost-reduction approach usually adopted when considering a PAT.

c) Alternatively, if a storage pond is not possible, flow control on a PAT could be accomplished using the **control valve as a throttling device**. This can be done automatically by a governor sensing the water level in the forebay and a servo motor acting on the valve, or by manual adjustment of the valve according to the river discharge. As discussed earlier, the control valve is not an efficient flow regulation device and the PAT might not deliver power below 40 % to 60 % of nominal flow (see Figure 3.10 c).

d) Seasonal variations of flow can be accommodated by having **several PATs in parallel** and switching them on and off according to the available streamflow. Selecting different sizes of PATs can provide a

variety of turbine flows, thus using the available streamflow more efficiently. However, the low-cost advantage of the PAT compared with a flow regulated turbine is severely reduced (Fig. 3.10 d).
The use of PATs in parallel has advantages from the point of view of O&M. Firstly, it allows maintenance to be carried out on one unit without shutting down the complete unit. Secondly, each unit will run for less time and will therefore wear out less quickly.
Care must be taken when selecting the PATs and computing their output for a given site. When two PATs are run simultaneously, the effect of the drop in head due to the increased penstock losses for combined flow may result in the smaller PAT producing very little additional power. As a rule of thumb, it is not worthwhile to run two PATs of a ratio of rated flows of 2 : 1 simultaneously, if the penstock head loss for the smaller PAT exceeds 2% of the available head.

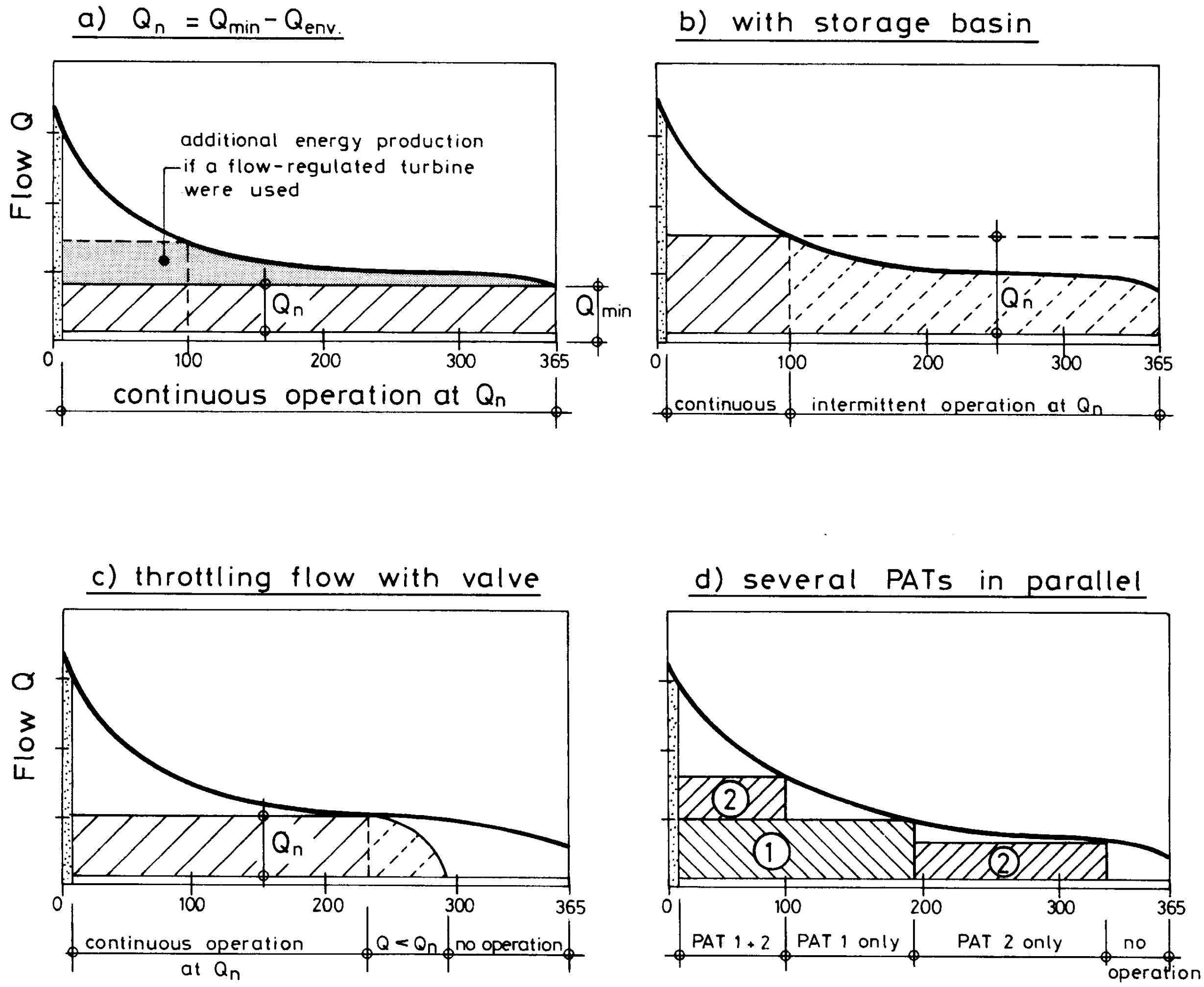


FIGURE 3.10 :
Grid-linked electricity generation using PATs of different design flows and arrangements

3.4.4 Stand-alone Electricity Generation using a PAT

The plant is the only energy producer in an isolated grid. To avoid system break-downs, the generator must at any instant cover the power demand of the consumers. Therefore, the main design parameter for stand-alone systems is **peak load** rather than efficient use of the hydropower potential as in parallel operation.
The design flow of an MHP therefore corresponds to peak demand (+ distribution losses + safety margin) of the consumers.

a) Sufficient streamflow throughout the year

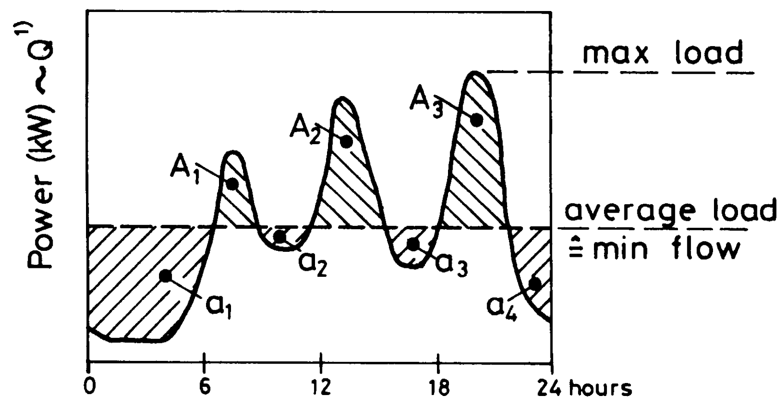
As long as the available streamflow always exceeds the MHP design flow, a straightforward design as explained under 3.4.2 above can be adopted and the use of a PAT instead of a conventional turbine is suitable in most cases. PAT and generator may be operated at constant nominal output, thus requiring a load controller to account for variations of power demand (see Figure 3.12, case a).

b) Insufficient Streamflow

If streamflow falls short of the demand, measures to avoid or reduce a restricted use of energy during peak hours must be taken. These can be:

- Storage ponds or basins which store water during periods of low power demand and which supply the MHP at periods of peak demand

  Unrestricted use of energy is possible, if the minimum streamflow is at least equal to the water demand at average load. The required size of the storage pond can then be computed from the load curve as shown in Figure 3.11. Note that the storage basins considered here are designed to balance out only daily supply and demand of power. Accommodating to seasonal variations of flow requires very large storage basins which are not regarded as relevant for micro hydropower unless a natural lake can be used.



1) if η_{total} assumed constant

FIGURE 3.11 :

Daily load curve of an MHP in isolated operation and computation of required storage volume for unrestricted energy supply during the dry season (valid for turbines with flow control only)

Estimating the minimum storage volume required:

A, a = areas on load curve = power demand * time = energy demand (kW * h)
Principle: Energy not used during low demand must be stored in basin for peak demand
sum of surplus energy = (a1 + a2 + a3 + a4 + ...)
sum of energy shortage = (A1 + A2 + A3 + ...)

since energy supply by the MHP is $E = \Sigma(a) = \rho * g * Q * Ht * \eta_{total} * t$ **and Q*t = volume of water V, we can write:**

$V_{surplus}$ **during low demand** $= \Sigma(a) / k$ **where** $k = 3600 * \rho * g * Ht * \eta_{total}$ **and**

$V_{shortage}$ **during peak demand** $= \Sigma(A) / k$

To avoid energy supply restrictions, $V_{surplus}$ **must be at least equal to** $V_{shortage} = V_{basin}$

- The second possibility to avoid bottle-necks in power supply in stand-alone electricity generation is the use of other, non-hydropower energy sources such as diesel generating sets during the dry season (see Figure 3.12, case b)

It is obvious that a PAT lacking hydraulic control is not suitable if maximum water saving is required since operation under part load is inefficient or not possible at all. Thus, in cases where restricted energy supply is to be avoided, a conventional turbine with flow control in connection with a storage pond should be aimed at (see Figure 3.12, case c).

Depending on the level of reliability of the energy supply required in an isolated system, a PAT might still be a feasible solution:

- if restricted supply during the dry season is acceptable, two (or more) PATs might be installed. While one of them supplying a base load throughout the year (for refrigerators etc.), the other(s) would cover peak demand when streamflow is sufficient (see Figure 3.12, cases d and e)

- if complete stoppage of power supply is acceptable, a single PAT may be used either working only intermittent under nominal flow (using a storage pond as a buffer) or stopping supply completely at very low streamflows (see Figure 3.12, cases f and g).

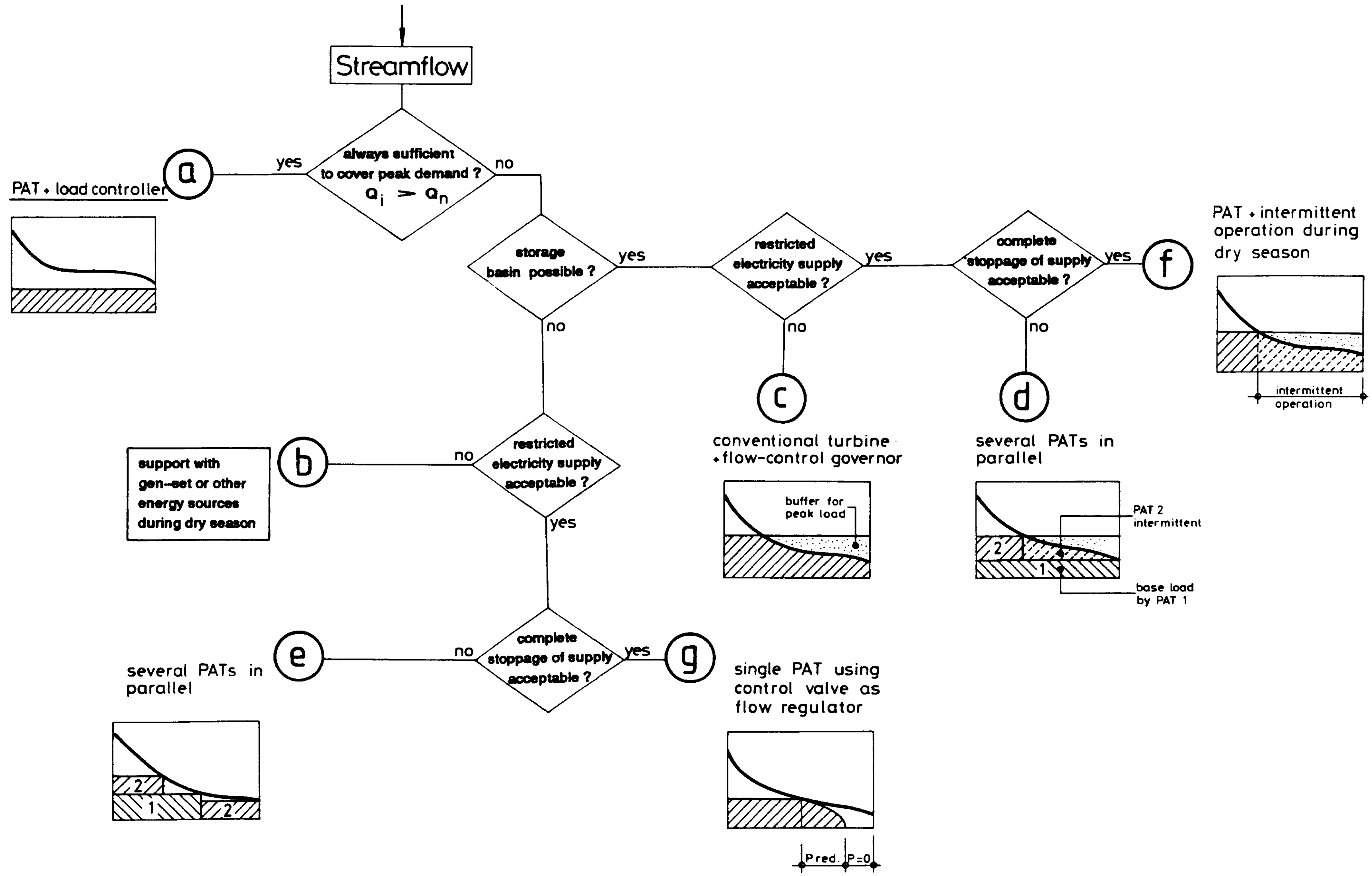
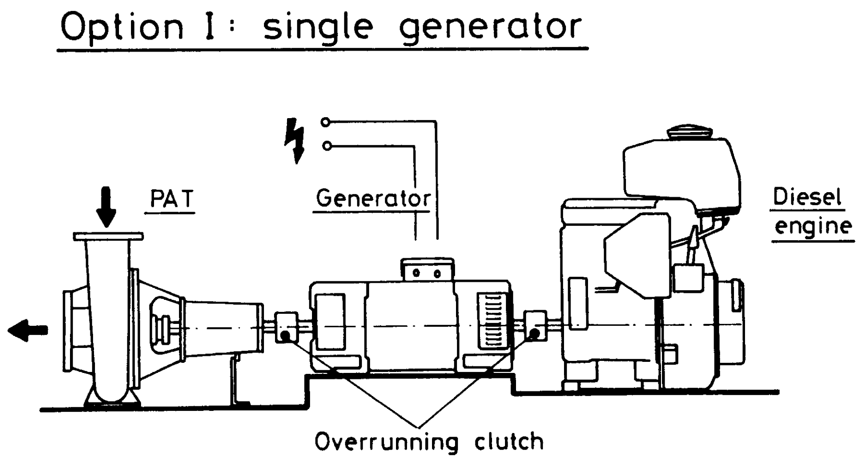


FIGURE 3.12 :
Stand-alone electricity generation and possible applications of PATs

3.4.5 Special Cases

As mentioned above, the use of stand-by generating sets in combination with a PAT in stand-alone electricity generation might be a very effective solution in many cases. The gen-set can be used to either cover peak load (operating simultaneously with the PAT) or the base load during PAT standstills due to insufficient streamflow or maintenance of the hydropower installation.
Several layouts and control systems of PAT and gen-set are possible; a simple system being the one where the two energy producers are independently switched on and off by an operator. More sophisticated systems with automatic regulation of the stand-by equipment according to power demand are however more convenient. In all cases, some form of protection (overrunning clutch, reverse power relay) is required to avoid the PAT or the gen-set to drive the other equipment as a motor or pump in case of low flow or low power demand. As in all isolated plants, a load controller would dissipate excessive power produced by the PAT running continuously at design flow.



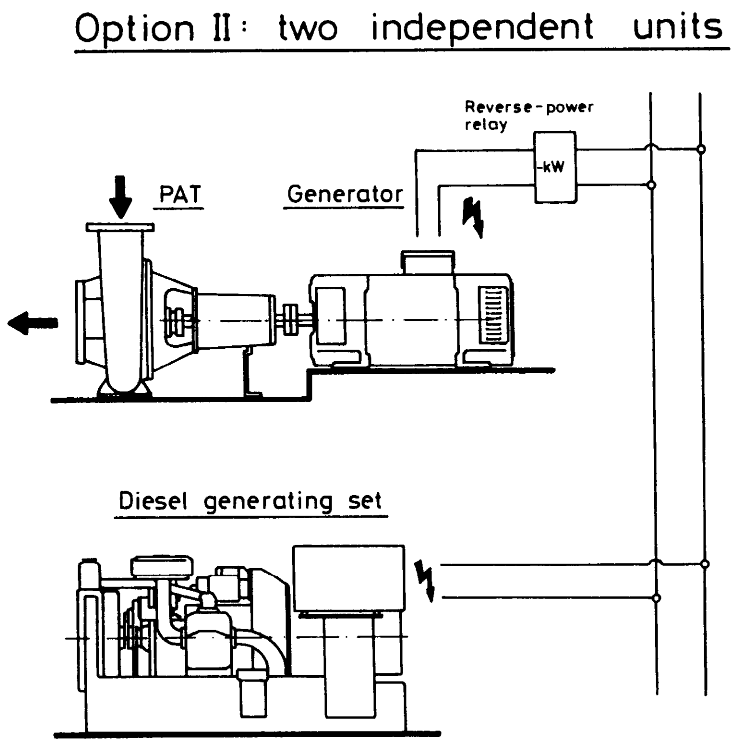
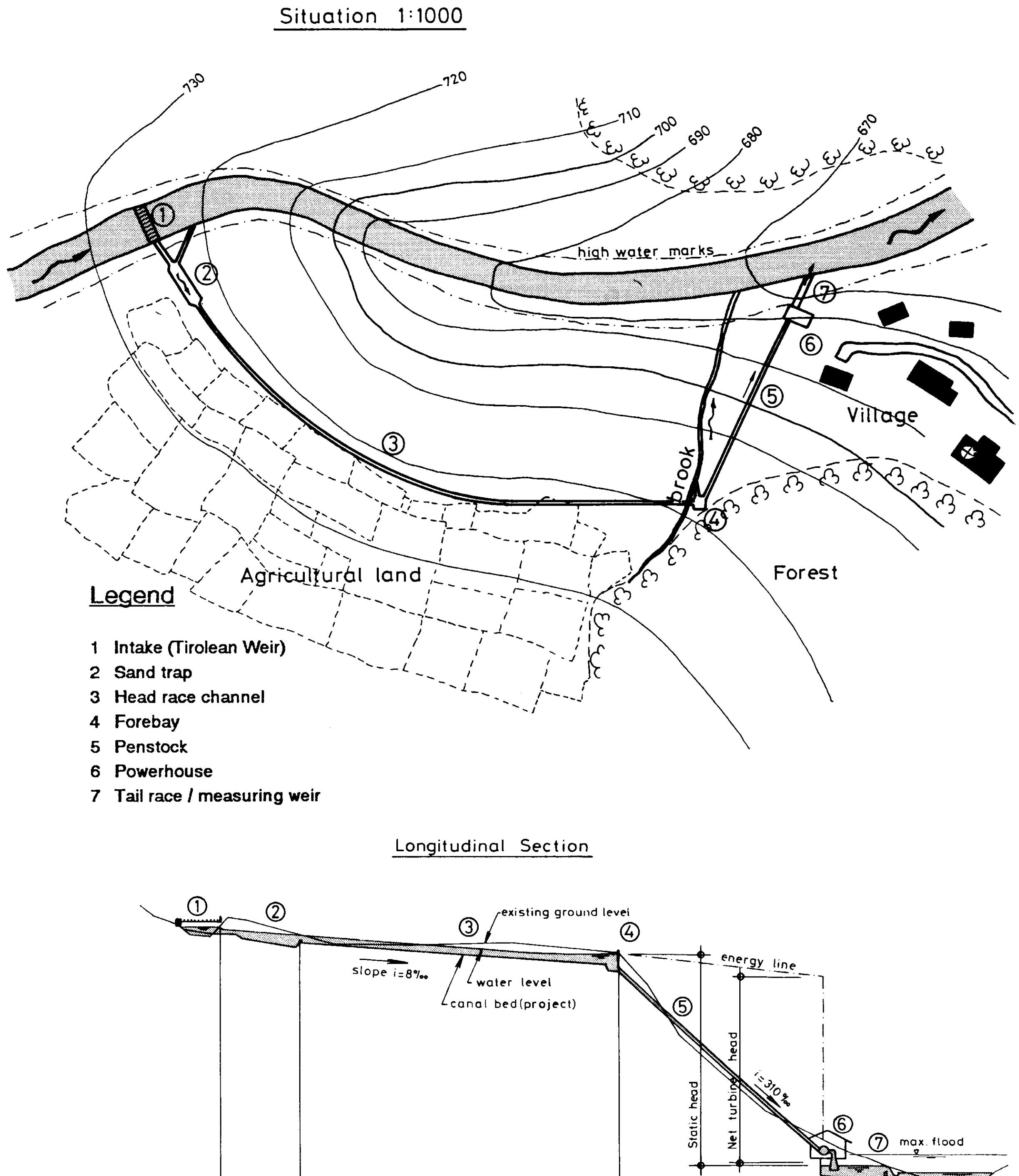


FIGURE 3.13 :
PAT in stand-alone electricity generation in combination with stand-by diesel generating set

3.4 .6 Preliminary Design - Estimate of Economic Feasibility

Having determined design flow (from the flow-duration curve) and the overall configuration of the plant (with / without storage basin, etc.) , a preliminary design can be made using the maps and profiles as outlined under 3.3.2 above. From these, the available net head of the PAT can be determined (see Figure 3.14, for penstock design and head loss calculations see Appendix A).



Horizontal distance		-40-	-425-	-255-	-28-	-35-
Existing ground level	725	720	718	678	668 668	663
Canal bed level (project)	722	718	715		666 666	
Water level (project)	723	720	718		668	

FIGURE 3.14 :
Preliminary design of an MHP using a PAT

Prior to selecting a pump operating at the design parameters (net head, nominal flow and speed) in turbine mode, a rough estimate of the economic feasibility of the MHP should be carried out especially as regards to whether the low-cost advantage of a PAT is not weighed out by the more efficient use of the available water by a conventional turbine (for details see Chapter 5).

3.5 Selection of PATs

3.5.1 State-of-the-art

Ideally, the selection of a PAT should be as easy as the selection of a standard pump: performance diagrams of different manufacturers are reviewed in search of the pump delivering the required discharge at the available head and at optimum efficiency. Unfortunately,this is not the case for PATs at the present time. Information about the turbine-mode performance of pumps are not yet sufficient.
Several approaches - empirical and theoretical methods - to predict turbine-mode performance of pumps have been proposed in recent years. The aim is to predict PAT performance from pump data which are usually readily available, namely head, flow and efficiency at the best efficiency point (bep) .
However, neither of the methods proposed so far are 100 % reliable. Errors between actual and predicted turbine-mode performance of standard pumps have been reported to reach 20 % and more.

Therefore, the design engineer of MHP schemes must be well aware that a certain degree of inaccuracy will be inherent in all his computations regardless of the method he is using. This manual summarizes the basic know-how of the design and application of MHPs using PATs. Despite the large number of PATs (> 80) analyzed over recent years, the selection diagrams presented in this manual cannot provide error-free results. A deviation between actual and predicted PAT performance will always occur but will not have detrimental effects on the proper operation of the MHP in most cases (provided that the PAT layout point was appropriately chosen).

3.5.2 The difference of performance in pump and turbine mode

3.5.2.1 Performance Curves

The performance of a pump or a turbine is usually presented in head versus flow diagrams (see Appendix B). Pump <u>and</u> turbine mode performance can be plotted in a single diagram by extending the flow axis (Q) into negative values representing the reverse operation of the pump, viz. the turbine mode performance. These diagrams are usually referred to as complete or total characteristics of a machine . Figure 3.15 shows the total head-flow characteristic of a PUMP/PAT for the same speed in pump and turbine operation.

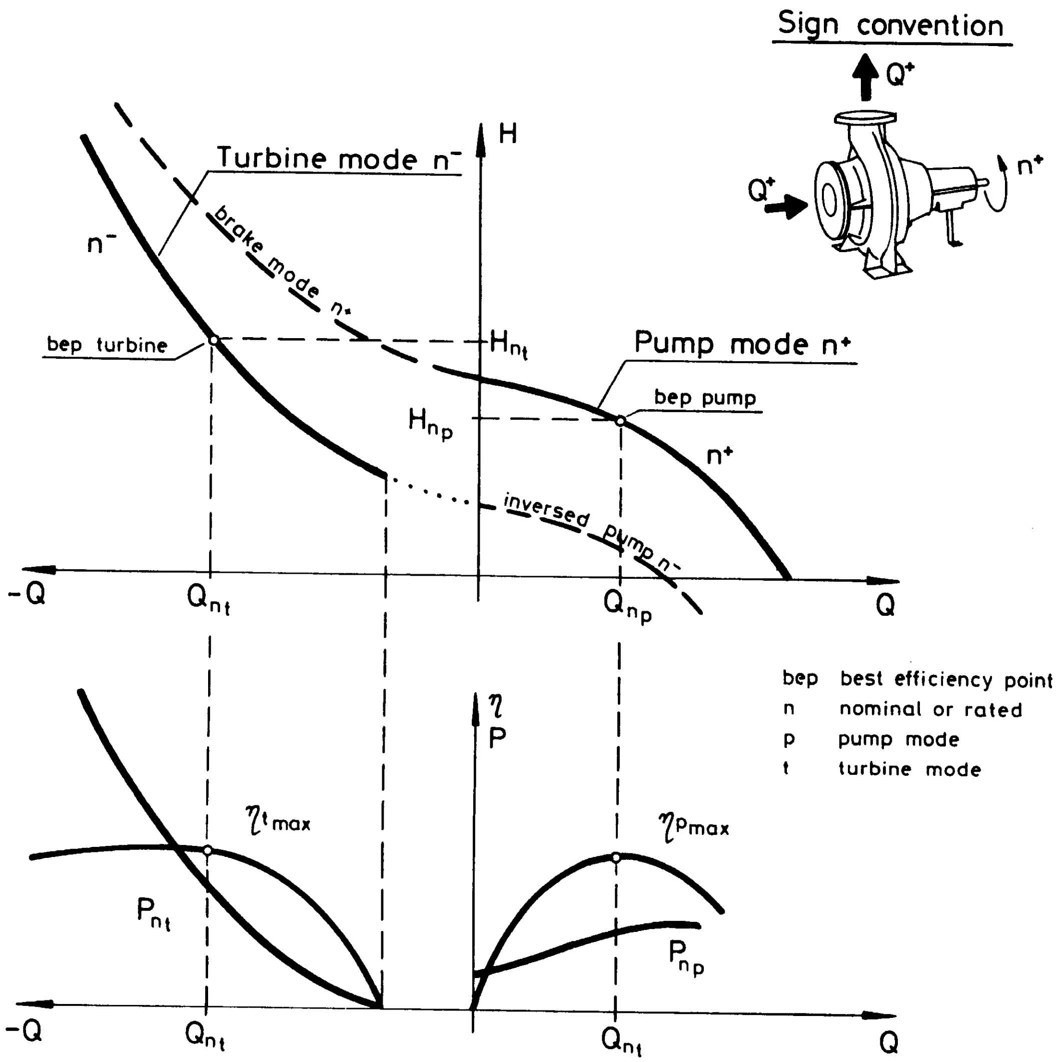


FIGURE 3.15 :
Complete characteristic of a pump for constant speed

From the complete characteristics, the following main difference between pump and turbine-mode performance can be drawn:

Parameter	*Pump mode*	*Turbine mode*
Flow	decreases with increasing head, reaches zero at the maximum pump head (shut-off head).	increases continuously with increasing head.
Power	radial flow: minimum power at maximum head. Power increases with decreasing head. At heads lower than nominal, pump may be slightly overloaded. axial flow: (not shown in Figure 3.15) maximum power at maximum head, power decreases with decreasing head (see Appendix C)	turbine starts to produce power if the flow exceeds a minimum. The nominal output increases progressively even beyond the nominal head. The power output at BEP_t is higher than for pump BEP_p (higher torque on the shaft).
Efficiency	efficiency increases from zero at no flow to a clear peak at the nominal point and decreases rapidly for higher flows	efficiency increases from zero at minimum flow, reaches a peak at nominal point and decreases slowly for higher flows

TABLE 3.16 :
Main features of pump performance in pump and turbine mode

Figure 3.15 shows clearly that the best efficiency point in the pump mode (bep_p) does not occur at the same head and flow values as the best efficiency point in turbine mode (bep_t). Obviously, the basic task when selecting a PAT is to convert the pump performance (head, flow) of a suitable machine into its turbine-mode performance. As indicated above, the relationship between the pump and turbine mode is not the same for all types and sizes of pumps but depends on the flow pattern through the machine (expressed by the specific speed) and the losses incurred which are expressed by the efficiency of the machine. The problem is, that these losses in the pump mode are not exactly the same when flow is reversed in the turbine mode. Depending on the pump design (number of impeller vanes, vane angles, volute casing or guide vanes) the relation between turbine-mode and pump-mode performance may differ considerably from one machine to another although specific speed (and pump efficiency) may be the same.

3.5.2.2 Basic Theory

A theoretical explanation why performance in turbine mode differs from the pump-mode performance of the same machine at the same speed is given hereafter.

Applying the Euler equation for ideal machines (see Appendix B), we can draw ideal performance curves (lines) in both pump and turbine mode (see Figure 3.17). For ideal conditions, the design flow and design head in both pump and turbine mode are identical. However, for <u>real</u> fluids and machines, two effects must be considered:

a) geometry of the pump;

b) hydraulic losses of the real fluid.

<u>**Geometric Effect (influence of the finite number of blades)**</u>

Optimum operation of a pump is reached at a flow which corresponds to whirlfree inlet conditions. The corresponding head is determined by the outlet velocity vector diagram with the vane angle β_2 as the decisive element in the pump mode.
The theoretical performance of a pump as described by Euler (see Appendix B) assumes an infinite number of blades. Flow through impellers of <u>real</u> pumps having a finite number of blades is subjected to a secondary flow pattern in the impeller passages known as the <u>circulation loss</u> caused by the rotation of the impeller (see Figure 3.17).
Because of this circulation loss, the impeller outlet velocity of the fluid is slightly deflected and leaves the impeller not at the blade angle $\beta_{2\,\infty}$ but at the fluid angle β_2. This produces a reduction of the generated pump head ($H_{n_p\ \mathrm{frictionless}}$ instead of $H_{pth\ \infty}$).
Note that the circulation loss is not a loss of energy as e.g. friction losses. It is only the energy transfer between fluid and impeller which is reduced but not the energy content of the fluid itself.

If this pump is operated in reverse, i.e. as a turbine, its performance is mostly determined by the inlet triangle with the governing element being the volute casing (angle α_2). The circulation loss now occurs at the <u>inner</u> periphery of the impeller which, due to the smaller diameter of the impeller at the suction side, is practically negligible. The frictionless performance in turbine mode therefore corresponds to the ideal Euler condition (infinite number of blades, no circulation loss) and <u>**both head and flow at the best efficiency point**</u> (inlet velocity triangle with α_2 and $\beta_{2\,\infty}$) <u>**will be bigger than in pump mode.**</u> We refer to this phenomenon as the geometric effect since it is basically caused by the different <u>***geometric parameters determining the energy transfer***</u> between the fluid and the impeller of a machine:
In pump mode: fluid flow leaving the impeller (decisive element: angle β_2)
In turbine mode: fluid approaching the impeller, directed by the volute casing (decisive element: angle α)

This increase of flow in turbine mode will affect the outlet triangle: conditions are no longer whirlfree since the vane angle at the suction side is not designed for this higher flow. However, it has been found in practice that a slight whirl in the suction or draft tube of a PAT (and turbines in general) is favourable especially in view of losses due to the diffusion process (deceleration of flow) at the suction side of a PAT or turbine.

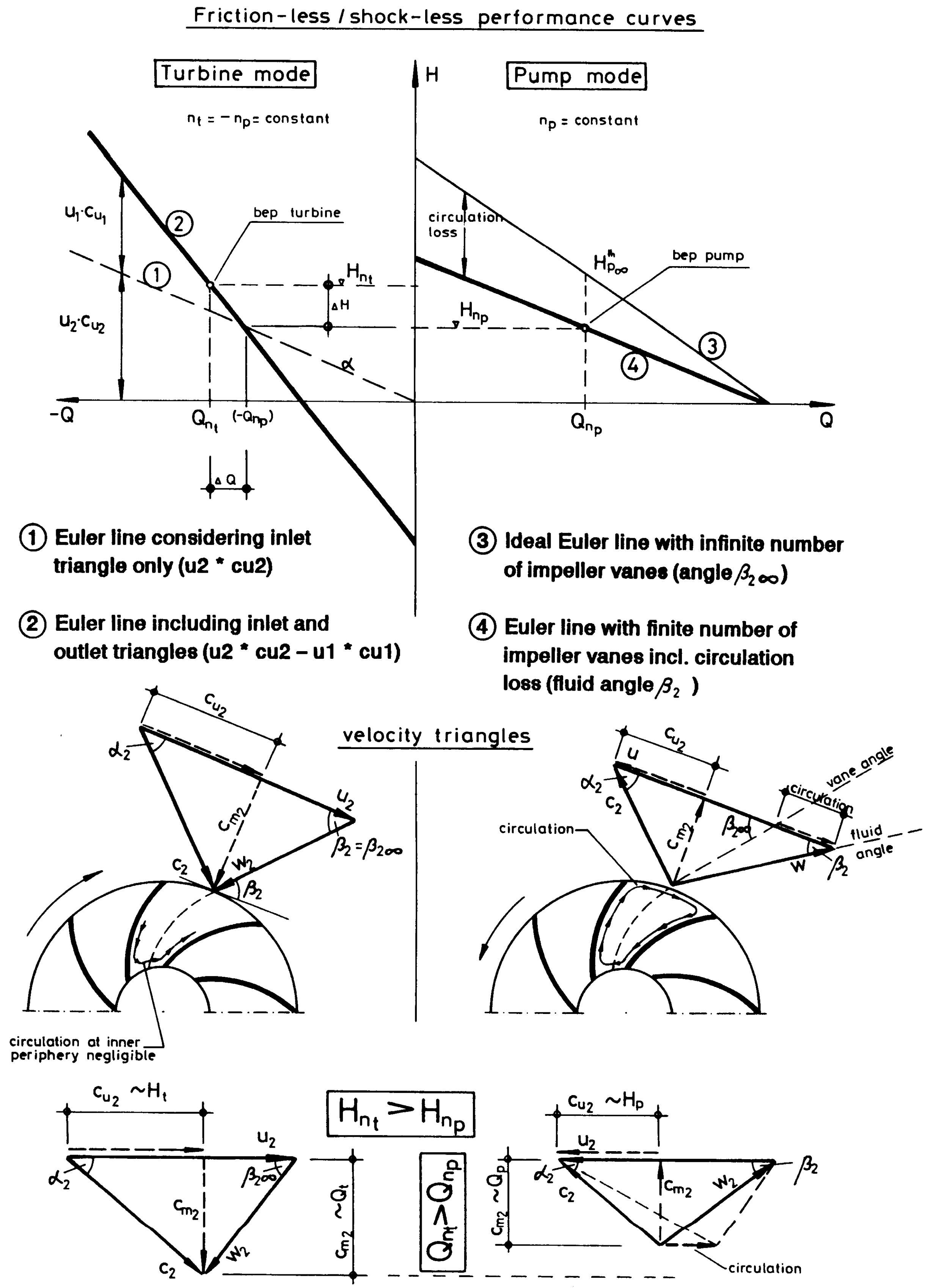


FIGURE 3.17 :
Performance differences between pump and turbine mode due to circulation loss (geometric effect)

Hydraulic Losses

As the fluid passes through the impeller, it is subjected to friction and shock losses (see Appendix B). Due to these losses, the ideal energy transfer from the rotating impeller to the fluid as expressed by the Euler equation is not achieved. Total dynamic head generated by the pump is always lower than for ideal, frictionless conditions. This reduction of head is expressed by the <u>**energetic efficiency**</u> of the pump.
In turbine mode energy transfer is in reverse. In order that the PAT operates at optimum flow conditions, an increased pressure must act on the PAT. Therefore, friction and shock losses must be <u>**added**</u> to the ideal head according to Euler (see Figure 3.18).

Additional hydraulic losses occur in a pump due to the leaking-back of fluid from the high pressure side to the low pressure side reducing the total flow pumped; these losses are called volumetric losses and the efficiency corresponding to these is the <u>**volumetric efficiency**</u> (note: total hydraulic efficiency = $\eta_{\text{energetic}} * \eta_{\text{volumetric}}$). Similar losses exist in turbine mode, i.e. small quantities of fluid by-pass the impeller of the PAT and do not contribute to the energy transfer. To maintain optimum head/flow conditions (best efficiency) the flow approaching the PAT must be <u>**increased**</u> to compensate for the by-pass flow.

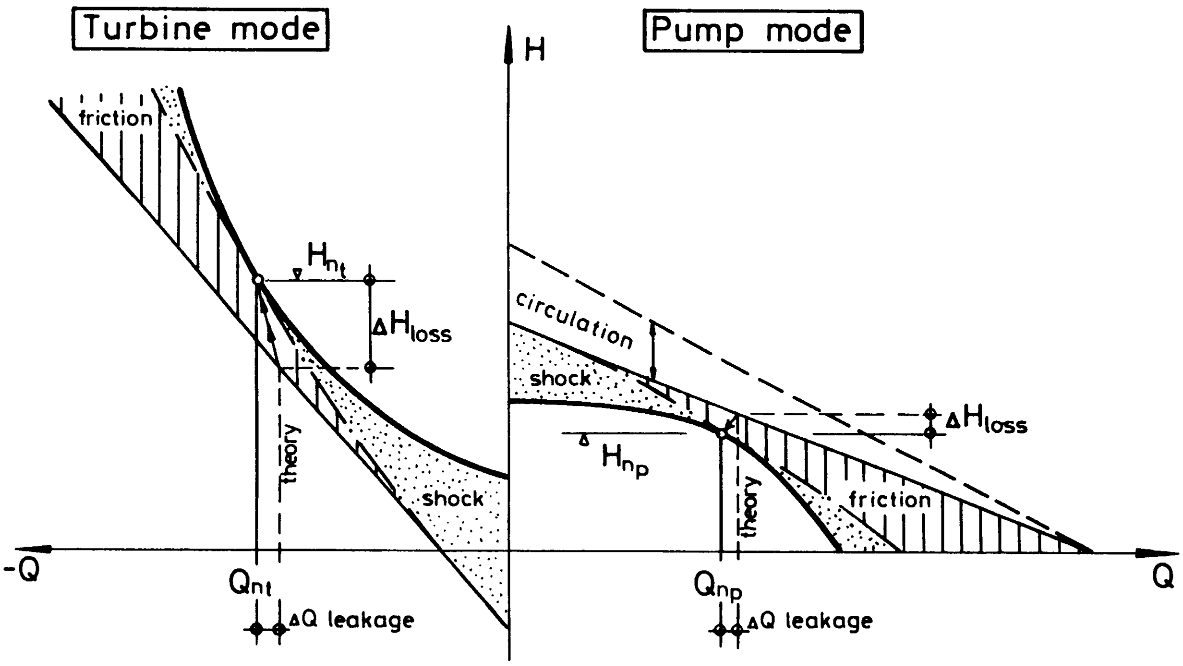


FIGURE 3.18 :

Performance differences between pump and turbine mode including hydraulic losses

In order to visualize the effect of hydraulic losses on the relation between pump and turbine mode performance, the following simple computation can be made (geometric effect = circulation loss not considered):

$$\boxed{H_p = H_E\ \eta_p} \quad \text{and} \quad \boxed{H_t = \frac{H_E}{\eta_t}} \qquad (3.1a/b)$$

Since the ideal heads (H_E) according to Euler (no circulation loss according to Figure 3.17) are the same in both pump and turbine mode, we can immediately write:

$$\boxed{\frac{H_p}{H_t} = \eta_p\ \eta_t} \qquad (3.2)$$

Experimental data have shown that efficiencies in turbine mode may reach the values of the pump mode. Assuming an efficiency of 80 % leads to:

$$\boxed{\frac{H_p}{H_t} = \eta_p^{\ 2} = 0.8^{\ 2} = 0.64} \quad \text{or} \quad \boxed{\frac{H_t}{H_p} = \frac{1}{0.64} = 1.56} \qquad (3.3)$$

(Note: the geometric effect has not been taken into account)

Conclusion: Pumps operated as turbines require a net head which is between 30 and 150 % higher than in pump mode to operate in their best efficiency point. In other words, for a given site (head/flow conditions) a smaller pump must be selected in turbine mode than for the same conditions in pump mode.

3.5.3 Review of existing approaches to predict turbine-mode performance of PATs

- Turbine-mode performance related to pump efficiency

  Based on theoretical considerations, **Stepanoff** (1957) states that the performance of a pump operated as a turbine at the same speed would take the following values:

$$\boxed{H_{nt} = \frac{H_{np}}{\eta_{np}}} \qquad (3.4)$$

$$\boxed{Q_{nt} = \frac{Q_{np}}{\sqrt{\eta_{np}}}} \qquad (3.5)$$

McClaskey and Lundqvist (1976) recommended for Q_{nt} the equation:

$$\boxed{Q_{nt} = \frac{Q_{np}}{\eta_{np}}} \qquad (3.6)$$

Comparison between actual test results and the recommendations of both Stepanoff and McClaskey show relatively large discrepancies; therefore, the use of these formulae is recommended for preliminary design or for pre-selection purposes only.

The **BUTU method**, first developed in Mexico (BUTU = PAT in Spanish) later refined in Great Britain, proposes empirical formulae which were found by curve-fitting of experimental data for pump and turbine-mode performance of standard pumps. The BUTU method predicts turbine-mode performance from pump data but not vice-versa; therefore, several pump models have to be considered in order to find the one matching best the required head-flow conditions for a given MHP.
The relations between bep p and bep t parameters are given by the following functions of pump maximum efficiency:

$$\boxed{\frac{P_{hnp}}{P_{hnt}} = 2\,\eta_p^{9.5} + 0.205} \tag{3.7}$$

where P_{hn} = hydraulic power = $\rho * g * Q_n * H_n$ at the nominal point

$$\boxed{\frac{H_{np}}{H_{nt}} = 0.85\,\eta_p^{5} + 0.385} \tag{3.8}$$

$$\boxed{\eta_{nt} = \eta_{np} - 0{,}03} \tag{3.9}$$

In a second step, the BUTU method predicts turbine mode performance of a PAT away from its best efficiency point. This is useful since, given the range of available pumps, it is unlikely that the selected PAT will operate exactly at its best efficiency point at the given site but somewhere near it. The method assumes that the shape of the performance curve is dependent on the specific speed of the machine. The formulae are the following:

$$\boxed{\frac{P_t}{P_{nt}} = (1-k)\left[\frac{Q_t}{Q_{nt}}\right]^2 + k\left[\frac{Q_t}{Q_{nt}}\right]} \tag{3.10}$$

$$\boxed{k = \frac{-1}{0.96\,(\omega_{st} - 0.2)^{-0.92} + 0.13}} \quad \text{with} \quad \boxed{\omega_{st} = \frac{2\,\pi\,n_{nt}}{60}\,\frac{\sqrt{P_{nt}/\rho}}{(g\,H_{nt})^{5/4}}}$$

$$\boxed{\frac{P_{ht}}{P_{nht}} = \frac{e^{\left(0.37\frac{P_t}{P_{nt}} - 1\right)} - 1}{0.37} + 1} \tag{3.11}$$

Note that the formulae do not reflect any theoretical considerations as for example the Stepanoff formulae but are simply the result of mathematical curve-fitting functions applied on performance curves of tested machines. Unfortunately, the use of the formulae are relatively time-consuming without a computer. Despite the "scientific" appearance of the formulae, the errors between actual and predicted PAT performance using the BUTU method may also reach 10 % and more.

- Turbine-mode performance related to specific speed of the pump

  It was first <u>Kittredge</u> (1961) who related turbine-mode performance of a pump to its specific speed. He assumed that all pumps with the same specific speed would generate similar head-flow characteristics in the turbine-mode. He presented the test results of four pumps in both modes normalized to their pump best efficiency point. Thus a designer could use a set of normalized pump performance curves and compare these with the Kittredge standards. For those pumps fitting best into one of the four Kittredge characteristics, the turbine-mode performance could be obtained from the corresponding curve of the Kittredge test results. While this method might give accurate results for some pumps of a similar design, it might not for others with different design features or different size. Furthermore, the four test sets presented are not representative for the vast range of different pumps available nowadays.

Diederich (1967) , Buse (1981) and Lewinski-Kesslitz (1987) propose the same approach using specific speed as the decisive parameter. While Diederich and Lewinski-Kesslitz see a direct relation between pump and turbine modes only given for the best efficiency points in both modes, Buse goes even further in this respect and claims that the whole turbine-mode performance curve and the efficiency curve could be predicted as a function of specific speed of a pump. However, Yang (1983) and Yedidiah (1983) have shown, that the correlation assumed by Buse can be very poor for some machines.

- Combined Methods

Recent research by Engeda (1987) aimed at combining both efficiency and specific speed parameters to predict turbine-mode performance. Assuming equal maximum efficiencies in both pump and turbine mode, he related a so-called characteristic product

$$\boxed{\frac{H_{np}}{H_{nt}} \frac{Q_{np}}{Q_{nt}} = \eta_p^3} \qquad (3.12)$$

to specific speed of a pump (all values at the best efficiency points). He further proposed this relationship for approximate PAT selection: using the required turbine data (H_t*Q_t) and trying to solve the equation above for several pump performance data (H_p*Q_p/η_p)bep would lead quickly to the required PAT. Similar to the BUTU method, Engeda provides diagrams to predict turbine-mode performance away from the best efficiency point. Like all other methods presented so far, Engada's proposal works perfectly well for a certain range of pump designs but lacks accuracy for others.

- Others

Calculation of the performance of a pump as a turbine using geometric features of the machine was mentioned by Yedidiah (1983) with apparently "promising results". However, the number of parameters involved and the difficulties to find these makes this method rather inconvenient for practical use.

3.5.4 Practical Selection Procedure of PATs

Turbine-mode performance from actual experimental data/test results

The most reliable method to select a PAT is certainly the use of experimental data from pump manufacturers. The selection of a PAT based on test results virtually excludes errors in the prediction of turbine-mode performance as described in section 3.5.1 above. Unfortunately, manufacturers with experience in the field of PATs are rare and those who have tested some of their standard pumps in turbine mode do not publish the results unless they are asked to make an offer for a specific site. (Some of the more experienced manufacturers offering PATs are listed in Appendix J.)
Additionally, PATs offered by manufacturers are 30 to 100 % more expensive than if the same machine is offered as a standard pump. Manufacturers claim that a higher price would have to be charged to cover the additional costs incurred for developing the know-how on PATs, testing and modifying the standard pump for its use as a PAT. The price advantage of a PAT as compared to a conventional turbine might therefore be lost to some extent. It might be more economical for a micro-hydropower design engineer to predict the turbine mode performance himself, to carry out the necessary checks and modifications (as will be shown below) and to simply order a standard pump. In this case, it is, however, the design engineer who will have to carry the risk of inaccurate prediction of turbine-mode performance or possible damage caused by the reverse operation of the machine.

Predicting the best efficiency point in turbine mode (bep t) from the pump-mode (bep p)

This is the commonly used method for the selection of a PAT. The bep_t is calculated from the bep_p which is usually known for a pump. Using the measured performance data in both modes from over 80 machines, the following diagrams have been compiled. Both main parameters influencing the turbine-mode performance of a PAT, i.e., specific speed and pump efficiency, have been included. The relatively large scatter indicates the uncertainties which originate from undefined geometric features of the pumps tested. (Note, a large diagram to be used as a working tool is given in Appendix D.)

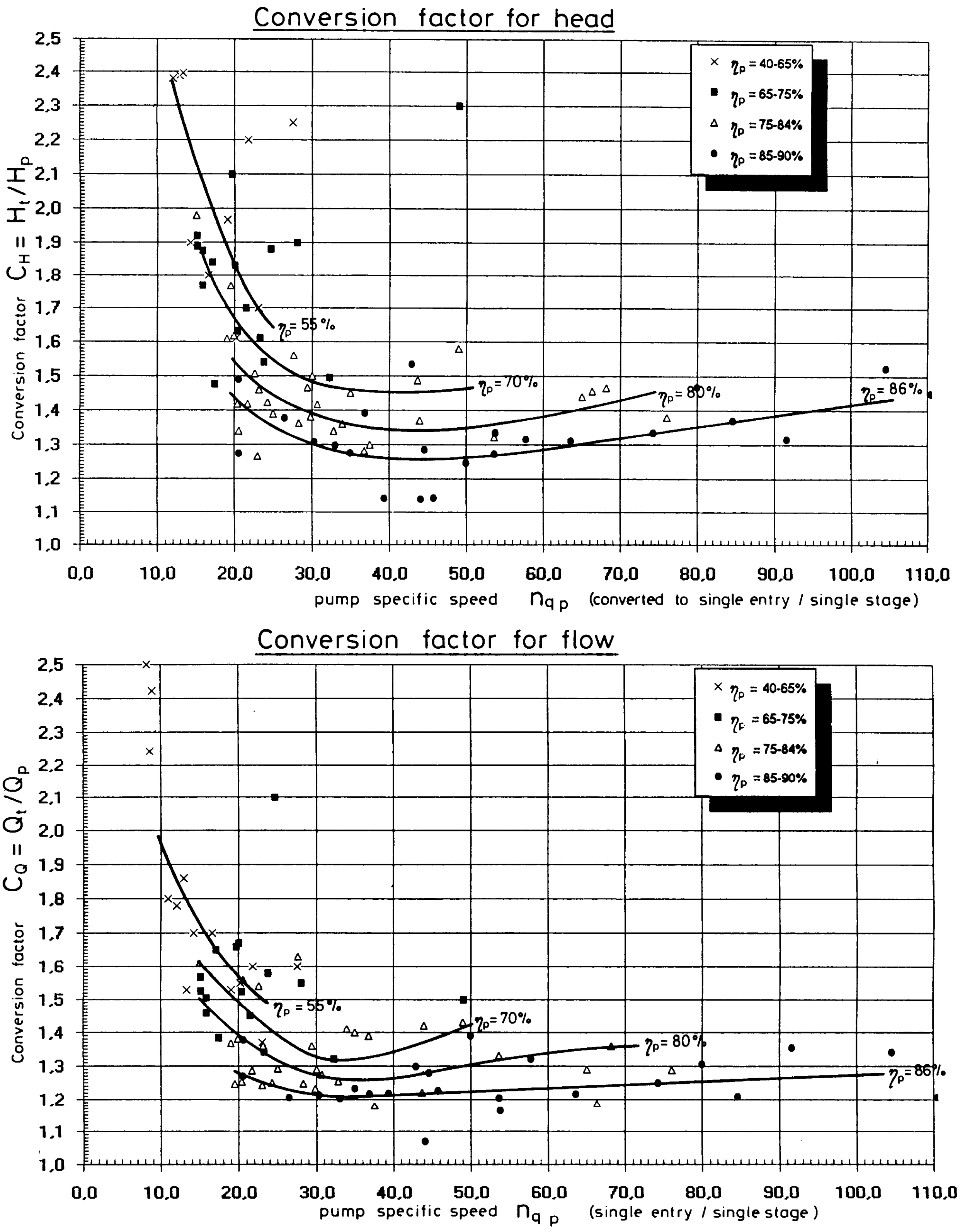


FIGURE 3.19 :
Turbine-mode performance (conversion factors related to pump best efficiency) in function of pump specific speed n_{qp} and maximum pump efficiency $\eta_{p\ max}$

In practice, the problem usually encountered by MHP design engineers is that turbine design conditions are known and the pump which, when used as a turbine, will satisfy these conditions, is to be selected. The following procedure may be adopted:

(A summary / check list of the selection procedure is given in section 3.8 below)

1. **Design head and flow**

From the preliminary design/layout, the following data are known (see section 3.4 above):

- Q_{nt} nominal turbine flow (from flow-duration curve and/or load configurations)
- H_{nt} nominal or net turbine head (from preliminary design)

The PAT to be selected should run (under these head flow conditions) near its best efficiency point (bep t).

Figure 2.8 above indicates the type of pump which would absorb this design flow at the net head available. It assists the designer by indicating what type of PAT he should be looking at when calculating specific speed or reviewing manufacturers' brochures (single stage, multistage, double flow, etc.)

2. **Specific speed in turbine mode**

Calculate the specific speed of the installation (turbine mode):

$$\boxed{n_{qti} = n_t \frac{\sqrt{Q_{ti}}}{H_{ti}^{3/4}}} \qquad (3.13)$$

where n_t is the proposed PAT speed in rpm ($=\text{min}^{-1}$)

If Figure 2.8 indicates that a multi-stage (for high heads) or a double flow PAT should be used, relate the design head and flow to a single-stage and single-entry impeller as follows:

multistage pumps i_{st} = number of stages	multiflow pumps i_{fl} = number of entries
$H_{ti} = \frac{H_t}{i_{st}}$	$Q_{ti} = \frac{Q_t}{i_{fl}}$

Recalculate the specific speed n_{qti} using formula 3.13 above

The **nominal speed in turbine mode** should be chosen as high as possible to limit the size of the PAT . However, the designer must take into account that in turbine mode, the PAT might reach a speed much higher than the pump rated speed, if all loads are removed from the shaft. This maximum speed, called runaway speed, must not be higher than the maximum permissible speed of the selected pump as indicated by the manufacturer. In practice, this maximum permissible speed will usually be 3000 or 3600 rpm (for pumps driven by motors with 60 Hz grid frequency) and the PAT speed to be chosen for turbine operation would therefore be near 1500 rpm as a first estimate (see also section 3.6 below).

The application of the PAT should also be considered: if a generator is to be coupled directly, a nominal speed corresponding to one of the synchronous speeds (e.g. 750, 1000, 1500 or 3000 rpm) should be chosen. For induction generators, slip must be taken into account (see Appendix G). More flexibility, also for driving machinery, is obtained when using transmission gearing, flat or V-belts.

3. **Pump-mode specific speed**

In order to be able to use the diagram 3.19 (conversion factors), the turbine mode specific speed calculated with Formula 3.13 above must be related to a pump-mode specific speed. It has been found from test results that the relation between turbine and pump-mode specific speed takes a fairly constant value: turbine specific speed corresponds to approximately 0.89 times pump specific speed. Figure 3.20 presents this relationship graphically including the underlying test results.

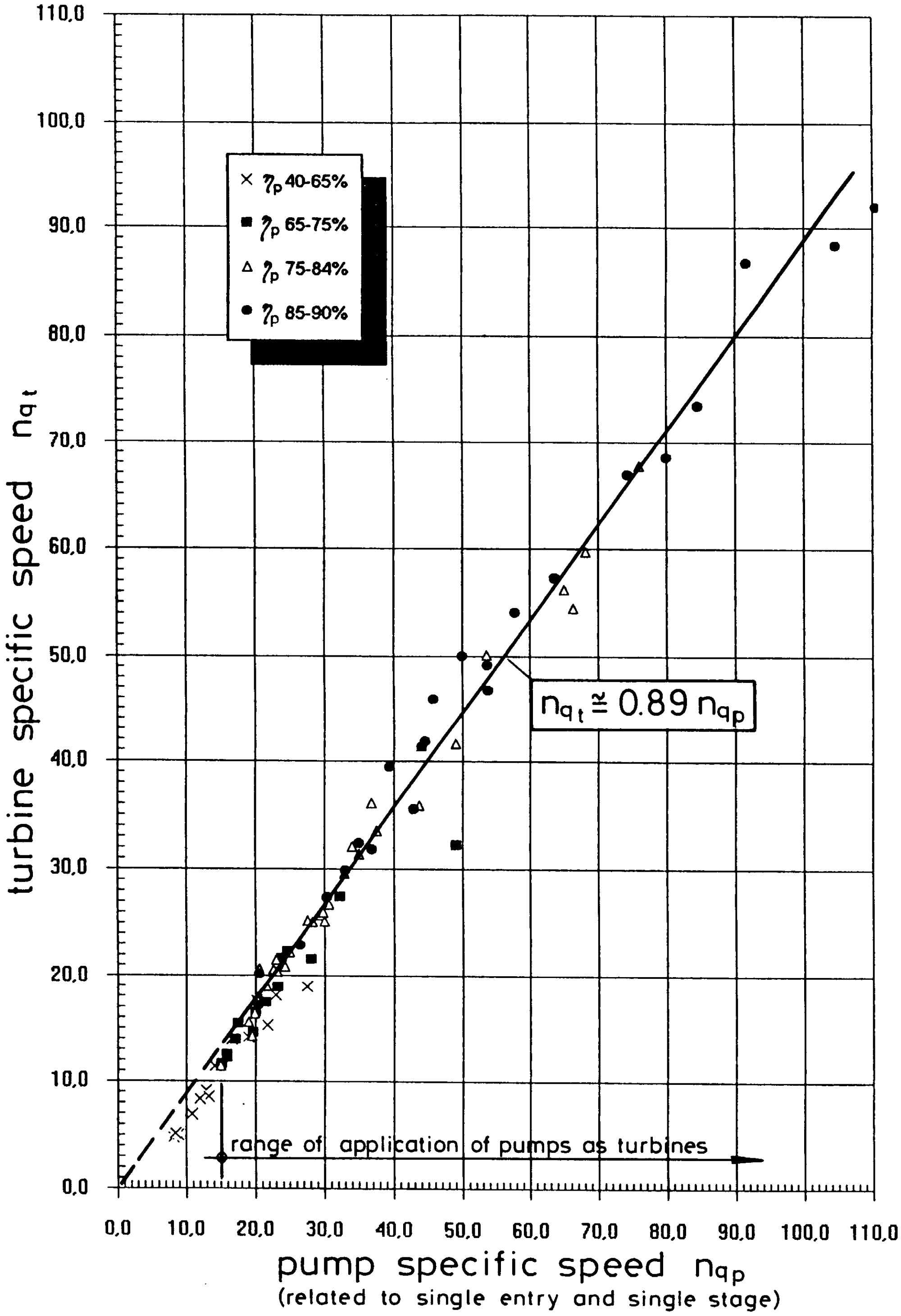


FIGURE 3.20 :
Turbine-mode specific speed versus pump-mode specific speed

$$\boxed{n_{q_p} = \frac{n_{q_t}}{0.89}} \qquad (3.14)$$

Note that **pumps of specific speeds n_{qp} < 15 should not be used as turbines**. Efficiencies of such impellers are low (long and narrow passages incurring friction losses) and the performance of such pumps as turbines cannot be predicted accurately. Additionally, this range of specific speed is generally the domain of small Pelton turbines which can be as advantageous in terms of costs as PATs.

4. **Pump efficiency**

The conversion of turbine design conditions into pump design conditions depends on the efficiency of the machine. Figure 3.21 below indicates the maximum pump efficiency which can be attained by standard pumps for given head / flow conditions. Note that the diagram provides maximum pump efficiencies in function of pump specific speed n_{qp} (use the value from Figure 3.20 above) and rated pump flow. The latter is not yet known. Using the **available flow Q_t divided by 1.3** (=average conversion factor) provides sufficiently accurate values for a pre-selection.

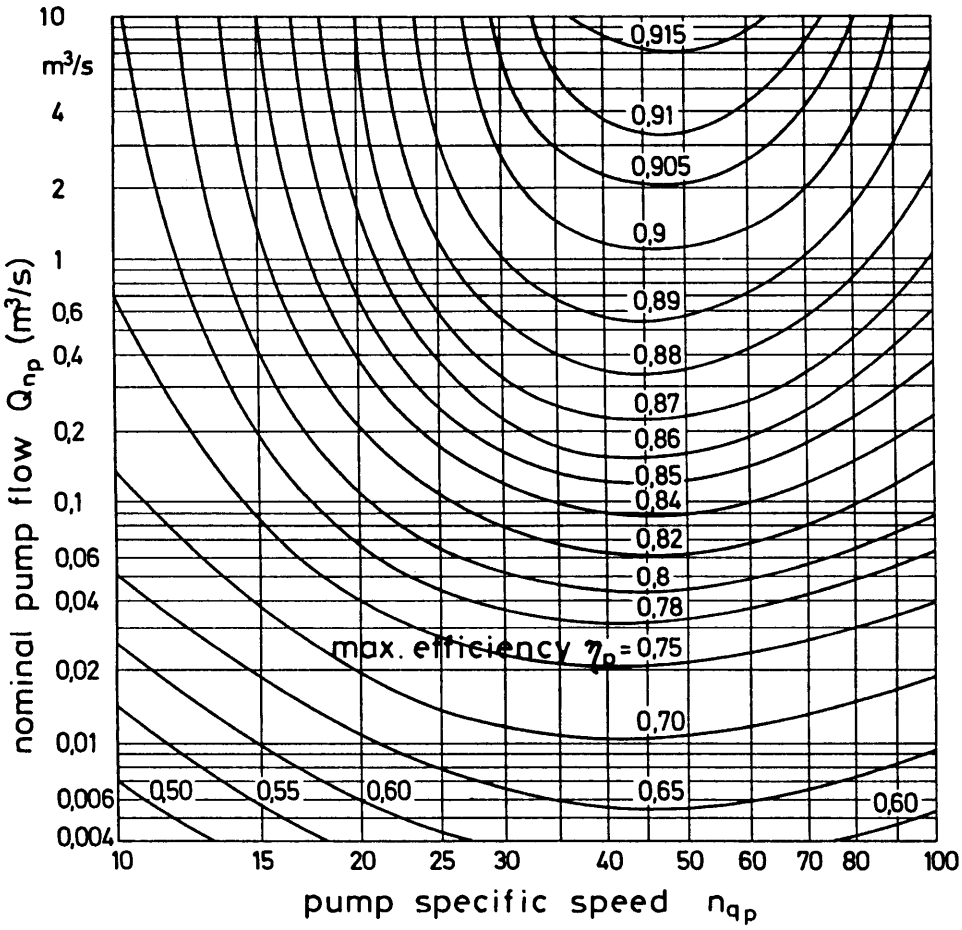


FIGURE 3.21 :
Maximum pump efficiency in function of specific speed n_{qp} and flow Q_{np}
(source: Willi Bohl, Stroemungsmaschinen, Berechnung und Konstruktion)

5. **Converting turbine design conditions into pump design conditions**

Entering Diagram 3.19 above with pump specific speed and efficiency, read the conversion factors for head C_H and flow C_Q

conversion factor for head	conversion factor for flow	
$\frac{H_{nt}}{H_{np}} = C_H$	$\frac{Q_{nt}}{Q_{np}} = C_Q$	(3.15)

Applying the conversion factors C_H and C_Q on the design head and flow in turbine mode, we obtain pump performance parameters at bep p (pump best efficiency point) at the proposed turbine speed:

nominal pump head at turbine speed n_t	nominal pump flow at turbine speed n_t	
$H_{np\,(n_t)} = \frac{H_{nt}}{C_H}$	$Q_{np\,(n_t)} = \frac{Q_{nt}}{C_Q}$	(3.16)

6. **Converting pump design conditions at turbine rated speed into pump rated speed**

The general selection charts in pump manufacturers' brochures indicate the type of pump and its rated speed which would probably accommodate the required head/flow conditions for the installation. However, the rated pump speed n_p does in most cases not correspond to the proposed turbine speed n_t .Therefore, the head/flow values H_{np}/Q_{np} according to formula (3.16) must be transformed into new head/flow conditions valid for nominal pump speed n_p:

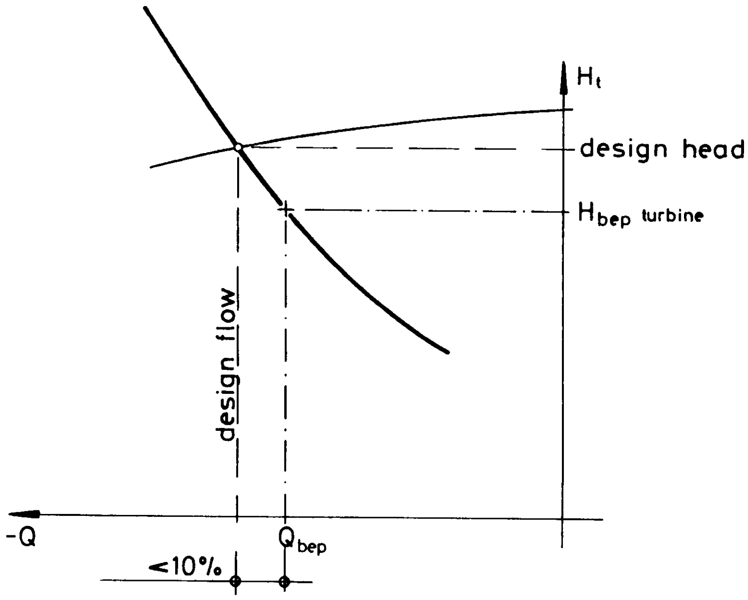
The affinity laws apply (see also Appendix B):

nominal pump head at pump speed n_p	nominal pump flow at pump speed n_p	
$H_{np\,(n_p)} = H_{np\,(n_t)} \left[\frac{n_p}{n_t}\right]^2$	$Q_{np\,(n_p)} = Q_{np\,(n_t)} \left[\frac{n_p}{n_t}\right]$	(3.17)

If no standard pump is available for these head/flow conditions at the proposed pump speed, recalculate from point 2 using a different turbine speed or assume a different number of impeller stages or entries.

7. **PAT selection recommendations**

Usually, a pump generating exactly these head-flow conditions might not be found in manufacturers' charts. Select a pump with a rated flow (Q_{np}) slightly lower than the value required according to formula (3.16) above. The PAT will then operate at a flow higher than the rated flow in turbine mode (beyond bep_t), i.e., **it will operate in the overload range**. This is a very important feature of PAT design since it helps to make errors and inaccuracies in the prediction of the turbine-mode performance less severe. Details of the advantage of such a layout are given in Section 3.5.5 below.



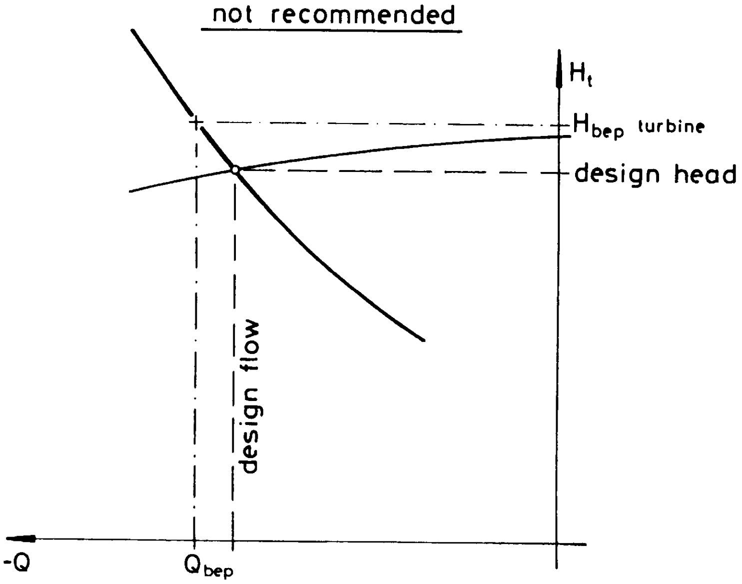


FIGURE 3.22 :
Principle of PAT selection: the PAT should operate in the overload range in turbine mode

8. **Converting the best efficiency point of the selected pump into turbine mode**

a) Calculate the specific speed of the selected pump (best efficiency head and flow related to single-single and single-entry conditions i)

$$n_{qpi} = n_p \frac{\sqrt{Q_{pi}}}{H_{pi}^{3/4}} \qquad (3.18)$$

multistage pumps i_{st} = number of stages $H_{pi} = \frac{H_{np}}{i_{st}}$	multiflow pumps i_{fl} = entries $Q_{pi} = \frac{Q_{np}}{i_{fl}}$

b) Read the conversion factors for head and flow from Figure 3.19 above using the maximum efficiency of the selected pump as indicated by the manufacturer.

> **Important note:** As explained above the accuracy of the conversion of pump performance data into turbine-mode performance is limited. It is unlikely that the PAT will yield exactly the computed performance. In order to estimate the possible **performance range** in turbine mode, apply the conversion twice, once using minimum conversion factors and once with maximum factors.
> The following scattering factors are proposed:
>
> **- on the conversion factor for head: ± 10 %**
> **- on the conversion factor for flow : ± 7.5 %**

conversion factor for head: $C_{H\,max} = 1.1 * C_H$ $C_{H\,min} = 0.9 * C_H$
conversion factor for flow: $C_{Q\,max} = 1.075 * C_Q$ $C_{Q\,min} = 0.925 * C_Q$

c) Calculate the turbine best efficiency point (bep t) for both maximum and minimum conversion factors

nominal turbine head at pump speed n_p	nominal turbine flow at pump speed n_p	
$H_{nt\,(n_p)} = C_H \, H_{np}$	$Q_{nt\,(n_p)} = C_Q \, Q_{np}$	(3.19)

d) Convert these turbine bep conditions into the nominal turbine speed n_t using the affinity laws as under point 6 above:

nominal turbine head at turbine speed n_t	nominal turbine flow at turbine speed n_t	
$H_{nt\,(n_t)} = H_{nt\,(n_p)} \left(\frac{n_t}{n_p} \right)^2$	$Q_{nt\,(n_t)} = Q_{nt\,(n_p)} \left(\frac{n_t}{n_p} \right)$	(3.20)

9. **Power output in turbine mode at the best efficiency point**

Test results have shown that maximum efficiency in turbine mode is on average slightly lower than in pump mode. In the following, we propose a turbine maximum efficiency of 3 % below pump maximum efficiency. This assumption is based on Figure 3.23 below showing the test results of around 70 pumps.

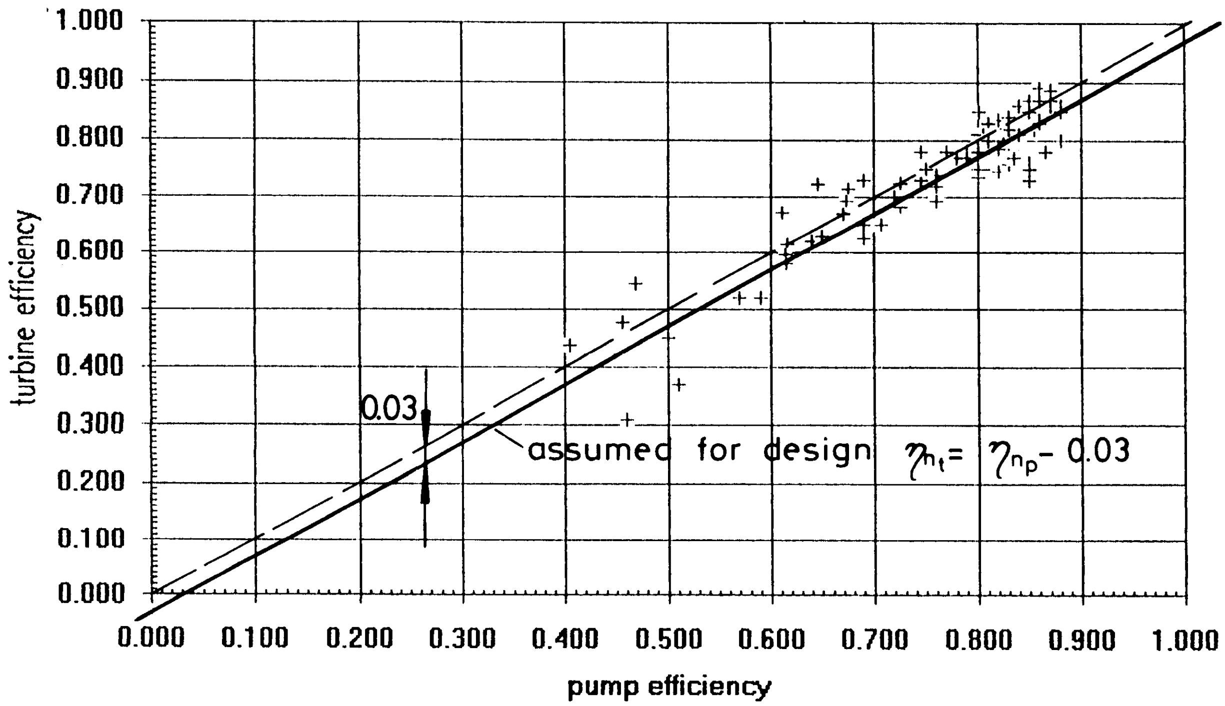


FIGURE 3.23 :
Turbine best efficiency versus pump best efficiency and assumed law for design computations

$$\boxed{P_{nt} = \rho\, g\, Q_{nt}\, H_{nt}\, (\eta_{p\ max} - 0.03)} \qquad (3.21)$$

10. Predicting PAT performance away from the bep

The head/flow conditions (both minimum and maximum) calculated above will most probably not coincide with the design conditions. In order to determine the flow absorbed by the selected PAT under the net turbine head available, we must predict the PAT performance curve away from its best efficiency point. Based on test results, the following figures present an approximation of turbine-mode performance curves (head and efficiency versus flow) in function of specific speed and normalized to the turbine best efficiency point (H_{nt}/Q_{nt}). Note that the accuracy of these curves diminishes rapidly for operating points further away from turbine bep.

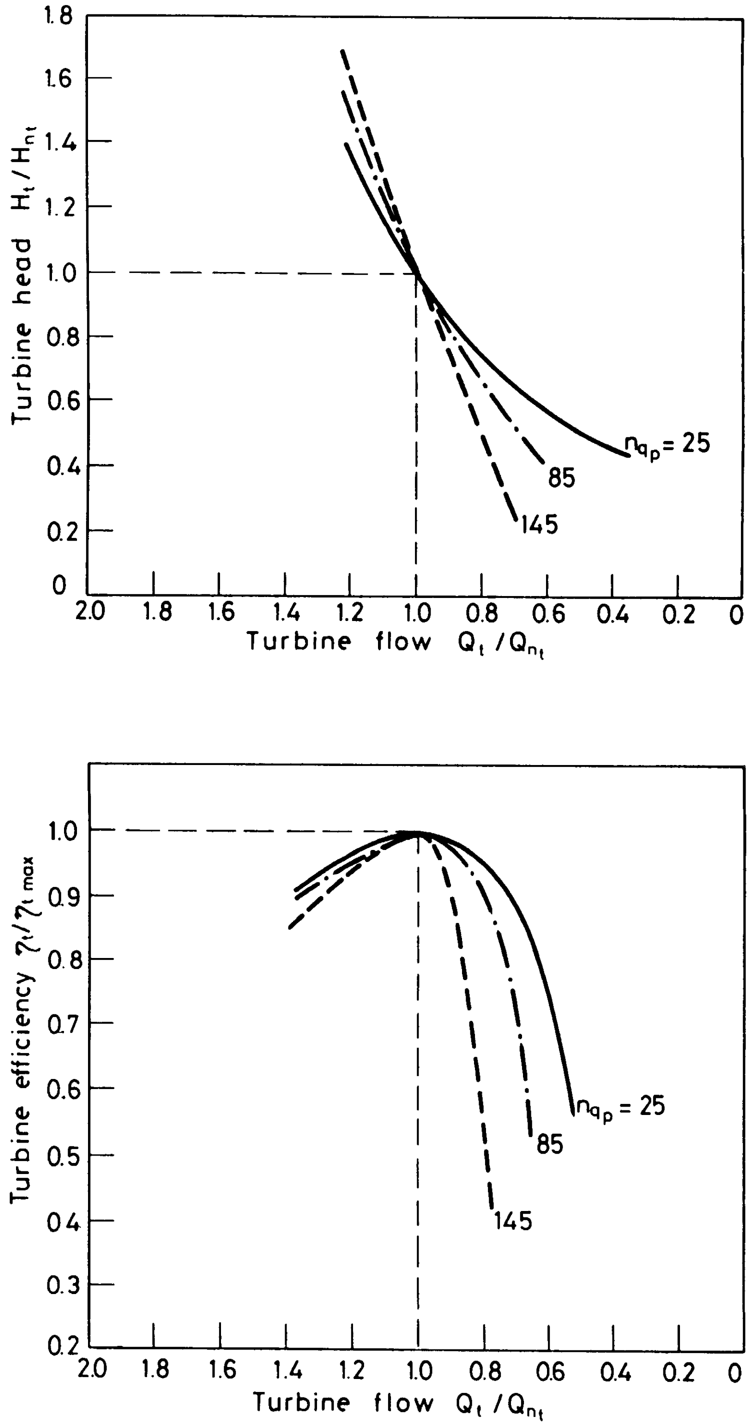


FIGURE 3.24 :
Approximation of PAT performance away from bep point in function of specific speed

Figure 3.24 shows the general tendency of turbine-mode characteristics in function of specific speed. It is not intended to be used for actual computations. Appendix D provides diagrams which facilitate the reading of the head and power values away from the best efficiency point (see Diagrams D4 and D5). Head and power (normalized to rated turbine head and output) are given for four different flow values (0.8, 0.9, 1.1 and 1.2 Q_{nt}) in function of pump specific speed. Using these four points, approximate turbine performance curves for head and power can be plotted.

The intersection points of the system resistance curve(s) (see Appendix A) and the predicted PAT performance curves (maximum and minimum values) provide the possible PAT operating range (see Figure 3.25). Note that the system resistance curve is not a constant line as assumed so far. The friction coefficient (see Appendix A) of the penstock pipes used in the computation may vary considerably both from product to product and over the lifetime of the installation (deposits in the pipes). Using maximum and minimum friction coefficients, the possible operating range of the selected PAT increases further.

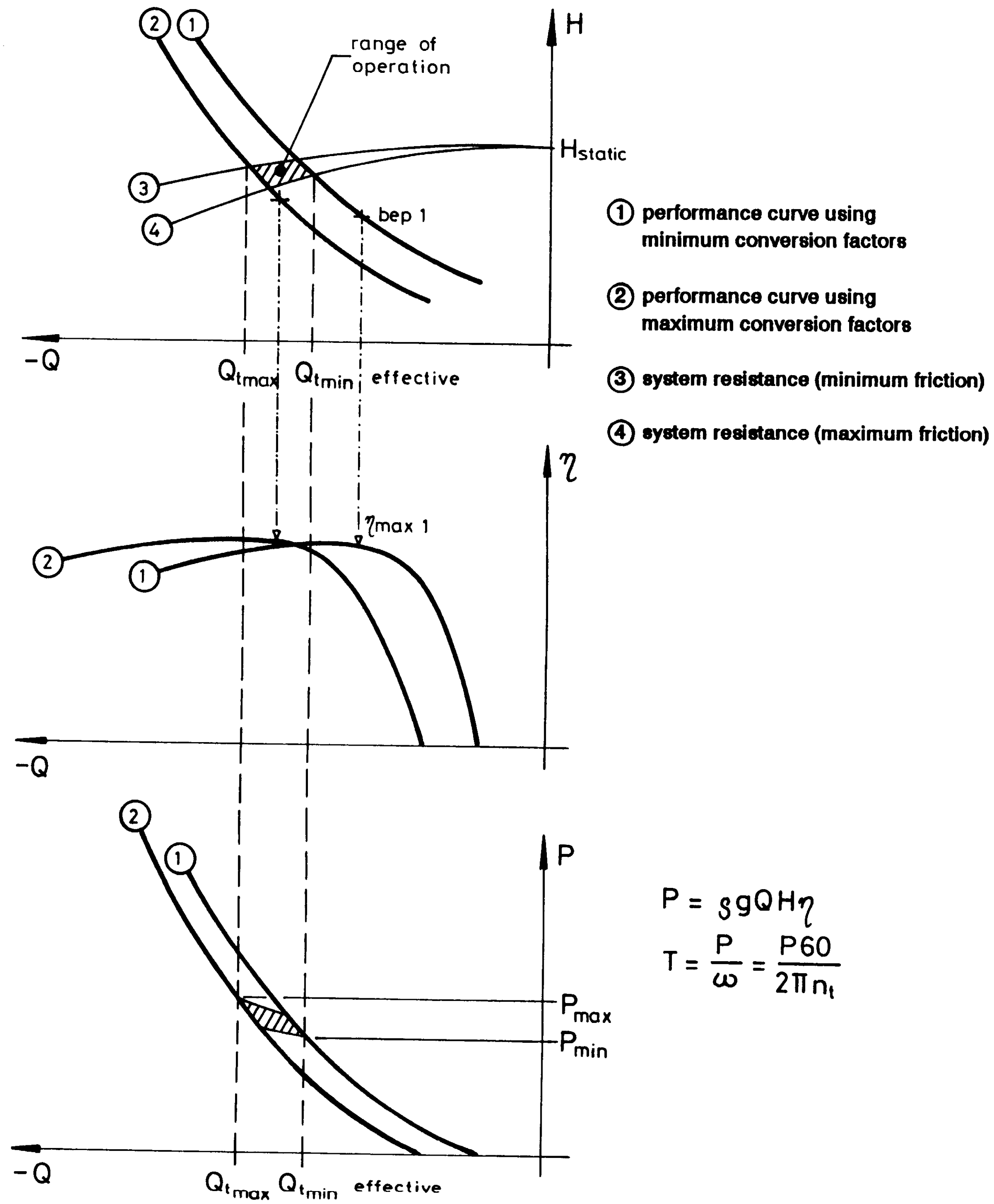


FIGURE 3.25 :
Operating range of the selected PAT

The designer must now determine whether :

a) the flow ($Q_{eff\,t\,max}$) absorbed by the selected PAT is acceptable as regards to the available flow (available flow > estimated maximum flow $Q_{eff\,t}$)

b) the actual power ($P_{eff\,t\,min}$ or $_{max}$) delivered by the PAT suits the application (machinery or generator)

$$\boxed{P_{eff\,t} = \rho\, g\, \eta_{eff\,t}\, Q_{eff\,t}\, H_{eff\,t}} \qquad (3.22)$$

$P_{eff\,t\,max}$ is used to select the generator rating (generator rated power 20 to 30 % above $P_{eff\,t\,max}$)
$P_{eff\,t\,min}$ should cover the consumer needs

c) the max pressure head and the torque acting on the PAT are within the limits given by the manufacturer of the pump (further details see section 3.7)

If one or several of these conditions are not acceptable, another PAT should be selected. If no better choice is available, modifications on certain design particulars and/or operating conditions might be considered.

3.5.5 Some Details on Appropriate PAT Selection

□ **Impeller design**

Pumps are sometimes available with special impeller designs to avoid clogging in case of solids in the water. However, there is no evidence that these impellers (open impellers, large passages due to reduced number of vanes) will handle a large solid content in the turbine mode as favourably as in the pump mode. Before further test results are available, we recommend that a conventional runner be chosen and the water be cleaned as much as possible (trash racks, sand traps).

□ **Spiral casing versus fixed guide vanes (diffusor)**

From the point of view of PAT performance, both designs are almost identical; if, however, the actual operating point does not match the desired head/flow characteristic, the diffusor type pump is to be preferred since the ring of fixed guide vanes could be adjusted to new conditions by grinding and/or welding on the vanes to alter passages and inlet angles (see also section 3.6.3).

□ **Layout point of a PAT - the advantage of selecting a slightly overloaded pump**

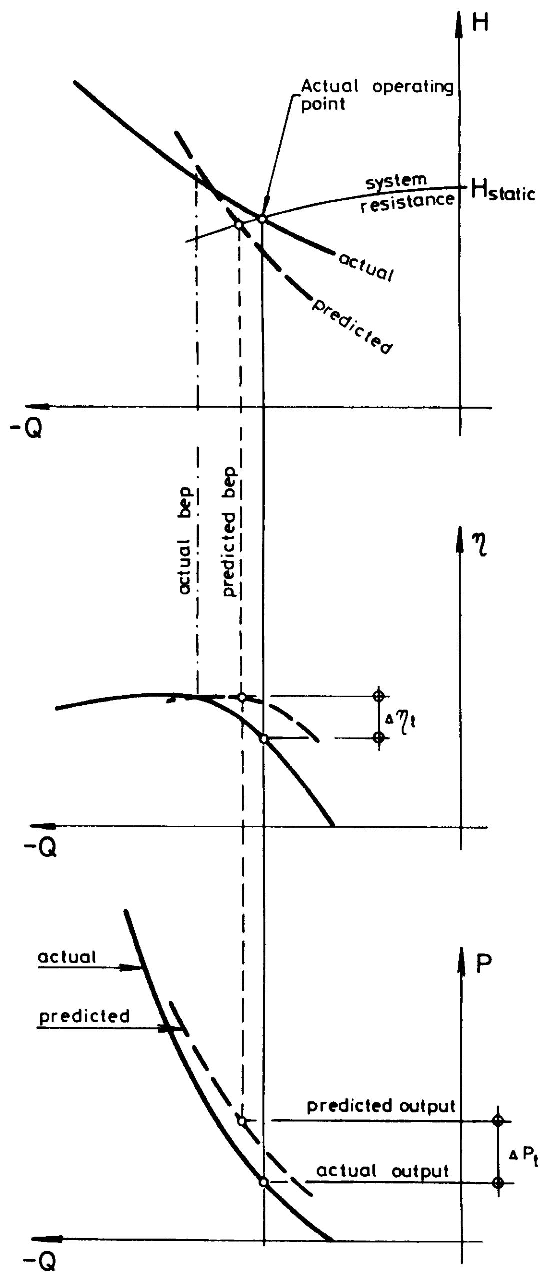
As a general rule, the PAT should operate slightly in the overload range, i.e., the expected operating point should lie beyond the calculated flow for best efficiency. In this way, we can make use of the fact that efficiency in turbine mode does not drop with increasing flow as rapidly as in pump mode since the losses are referred to a higher power output (in turbine mode, head increases with flow). Thus, the desired operating point lies in a relatively flat area of the efficiency curve; and if the actual head and flow absorbed by the PAT is lower than expected, efficiency does not fall off, on the contrary, it might even be closer to the bep.

Figure 3.26 below shows two possible design arrangements: on the left, the design engineer has chosen a PAT which should, according to his calculations, operate at its best efficiency point (broken line = expected PAT performance curve). However, it turns out after having installed the selected PAT that it has a slightly different performance curve (20 % higher H_{nt} and 20 % higher Q_{nt} than assumed). The actual operating point of this PAT falls into a range of low efficiency and the expected power P_t is by far not delivered.

The performance curve in Figure 3.26 on the right shows the same site but with a PAT which has been selected to operate slightly beyond its best efficiency point. The same deviation between expected and actual PAT performance is assumed but this time, the actual power delivered is even higher than predicted (see also section 3.5.7).

Not recommended design principle:
design flow < bep flow

Low efficiency and output due to poor correlation between predicted and actual turbine performance



Highly recommended:
PAT operating point slightly in the overload range

High efficiency and output is maintained even at poor correlation between predicted and actual turbine performance

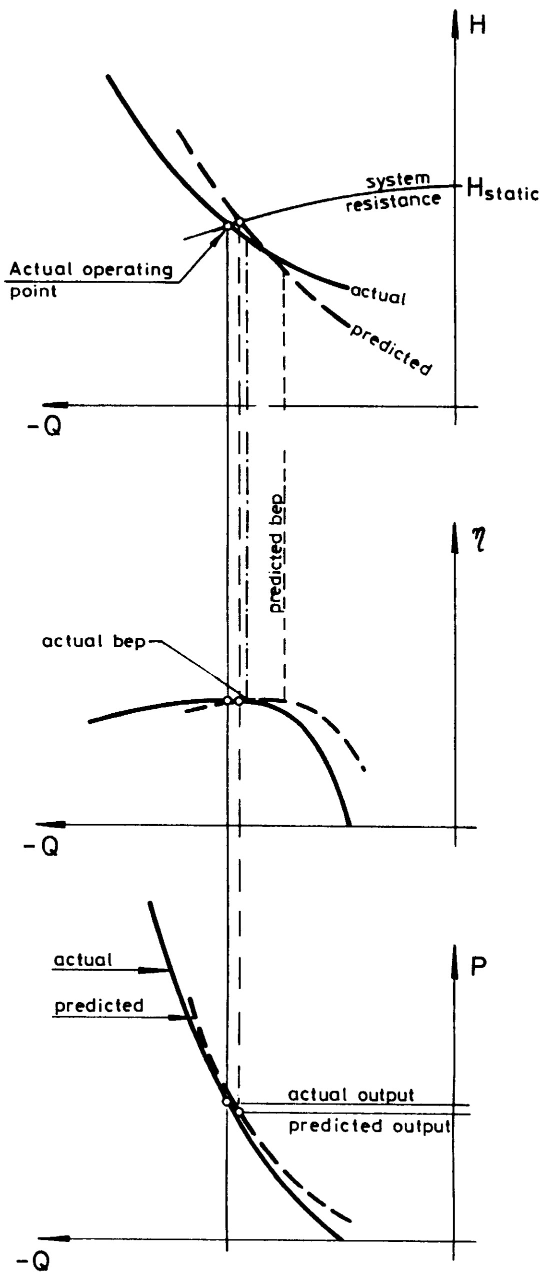


FIGURE 3.26 :
The advantage of PAT operation beyond the best efficiency point

□ **Impeller diameter and speed variations**

Performance graphs of pump manufacturers usually include a series of curves with both trimmed impellers (mainly for radial flow and less frequently for mixed flow pumps) and various nominal speeds. When selecting a PAT, it is possible to use a standard pump with a trimmed impeller and/or at other than rated pump speed. However, the effect of trimming and varying speed in turbine mode is very different from the well-known effects in pump mode: while the variations in pump mode show a tendency to shift the head/flow curve in a direction approximately perpendicular to the curve, in the turbine mode, each curve forms almost a continuation of the curve at a different impeller trim and at different speed (see Figure 3.27 below).

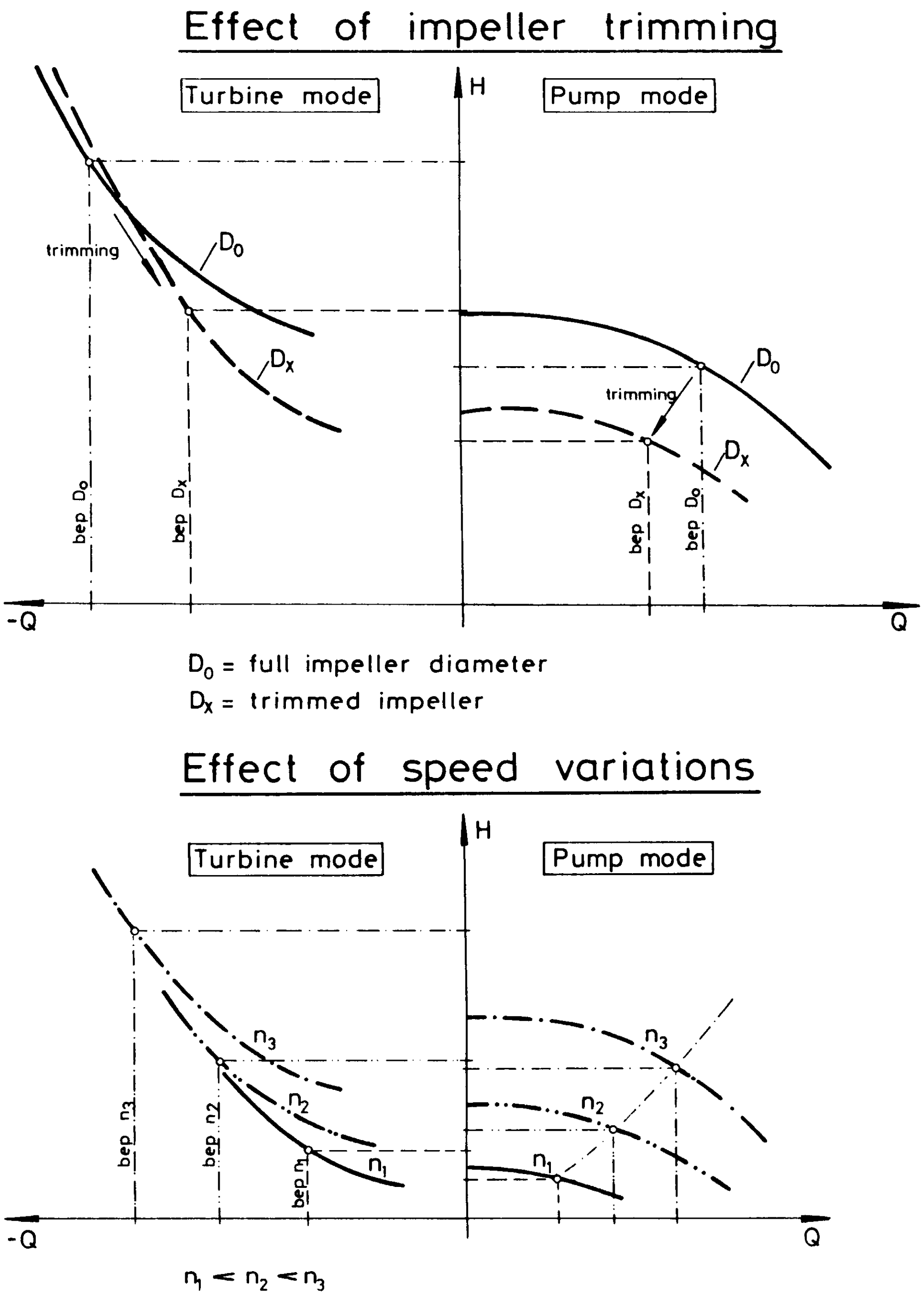


FIGURE 3.27 :
Performance curves in turbine and pump mode for different speeds and impeller trims

The conversion factors of Figure 3.19 above which are used to determine turbine mode performance from pump best efficiency (bep), are valid for different pump speeds and, with slightly less accuracy, also for trimmed impellers. Details about trimming are given in section 3.6.4 below.

3.5.6 Predicting PAT performance from pump geometry

In some cases, no performance data of pumps are available, neither performance curves nor nominal head, flow and speed data. To estimate the turbine mode performance of such a pump, the geometry of the impeller can be measured and approximate pump bep data may be established using Figure 3.28 below. Note, this rough estimate may be verified by comparing the result of this computation with a pump of similar geometry but whose performance is known from tests.

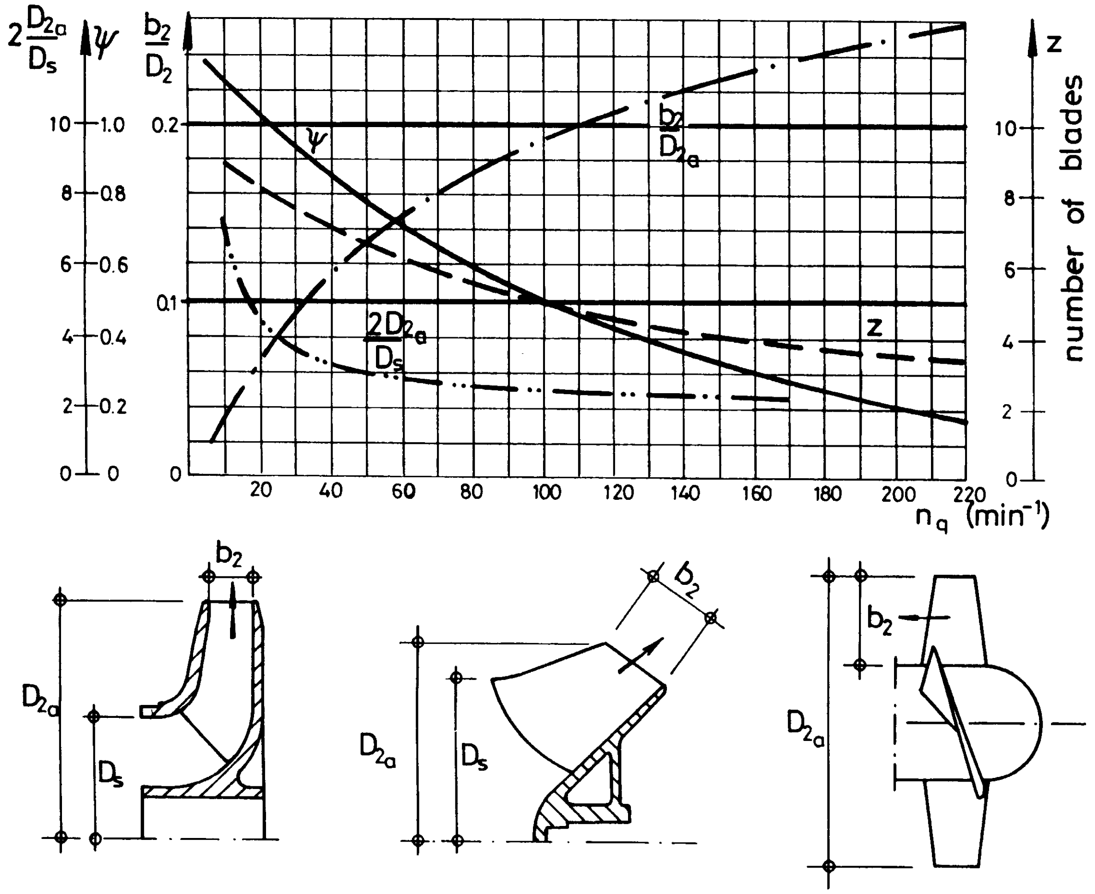


FIGURE 3.28 :
Establishment of pump bep data from impeller geometry
(source: Technisches Handbuch Pumpen, VEB Verlag, Berlin)

The application of Diagram 3.28 is as follows:

1. measure the impeller geometry (D_s, D_{2_a} and b_2 according to Figure 3.28)

2. calculate the ratios $2*D_{2_a}/D_s$ and b_2/D_{2_a}. These ratios and the number of blades z should give approximately the same specific speed n_{qp} on the abscissa of Diagram 3.28.

3. for this specific speed n_{qp}, read the value of ψ

4. ψ is called the energy number and is defined as follows:

$$\psi = \frac{2\,g\,H_{np}}{u_{2_a}^2} \qquad u_{2_a} = \frac{2\,\pi\,n_{np}}{60}\,\frac{D_{2_a}}{2} \qquad (3.23)$$

Nominal pump speed n_{np} is still unknown. For direct-coupled pumps, n_{np} can take either 3000, 1500, 1000 or 750 rpm corresponding to the synchronous speed of a 2, 4, 6 or 8 pole electric motor (50 Hz grid frequency).

5. calculate the nominal pump head for the four synchronous speeds:

$$\boxed{H_{np} = \frac{\psi\, u_{2_a}^{2}}{2\,g}} \qquad (3.24)$$

6. Now the nominal pump flow remains to be determined; calculate the flow for the same four synchronous speeds as for the head above using the well-known formula for specific speed n_q:

$$\boxed{Q_{np} = \frac{H_{np}^{1.5}\; n_{qp}^{2}}{n_{np}^{2}}} \qquad (3.25)$$

7. Knowing nominal pump head, flow and speed, the problem has been reduced to the determination of the turbine-mode performance from pump-mode data as presented in section 3.5.4 above.

3.5.7 General Range of Operation of a PAT at a Given Site

Basically, the range of operation of a PAT is formed by four curves (see Figure 3.29):

1. **No-load line** where torque T on the shaft is zero. The intersection point between the system resistance and this no-load curve represents the runaway speed of the PAT for each differential head (see also section 3.6.1).

2. **Standstill line** which represents the flow through the PAT for varying head if the impeller is completely blocked; for each differential head, torque on the shaft is maximum along the standstill line. Note that each performance curve for constant speed eventually merges with the standstill line at high heads, i.e. it approaches the standstill line asymptotically.

Mechanically, the range of operation of a PAT is limited by:

3. **Line of permissible torque** on the shaft (maximum shaft resistance, see section 3.7)

4. **Line of maximum rotational speed** of the machine (structural integrity of the rotating elements; see section 3.7)

   The practical lower limit of PAT operation is usually determined by economic considerations: a PAT operating below a minimum efficiency generates a very low power output which is practically of no use. Furthermore, operation at such low part loads is usually accompanied by irregular flow and undue wear on the PAT. Hence, a line of minimum efficiency (see line No. 5 in Figure 3.29, assumed 60 % of maximum) would be adopted.

For a given site, this range of operation (see hatched area in Figure 3.29) is further limited by the system resistance curve, i.e available static head reduced by penstock losses. The actual operating conditions may therefore not cover the area of high heads (transient conditions excluded).

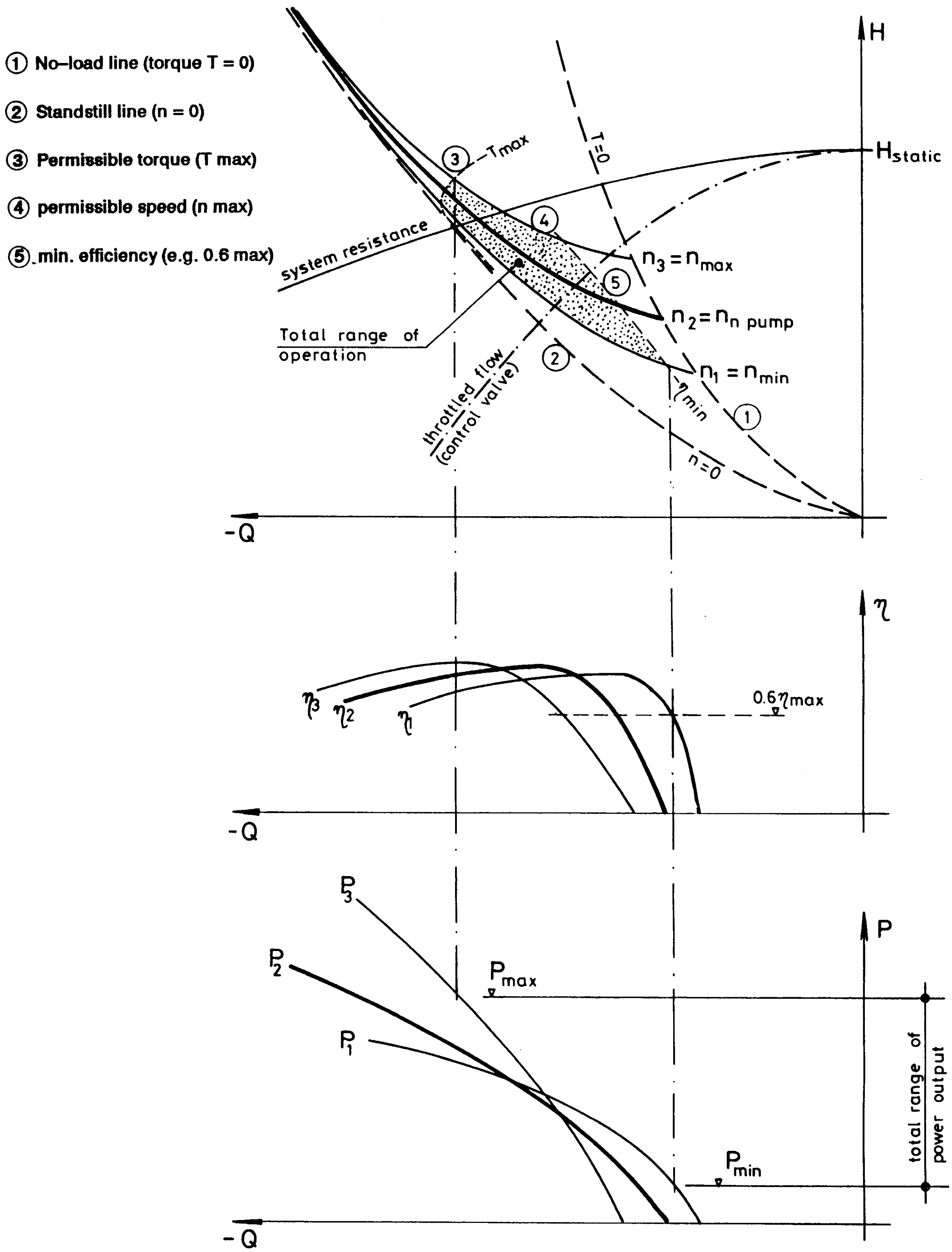


FIGURE 3.29 :

General Range of operation of a PAT including variation of speed and system resistance (control valve)

3.6 Design Particulars

3.6.1 Runaway speed

Runaway speed is defined as the speed of the turbine or PAT at a fixed head but with zero power output. The corresponding flow of this head and speed is called the runaway flow. Both runaway speed and flow depend on the specific speed of the machine. Design and efficiency have only a minor effect on the runaway conditions of a PAT. The high specific speed machine will have a higher runaway speed and flow than low specific speed pumps. Figure 3.30 shows the PAT runaway speed and flow (relative to pump-mode bep) versus pump specific speed n_{qp} for a number of tested machines. The relatively large scatter indicates the uncertainties involved especially for small machines where friction loss in seals (tight stuffing box) can have a great effect on the runaway conditions. The straight line in both graphs proposes the runaway speed and flow to be used for design calculations. Note that runaway speed and flow have been **related to pump-mode data**. The factors ε and κ indicate in fact the speed and flow (relative to rated values) of a pump running in reverse ($n_{p\,R}^{-}$) under rated pump head H_{np}. This situation can occur in pumping installations upon motor failure (pipeline without non-return valve) and is therefore an important parameter for the design of pumping plants. Thus, even without indicating the intended application as PAT, pump runaway speed and flow data may be available from the pump manufacturer. Whenever possible, ask your pump supplier for these data on maximum speed and flow for reverse pump operation (at rated pump head).

In order to compute the runaway conditions for the effective **operating point in turbine mode**, the following formulae apply:

$$\text{definitions:} \quad \varepsilon = \frac{n_{p\,R}^{-}}{n_{n_p}} \quad \text{at rated pump head } H_{n_p} \qquad (3.26)$$

$$\kappa = \frac{Q_{p\,R}^{-}}{Q_{n_p}} \quad \text{at rated pump head } H_{n_p} \qquad (3.27)$$

where $n_{p\,R}^{-}$ = pump runaway speed in reverse (as turbine) at rated pump head H_{n_p} and

$Q_{p\,R}^{-}$ = flow at this speed and head (direction of flow in reverse to normal pump flow = turbine-mode)

$$\boxed{n_{t_R} = \varepsilon\, n_{n_p} \sqrt{\frac{H_{t_R}}{H_{n_p}}}} \qquad (3.28)$$

$$\boxed{Q_{t_R} = \kappa\, Q_{n_p} \sqrt{\frac{H_{t_R}}{H_{n_p}}}} \qquad (3.29)$$

where H_{t_R} is the head of the PAT at steady state runaway flow Q_{t_R} (which will in most cases not coincide with the effective turbine head $H_{t\,eff}$ or the rated turbine head H_{nt} at the best efficiency point). Since H_{t_R} and Q_{t_R} depend on each other, an iterative calculation must be applied (see below).

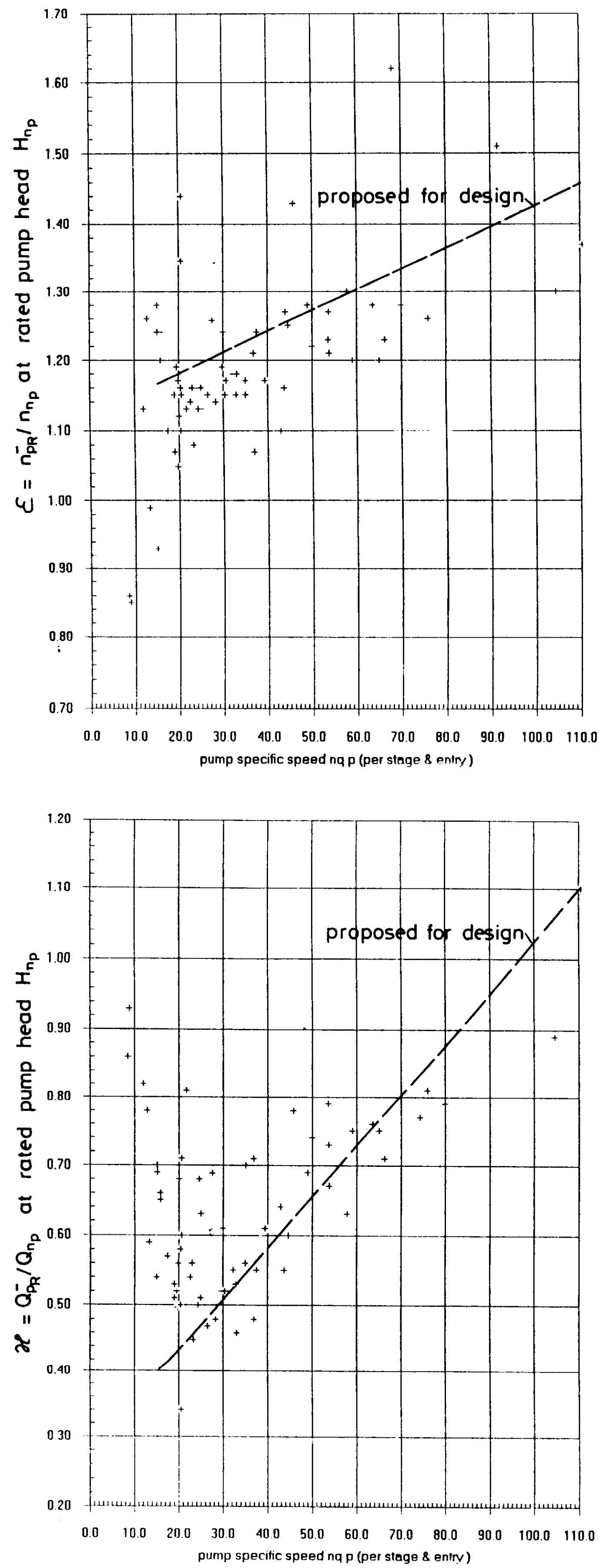
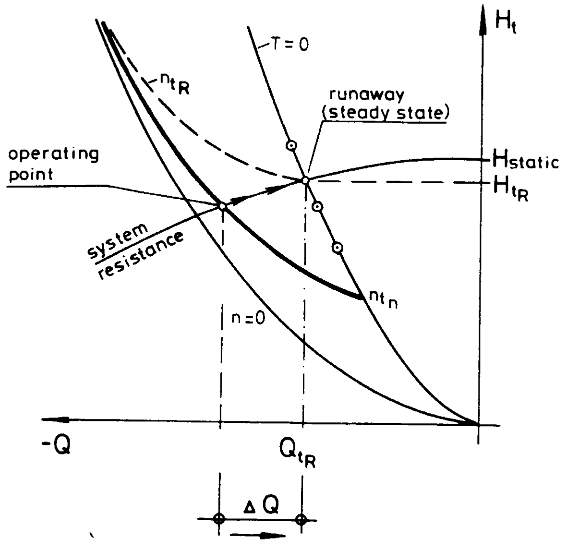


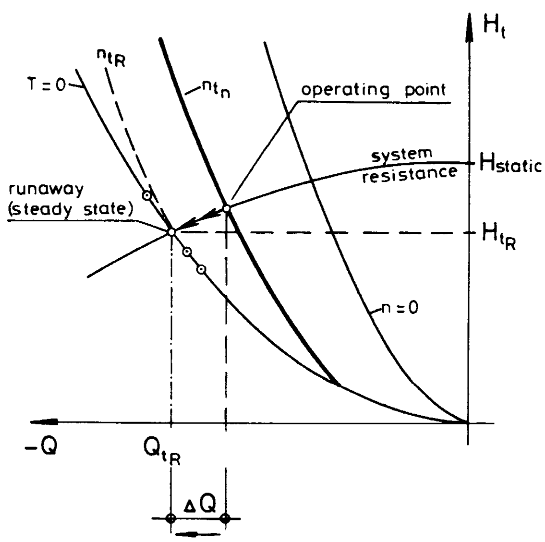
FIGURE 3.30 :
Pump runaway speed and flow (at rated pump head) versus pump specific speed

Runaway speed in turbine mode occurs, when (under normal operating conditions) all external loads on the PAT are removed. In such a case, speed increases and the operating point moves theoretically along the system resistance curve to the runaway conditions on the no-load line (see Figure 3.31 below). Using the diagram 3.30 and the corresponding formulae above, turbine runaway speed and flow can be determined. Since runaway head is not yet known, the runaway conditions must be found by trial and error. Assume two to three pressure heads H_t near the expected runaway conditions on the system resistance curve and calculate the runaway flow Q_{t_R} for each head using formula 3.29. Enter these points into your H-Q diagram (turbine performance curve) and plot the no-load line. The intersection of this no-load line with the system resistance curve gives the runaway conditions. Using the head of this point (H_{t_R}), PAT runaway speed (n_{t_R}) can be determined (Formula 3.28).

radial flow PAT



axial flow PAT



subscript R = runaway conditions

FIGURE 3.31 :
Determining runaway conditions for a given site for both radial and axial flow PATs

The flow at runaway conditions for a **low specific speed PAT** (radial flow) is less than at normal turbine operation, i.e. the PAT acts like a valve when load is removed (see Figure 3.31). This is due to the centrifugal forces acting on the fluid in the radial impeller producing a backpressure when speed increases. Therefore, flow through the PAT diminishes at increased speed.
For **axial flow impellers** , the situation is reversed. When the torque is removed from the impeller, the speed increase is high and so is the flow through the PAT (see Figure 3.31) since the propeller of an axial flow machine does not exert as much resistance to the flow as a radial flow impeller with its long curved passages.
The PAT and the coupled machines or generators must be designed to stand runaway conditions without damage.

In Figure 3.31 above the transition from the nominal operating point to steady state runaway conditions was assumed to develop slowly and gradually along the system resistance curve. In practice, however, the change of flow from one condition to the other takes place in a short time and may produce pressure surges in the penstock. Since the rotating elements of PATs usually have relatively low inertia, the effects of acceleration or deceleration are more severe for PATs than for purpose-built turbines. Pressure surges (= variation of pressure head) in the penstock during the transient conditions may drive the PAT temporarily to higher speeds than according to the steady state runaway condition. Both PAT and driven machinery or generators must be designed to stand also the speed under transient conditions. Whether pressure surges are likely to be severe and cause high runaway speed depends in the first place on the penstock and the hydraulic conditions around the PAT (see Section 4.4 Waterhammer). The maximum speed of the PAT may be estimated with the following formula which is based on the affinity laws (see Appendix B):

$$\boxed{n_{tRmax} = n_{tRsteady\ state}\sqrt{\frac{H_{max}}{H_{Rsteady\ state}}}} \qquad (3.30)$$

where H_{max} = maximum head according to section 4.4 Waterhammer
$H_{Rsteady\ state}$ = head corresponding to steady state runaway speed (see Figure 3.31 above)
$n_{tRsteady\ state}$ = steady state runaway speed according to Formula 3.28 above

When selecting a PAT, the manufacturer must guarantee that :
- the integrity of all rotating elements are assured at maximum runaway speed (n_{tRmax})
- the bearings are designed for this speed and
- lubrication is assured also at n_{tRmax}

The same applies to generators or machinery driven by the PAT (use the corresponding speed if a transmission gearing is installed).
The pressure surges invoked by the speed increase of the PAT may lead to waterhammer problems which are treated in detail in Chapter 4.

3.6.2 Cavitation

Just as a pump requires a minimum net positive suction head (NPSH), the turbine-mode of the same machine requires a certain backpressure called TREH (total required exhaust head) to avoid cavitation. While cavitation is most probable to occur first near the innermost tips of the impeller vane in the pump mode, the same discontinuity will develop vortices and finally cavitation also in turbine mode. But these conditions occur at a much lower pressure due to the reverse flow. Generally, TREH for purpose-built turbines is about 50 % of the net positive suction head (NPSH) of a pump of similar n_q and with similar head. We can therefore conclude, that the TREH for PATs must be somewhere in between the values for purpose-built turbines and pumps (see hatched area in Figure 3.32).

The principal factor affecting the total available exhaust head is the setting of the PAT,i.e. the distance of the machine above the tailwater. If it is too high, even a well-designed impeller might start to cavitate (for details of calculation see Appendix A).

To have a single value expressing cavitation independent of speed, the so-called Sigma value (or Thoma number) may be introduced which reads as follows:

$$\boxed{\sigma_p = \frac{NPSH}{H_p}} \qquad \boxed{\sigma_t = \frac{TREH}{H_t}} \qquad (3.31)$$

Sigma values for pumps and turbines have been plotted in Figure 3.32 in function of specific speed.
Note that the values of Sigma according to this figure are only valid if the machines are operated at the best efficiency point. Since the PAT should be selected in such a way that it operates slightly in the overload range (see section 3.5.5 above), i.e. operating above bep, the danger of cavitation increases. It is therefore recommended that a sigma value for PATs (in Figure 3.32) close to the pump curve be used. Additionally, the Sigma values of Figure 3.32 are valid for relatively large machines with smooth runner surfaces; small pumps usually tend to cavitate earlier. Sigma values for **small PATs** should therefore be taken at least equal to the values for pumps if not higher.
If NPSH required is directly obtained from the manufacturer's handbook, one must be aware that this value is only valid for the rated pump speed. If the PAT operates at a higher nominal speed, the NPSH required will increase accordingly. In this case better, use the Figure 3.32 below with n_{qp} and relate $NPSH_{req.}$ to the pump-mode head for the new speed. You will find that $NPSH_{req.}$ increases approximately with the square of the speed ratio.

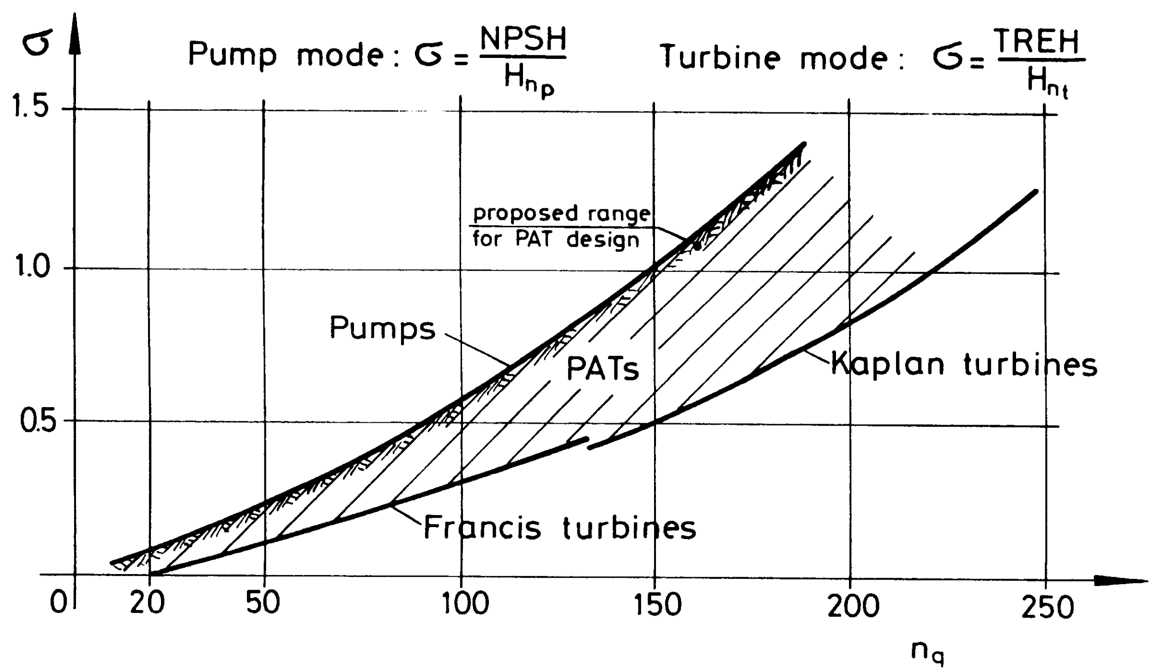


FIGURE 3.32 :
Cavitation for pumps, turbines and PATs expressed by the Thoma number Sigma versus specific speed nq (only valid if operated at bep) (source: R.K. Turton: Principles of Turbomachinery)

3.6.3 What to Do if the Selected PAT does not Yield the Required Performance

Generally, a deviation between actual and designed performance is indicated by a lack (or surplus) of power at the driven machines or generators. It is not very easy to determine the cause of low performance. Before questioning the selection of the PAT, one should first check the penstock (blocked pipes or valves,), the water level in the forebay, the smooth rotation of machinery, transmission pulleys and the PAT itself (stuffing box might be too tight). Furthermore, measurements of the differential head across the PAT and the discharge should be undertaken. Table 3.33 shows the causes of low power generation and proposes some possible remedies which will be further explained afterwards. (Note: to identify the possible cause of faulty operation , head and flow measurements must be undertaken; section 4.5 shows some basic measuring techniques and indicates measuring devices which should be incorporated into the plant design.)

Initial Indication	Indication after flow and head measurements	Cause	Remedy
Power not developed	Both head and flow according to design	PAT operating at low efficiency	a) reduce rotational speed (change pulley diameter if a belt drive is used b) impeller trimming if the available flow is sufficient (Attention: flow will increase)
	Pressure head too low	a) penstock friction higher than assumed b) static head not according to design c) penstock not flowing full	a) modify penstock b) redesign scheme according to new static head c) maintain head by partly closing the control valve
	Flow too small (but river discharge sufficient); head okay	actual PAT performance not as predicted	change PAT
Power delivered too high for machinery or generator	either flow or head (or both) too high	PAT not operating at duty point	dissipate surplus energy by a control valve or a throttling orifice in the penstock

TABLE 3.33 :
PAT performance not according to design: possible causes and remedies

The remedies proposed in Table 3.33 above can be divided into two categories, namely modifications on the PAT itself and alterations of the operating conditions.

- <u>modifying the geometry of the PAT</u> (spiral casing, diffusor and the impeller, mechanical details see section 3.6.4).

  Note that for a given turbine head, the possible shift of the performance curve when trimming the impeller is very limited. Once PAT and head are given, there is little room left for altering operating conditions by trimming impellers; this is different from the PAT selection procedure explained above where PATs with trimmed impellers have been used to widen the application range of a PAT but not the actual operating conditions under a defined head. Figure 3.34 below shows the effect of trimming for two different installations which use the same type and size of PAT. Installation I was not well designed.i.e the actual operating point was very far below the best efficiency point. By trimming the impeller, the power output could be improved tremendously.

  The second plant shows an operating point much closer to the bep of the PAT (at full impeller diameter). The effect of impeller trimming for this plant is very modest and affects mostly flow but not appreciably power output. This is an important fact of impeller trimming: in both cases, flow increases after trimming, hence, the river discharge must be sufficient when attempting to modify the power output by using different impeller cuts.

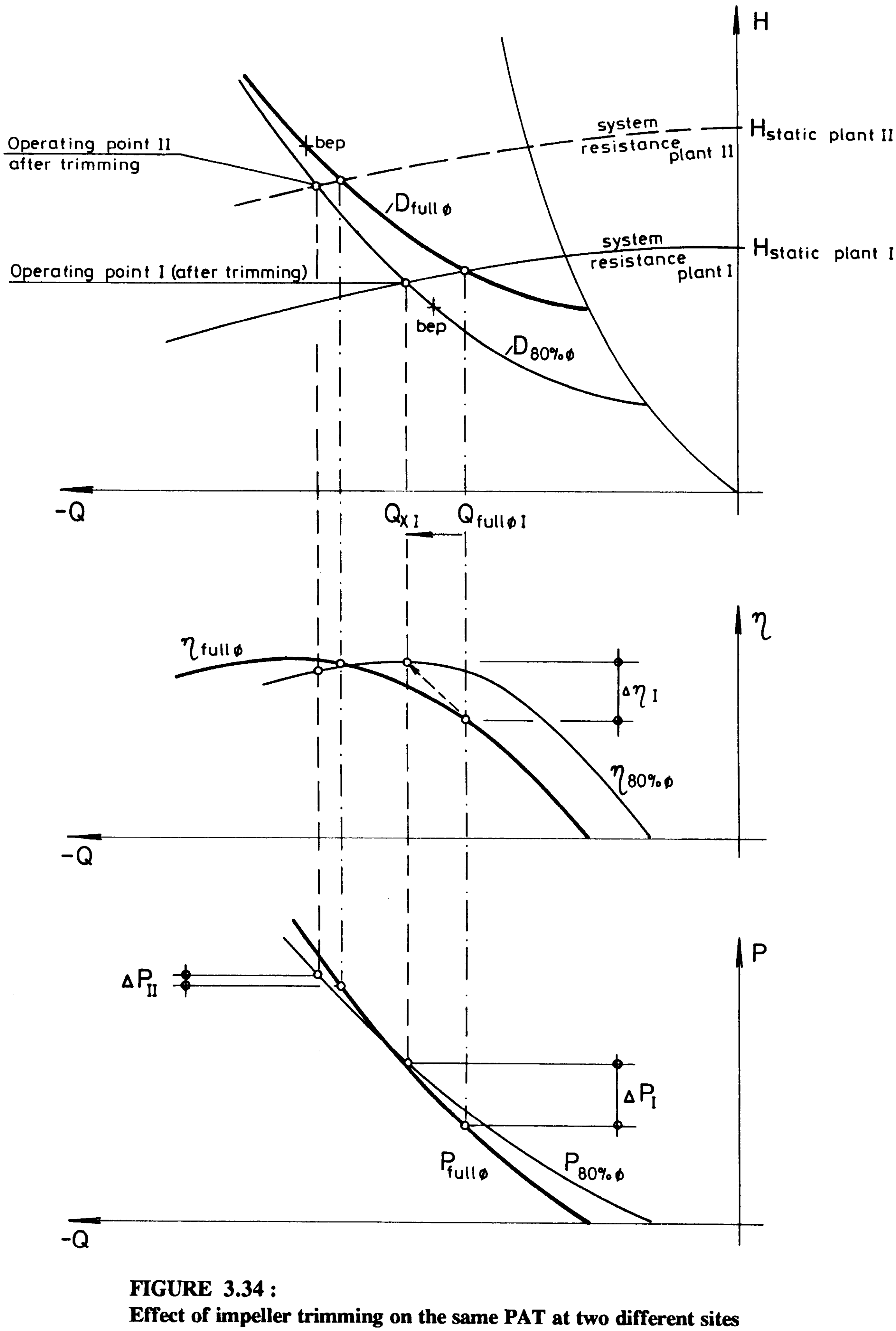


FIGURE 3.34 :
Effect of impeller trimming on the same PAT at two different sites

Generally, modifying the diffusor (cross sectional area of passages and/or angles of fixed vanes) has a more significant effect on the performance in turbine mode than impeller trimming (see also section 3.5.2: the importance of the inlet conditions in turbine mode rather than impeller design). However, such modifications require a profound knowledge of pump and turbine design and should not be undertaken on site. If a PAT is to be adapted to specific site conditions, advice should be sought by the manufacturer of the pump or at a laboratory (university or research centre) which might be prepared to carry out research into the effects of design modifications of a particular PAT.

□ **altering the operating point (OP) by:**

- changing the speed of the PAT.
If a belt drive is used, a change of speed is fairly simple (using different pulley diameters). The OP will change along the system resistance curve (see Figure 3.29 above). If the PAT was selected to operate slightly in the overload range, efficiency will not decrease rapidly. However, the effect of speed changes on the performance in turbine mode is fairly low, comparable to the moderate effect of impeller trimming as shown above.

- reducing the turbine net head with a control valve or an orifice in the penstock line.
This measure can only be applied if flow and output are too high. The effects are similar to increased losses in the penstock, i.e., the resistance curve falls steeper with increasing flow than with an opened valve. This usually means a shift of the operating point away from the best efficiency point and subsequently a rapid drop in power output (see Figure 3.26 above). The situation will not be as severe if the selected PAT operates beyond its best efficiency point (i.e., in the overload range, see section 3.5.5 above).

- it may turn out that the actual flow through the PAT is insufficient (PAT resistance higher or actual head lower than expected). If modifications on the PAT as explained above do not yield the desired increase of flow, there is, unfortunately, no way to increase flow through the PAT. In some cases, however, a certain minimum flow is required by the consumers downstream of the PAT (for irrigation, water supply, industrial process). Introducing a by-pass around the PAT might provide the required flow to downstream users.

3.6.4 Impeller modifications

- Underfiling vane tips

The high pressure side of the machine becomes more important in the turbine mode than in normal pump operation due to the reversed flow. The impeller of a pump is usually machined at its outer periphery leaving a sharp edge at the vane tips (see Figure 3.35 below). While being of no importance in pump mode, this sharp edge may cause vortices and separation of flow in turbine-mode with a subsequent decrease of efficiency. Underfiling these sharp edges or machining them might improve the efficiency of the PAT . Equal efficiencies in pump and turbine mode have been reported as a result of this simple modification.

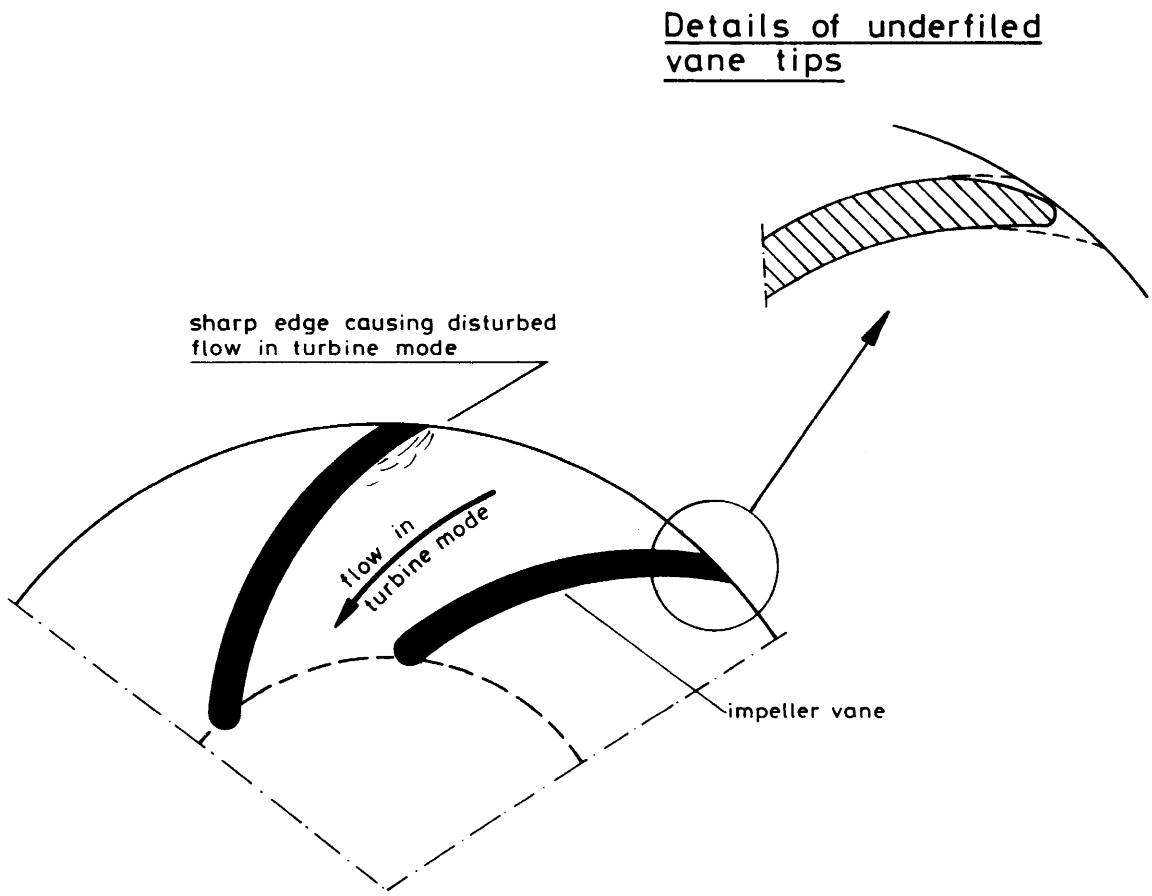


FIGURE 3.35 :
Improvement of efficiency of a PAT by underfiling the impeller vane tips

- Trimming impellers

Trimming the impeller diameter leads in both pump and turbine mode to a simultaneous reduction of the best efficiency head and flow of a machine. Thus, the same machine can be used for a variety of site conditions. Contrary to this, trimming of the impeller at a given installation (at fixed head) is more likely to increase flow than to reduce it !

Trimming of radial flow impellers (nq < 30) mainly results in a reduction of the peripheral speed of the high pressure side of the impeller and a subsequent change of the tangential velocity vector in the velocity vector diagram (see Appendix B).
For machines of specific speed beyond nq = 30 to 40 the whole geometry of the high pressure side is affected (change of impeller width and vane angle, see Figure 3.36 below) and the efficiency will drop. Such adaptations should not be attempted without contacting the manufacturer.

Figure 3.36 shows the details of impeller trimming. In pumps fitted with a diffuser-ring, the gap between the impeller and the diffuser should not become too large; therefore, only the vanes are turned down while the diameter of the shrouds is untouched. For pumps without diffuser, the whole impeller is turned down (preferably on a lathe).

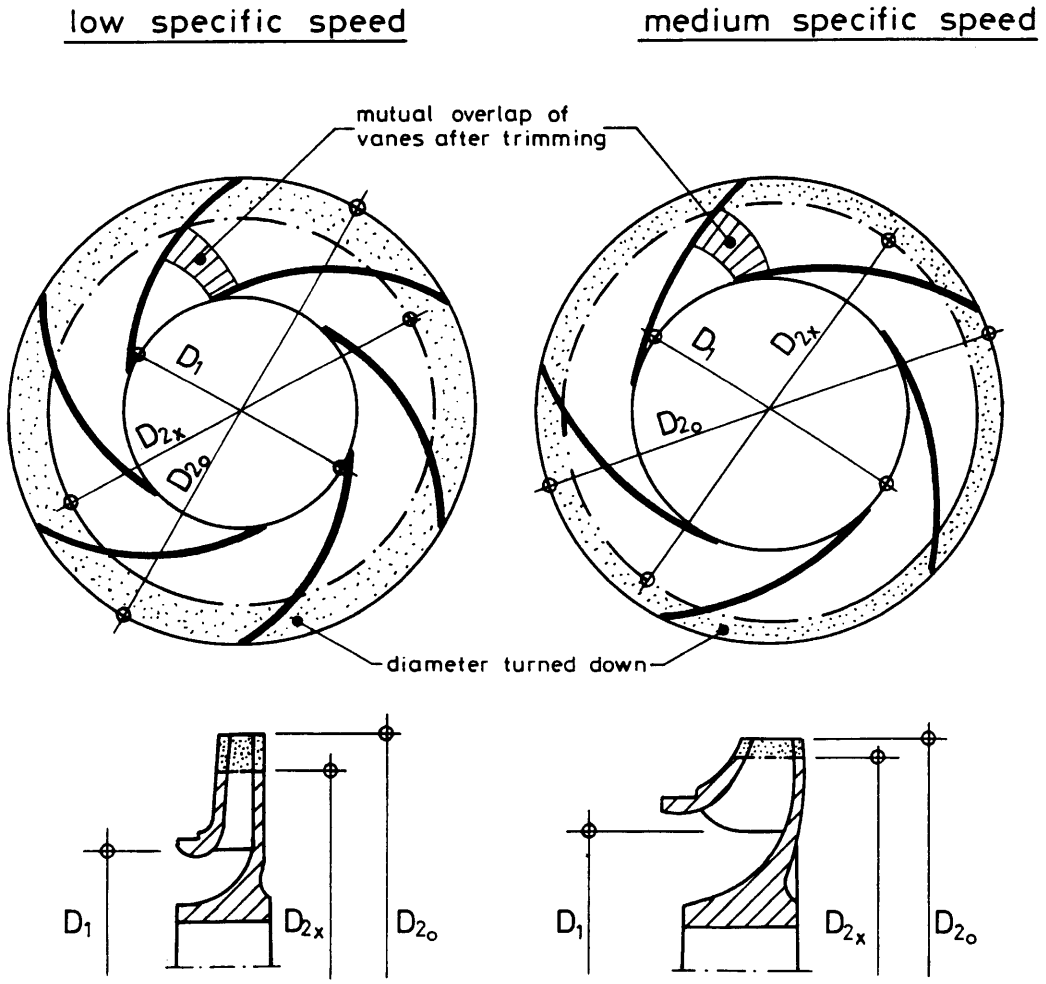


FIGURE 3.36 :
Design details of impeller trimmings

If the diameter reduction is kept within reasonable limits so that a mutual overlap of the vanes still remains (see Figure 3.36 above), the relationship between the impeller diameter, head and flow of the machine <u>in pump mode</u> can be expressed as follows (according to KSB):

$$\boxed{\frac{H_{np}}{H_{np_x}} \approx \frac{Q_{np}}{Q_{np_x}} \approx \left[\frac{D_{2_o}}{D_{2_x}}\right]^2} \qquad (3.32)$$

where H_{np} and Q_{np} are the head and flow at the best efficiency point in pump mode; the subscript "x" indicates the conditions of the trimmed impeller.

In practice, the turbine head H_{ntx} is known and the required impeller cut to shift the best efficiency point close to this turbine head is to be found. A trial and error method is usually applied to solve this problem. Starting with an assumed impeller cut D_{2x} , we can first compute the new head and flow in pump mode using Formula 3.32 . Turbine-mode performance can then be calculated according to section 3.5.4 above applying the conversion factors. The procedure is repeated with varying impeller cuts until the desired turbine head H_{ntx} is obtained.

3.6.5 Draft tube - Diffusor throat

The low pressure side of a pump is not designed to operate as a draft tube in turbine mode. The velocity energy at the diffusor throat may be partially regained if the last section of the outlet pipe to the tailrace is gradually expanded. The kinetic energy of the water is then transformed into pressure and not dissipated in the tail race; the outlet of the draft tube must, of course, always be submerged. The net turbine head, particularly for **low head installations**, may be substantially increased.
The potential for energy recovery may be estimated by calculating the available kinetic energy at the suction flange (using the mean velocity at the suction side diameter D_3, see Figure 3.37 below).

$$\boxed{H_{rec_{max}} = \frac{v_3^2}{2g}} \qquad (3.33)$$

Only a part of this available energy can be recovered with an economically justified draft tube. As a rule of thumb energy recovery in an effective draft tube may be in the range of 5% for high heads (nq = 20) and 50% for low heads (nq = 100) of the available kinetic energy according to Formula 3.33.

Figure 3.37 shows a typical draft tube design which may be used for PATs. For an economic design, Lein (University of Stuttgart) proposes the following diffusor throat velocities:

for high heads
$$\frac{v_{sm}^2}{2g} = 0.005\, H_{nt}$$

for low heads
$$\frac{v_{sm}^2}{2g} = 0.01 \text{ to } 0.03\, H_{nt} \qquad (3.34)$$

Applying the equation of continuity (see Appendix A), the required diffusor throat diameter can be calculated.

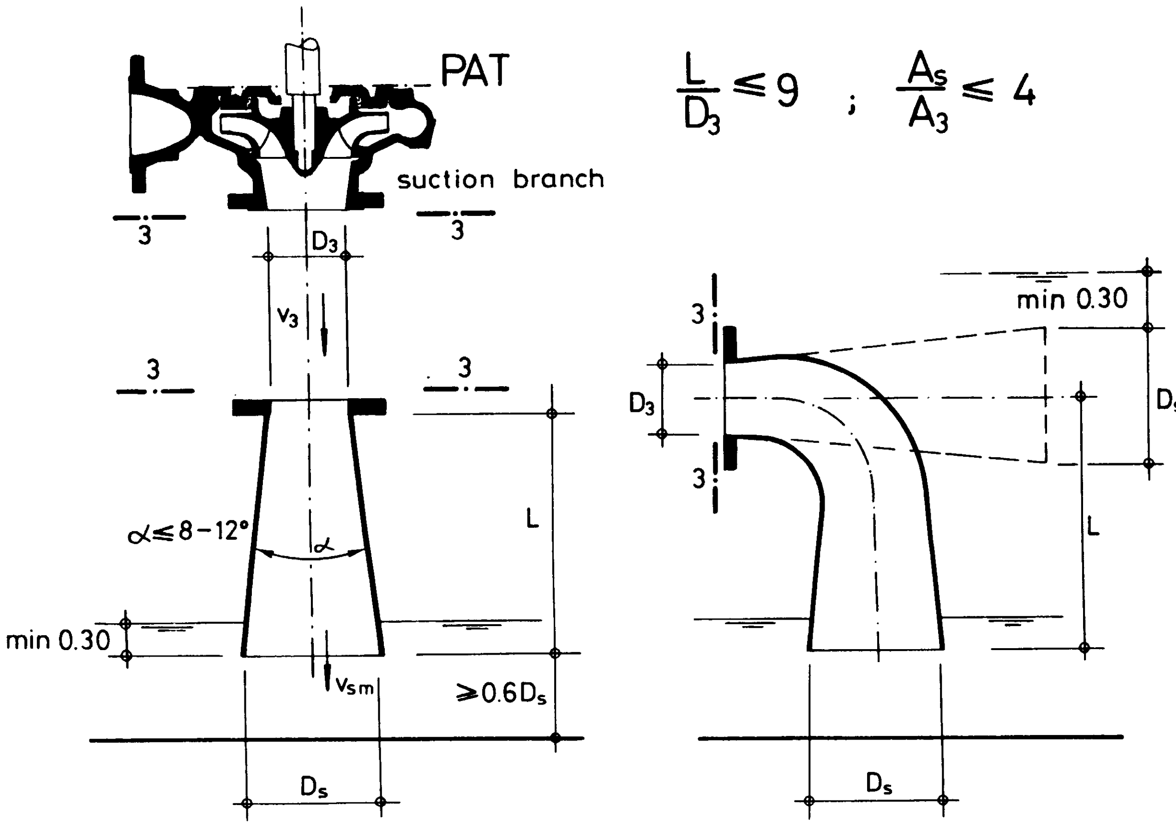


FIGURE 3.37 :
Typical draft tube design and main dimensions (according to Lein)

3.7 Modifications - Checks

3.7.1 General

The following sections deal with the design reviews and checks required for a pump to operate as a turbine. In any case these should be made by the manufacturer who has all the necessary data at his disposal.
Only in case the manufacturer or his representatives cannot be contacted, should the designer attempt to review the design himself using the formulae and diagrams of this manual. The indications might also be useful if the designer must check whether an existing pump (with few data from the manufacturer) could be used as a turbine at a new site.
The following figure gives an overview of the main parts of a standard spiral-casing pump; it is this type of pump which is highly recommended for MHP designs using a cost-reduction approach.

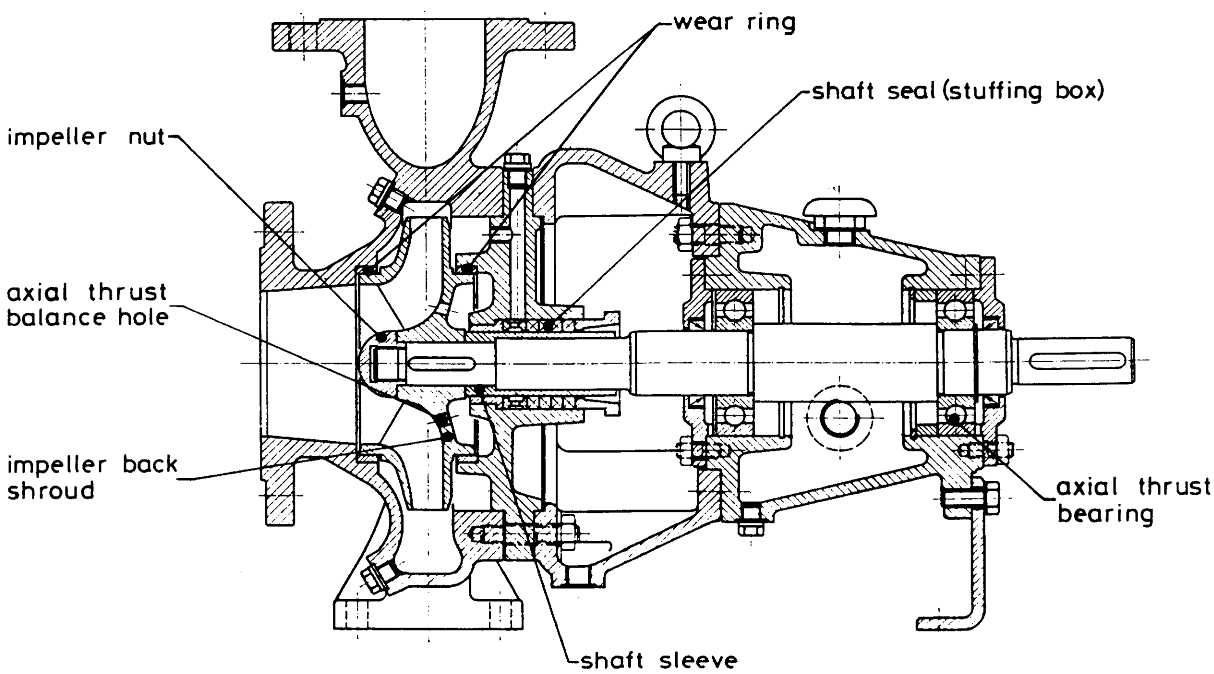


FIGURE 3.38 :
Overview of the main parts of a standard pump (single stage, spiral casing)

3.7.2 Reverse Rotation

It must be checked whether fixation of rotating parts mounted on the shaft could be loosened because of the reversed rotation in turbine mode. Although the sense of the torque on the shaft remains the same in both pump and turbine mode, friction in sealings, stuffing box etc. might still cause inversed torque opening threads. **Shaft sleeves screwed onto the shaft should be protected by set screws.** Virtually all pumps with such designs have set screws as a standard precaution measure, not actually intended for turbine operation, but against wrong connection of the power lines of electric motors. However, these set screws are sometimes not installed by the operator after tightening sleeves and stuffing boxes and damage will almost inevitably occur when the pump / PAT is restarted.
No special measures are required for the impeller nut. A locking plate or better a few set screws will usually be sufficient since reversed torque might only occur if the impeller is temporarily blocked (by debris etc.) and the inertia of the generator / flywheel assembly exerts a reversed torque on the shaft.

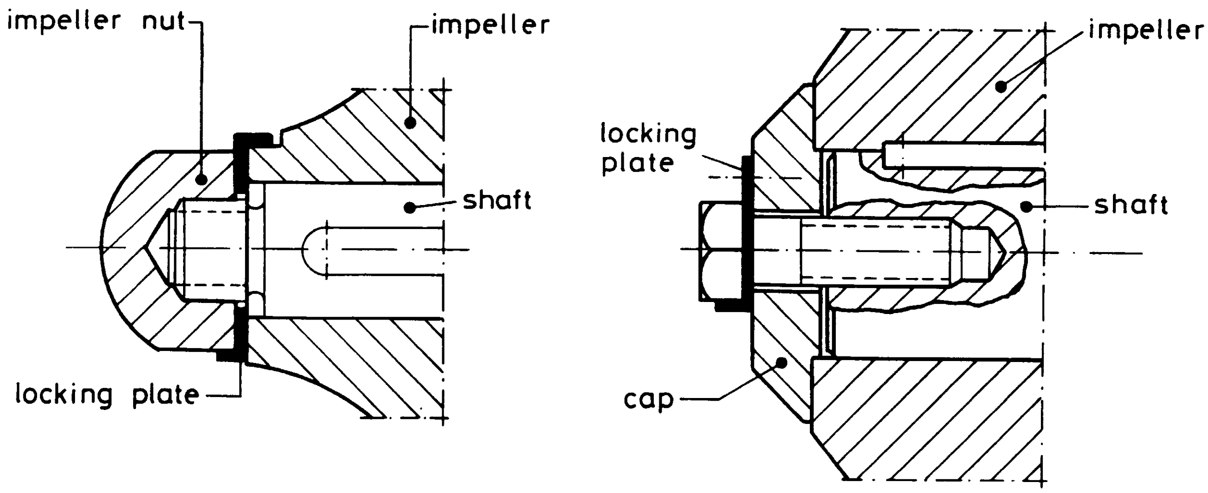


FIGURE 3.39 :
Locking plate on the impeller nut as a protection measure against unscrewing in reverse rotation

3.7.3 Bearing Design

It must be checked, whether the bearing is designed for reversed operation. This is especially important, if the selected PAT is equipped with hydrodynamic bearings, which may be designed for one direction of rotation only. Figure 3.40 below shows the principle of hydro-dynamic bearings (axial and radial bearings). When in doubt whether a specific machine may be operated as a PAT, seek the manufacturer's advice.

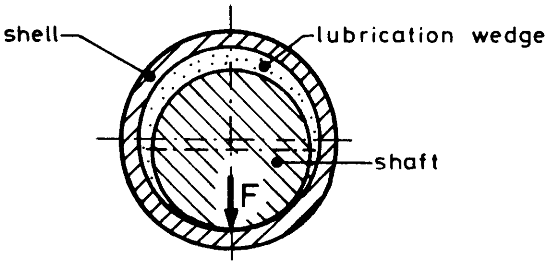
If a belt drive is used to transmit the turbine power, additional load is placed on the bearings; when ordering a PAT, this fact must be mentioned in order that the bearing arrangement will be adjusted to the higher radial forces (see also section 3.7.5 and 3.7.7 below).

Hydro-dynamic bearings

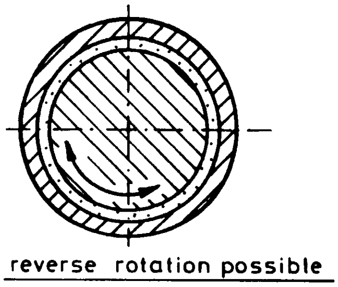
a) Radial type

with circular shell

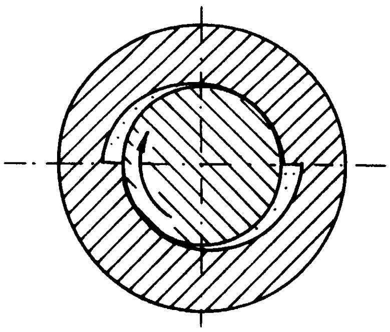
at standstill



during operation

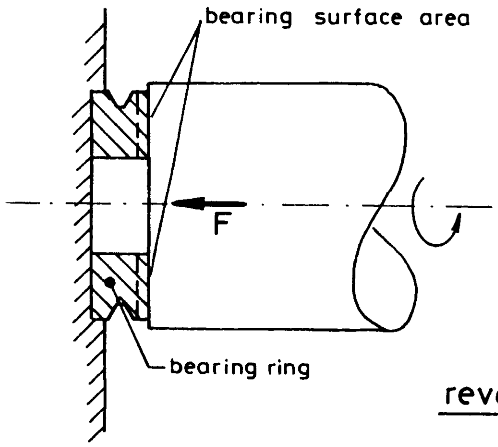


shell with lubrication grooves

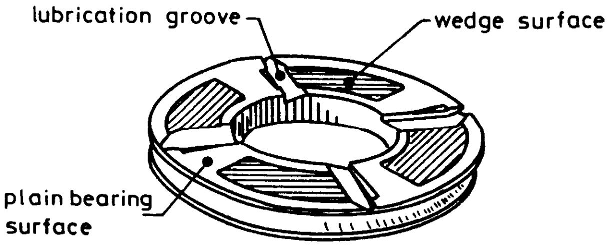


reverse rotation not possible !

b) Axial type



bearing ring



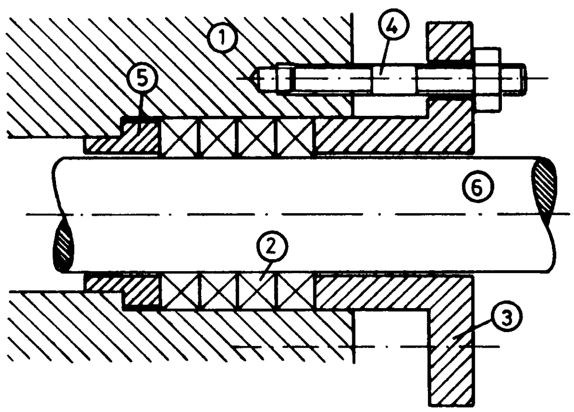
reverse rotation not possible !

FIGURE 3.40 :
Basic designs of hydro-dynamic bearings and their sensitivity to reverse rotation

3.7.4 Shaft Seals

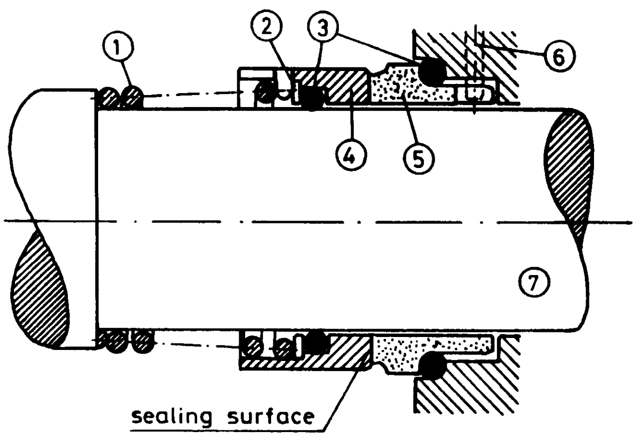
The conventional stuffing box should not cause problems due to the reverse rotation in turbine mode provided it is correctly packed with several gland packing rings.
On the other hand, the designer must be very cautious with modern mechanical seals which might require one specific sense of rotation to operate properly (supply of lubricant depending on sense of rotation, left or right-hand spring installed, etc.). However, mechanical seals are indispensable for PATs running under very high heads (beyond Ht = 200 m where the plain stuffing box without cooling or water seals is not recommended) or if the seal must be absolutely watertight (e.g. for motor protection in submersible pumps).
Figure 3.41 shows a conventional stuffing box arrangement and a mechanical seal.

a) stuffing box



1 stuffing box housing
2 gland packing rings
3 gland
4 stud
5 bottom ring
might be incorporated in housing
6 shaft

b) mechanical seal



1 compression spring
2 compression ring
3 O-rings (static seal)
4 slip ring (rotating)
5 counter ring (stationary)
6 pin
7 shaft

FIGURE 3.41 :
Basic options of shaft seals: stuffing box arrangement and mechanical seal

3.7.5 Belt Drives

Standard pumps are usually designed to be driven by direct-coupled prime movers (electric motors, diesel engines). In many MHPs however, standard pumps used as turbines cannot be directly coupled to the driven machinery or generator because the speeds do not match. The generator speed, for example, is given by the network frequency and can only take a distinct value (according to the number of poles), while the PAT nominal speed must be chosen in such a way that the available machine operates near its best efficiency point under the given site conditions. A transmission gearing (e.g. one or several V-belts or flat belts) is therefore required in many cases. Flat belts require a relatively high tension to avoid slippage; this places additional load on the shaft, the bearing and the mounting frame (see paragraph 3.7.7 b below).

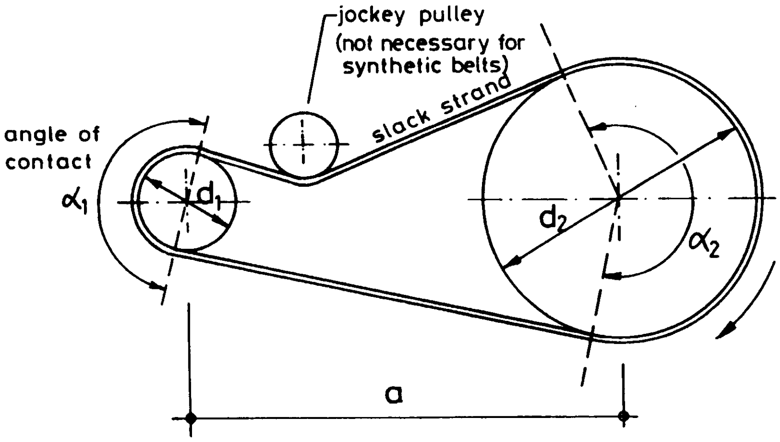


FIGURE 3.42 :
Basic layout of a belt drive and nomenclature

For a preliminary design (for power-house layout) or if no expert advice from belt manufacturers is available, use the following formulae and the diagrams of Figure 3.43 to determine the belt drive layout:

Relationship between pulley diameter and speed:

$$\boxed{\frac{d_t}{d_g} = \frac{n_g}{n_t}} \tag{3.35}$$

(indices: t = turbine, g = generator or machinery)

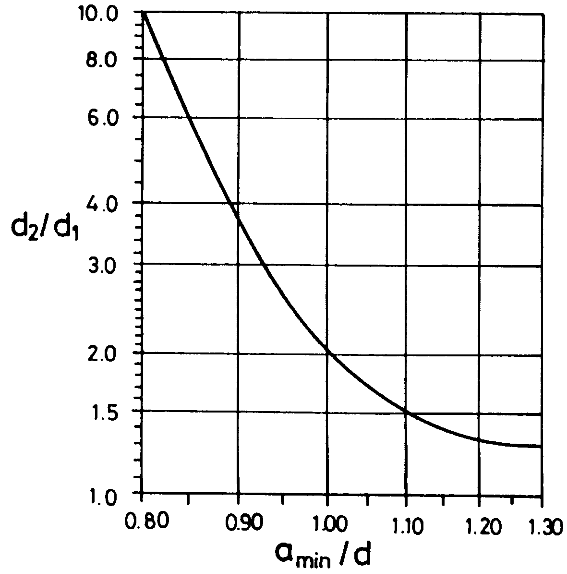
Traditionally, flat belts are made of leather which is still used today in many agro-processing plants, especially in developing countries. Modern synthetic belts are advantageous in terms of power transmitted and distance required between pulleys.

The distance between the pulley centres for leather belts should not be smaller than (a and d in meter):

$$\boxed{\text{for leather belts: } a_{min} = 2 + (d_1 + d_2)}$$

$$\boxed{\text{valid for speed ratios } n_1 / n_2 \text{ below } 1/5} \tag{3.36}$$

For synthetic belts use the following figure:



In any case, "a" should be as big as possible.

Figure 3.43 provides diagrams indicating the maximum power to be transmitted by leather and synthetic flat belts of 100 mm belt width and varying thickness. The indicated power P_0 must be reduced according to the operating conditions and the drive layout.

$$\boxed{P_{belt} = C\ P_0 \qquad \text{where } C = C_1\ C_2\ C_3\ C_4} \qquad (3.37)$$

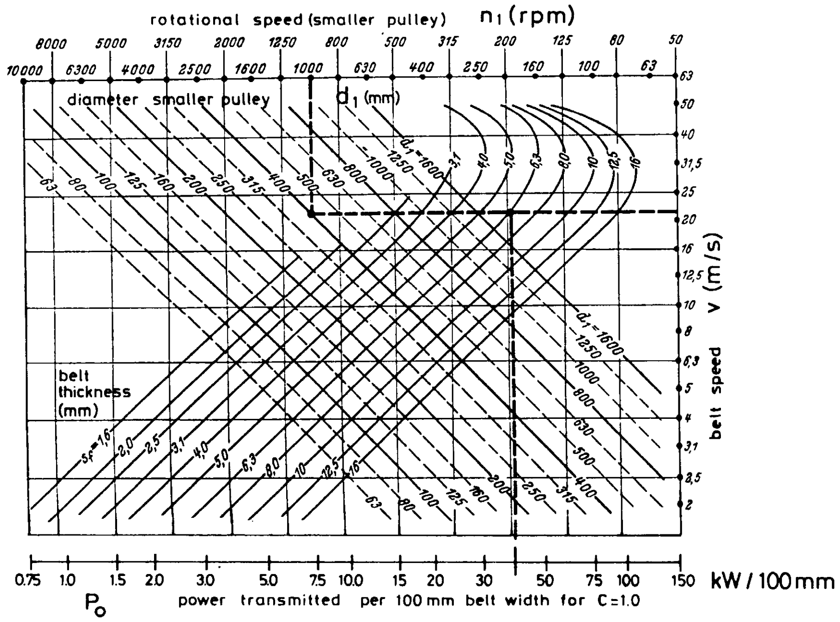
		Factor
Load Factor C_1 (non-uniform operation)	a) smooth-operating machines b) centrifugal pumps, fans c) drills, flour mills, lathe d) oil expeller, rolling mill, loom, sawmill, stone crusher	1.0 - 1.1 1.1 - 1.2 1.25 - 1.35 1.5 - 2.0
Environmental conditions C_2	a) dry air, normal temperature b) humid, dusty air, large temperature variations c) oil splashes on belts d) wet or very large temperature differences	1.0 1.1 1.25 1.3

Belt Lifetime C_3	**ratio of bending frequency B / B_{max}**							
	Bending frequency $B = v\ z / L$ v = belt speed (m/s), z = number of pulleys, L = belt length (m), leather belts: $B_{max} = 5 - 25\ s^{-1}$ synthetic belts: $B_{max} = 80\ s^{-1}$							
operating hours per day (h)	0.16	0.24	0.32	0.40	0.48	0.60	0.80	1.00
3 - 4	0.95	1.00	1.03	1.06	1.11	1.16	1.28	1.45
8 - 10	1.00	1.02	1.05	1.09	1.14	1.19	1.33	1.54
16 - 18	1.03	1.07	1.11	1.18	1.25	1.33	1.54	1.89
24	1.07	1.14	1.22	1.32	1.43	1.56	1.93	2.38

Angle of Contact α_1	100°	120°	140°	160°	180°	200°	220°
Factor C_4	1.33	1.21	1.12	1.05	1.0	0.96	0.935

Use diagrams 3.43 as follows: using the rotational speed n_1 of the smaller pulley and an assumed diameter of this pulley gives point (a) an the diagram (Note the belt speed should be around 10 - 30 m/s). As a first estimate, try a belt of 100 mm width to transmit the power P_o (= rated power of the smaller pulley P_{n1} / C). The intersection with the horizontal line from point (a) provides the required belt at point (b). Calculate the diameter d_2 of the larger pulley using the relationship given above. If no convenient solution results, vary the width of the belt and/or the pulley diameter d_1. The graph for leather belts also indicates the required belt thickness.

Leather flat belts



Synthetic flat belts

(Habasit, Siegling, Chiorino, etc.)

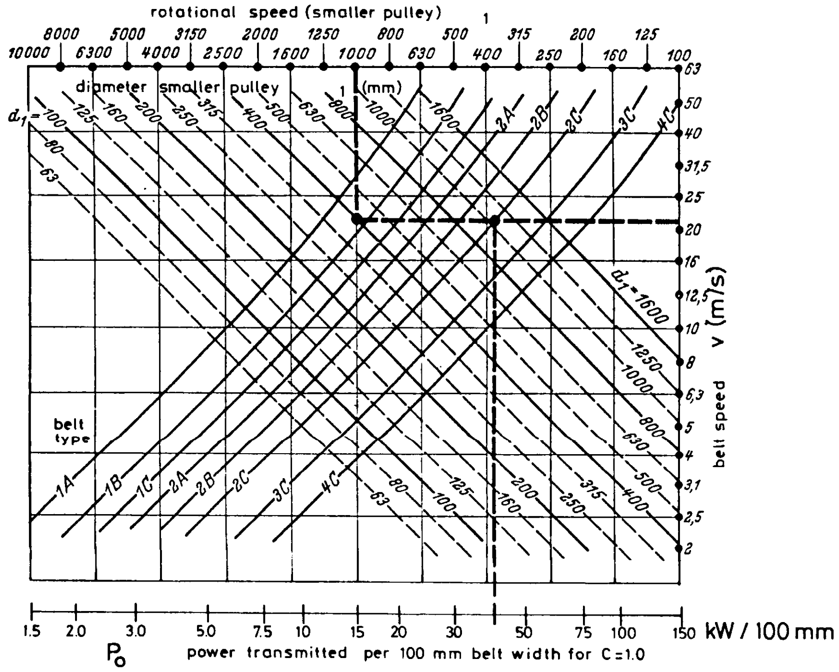


FIGURE 3.43 :
Power transmitted by flat belts (source: Niemann; Maschinenelemente)

3.7.6 Axial Thrust

Single-suction impellers (Fig. 3.44) are subjected to an axial thrust resulting from different pressures acting on the impeller shroud. Theoretically, the back shroud is on its front under the suction pressure p_1 and at its rear under the high pressure of the turbine head p_2. The axial thrust is usually carried by thrust bearings (see also next section).

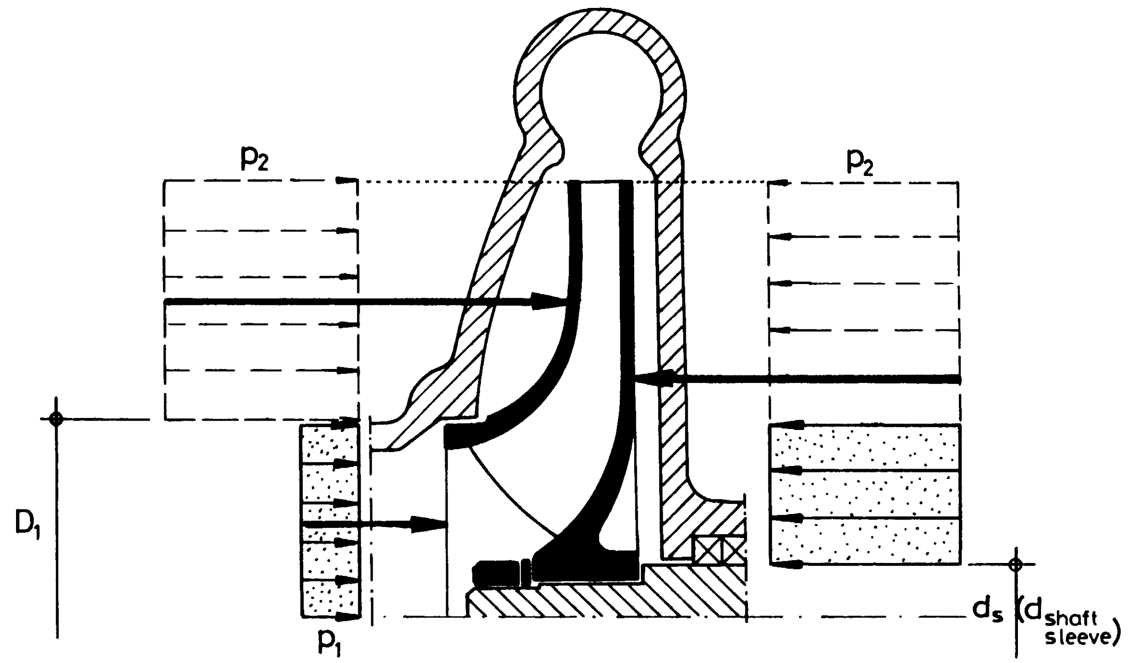


FIGURE 3.44 :
Axial thrust on a single-suction impeller

The magnitude of the axial thrust can be calculated as follows:

$$\boxed{F_{axial} = \frac{\pi}{4}(D_1^2 - d_s^2)(p_2 - p_1)} \qquad (3.38)$$

$(p_2 - p_1)$ is less than the total head because the liquid behind the impeller is in rotation.
In modern pump designs, axial thrust on bearings is reduced by either of two methods:

- a chamber on the back of the impeller is provided with a closely fitted set of wear rings and suction pressure is admitted to this chamber by drilled holes through the impeller back shroud (Fig. 3.44 a)

- radial ribs or vanes on the back shroud reduce the pressure in the space between the impeller and the pump casing because the liquid in that space rotates almost at impeller angular speed (principle of conservation of energy ; Bernoulli equation) (Fig. 3.44 b).

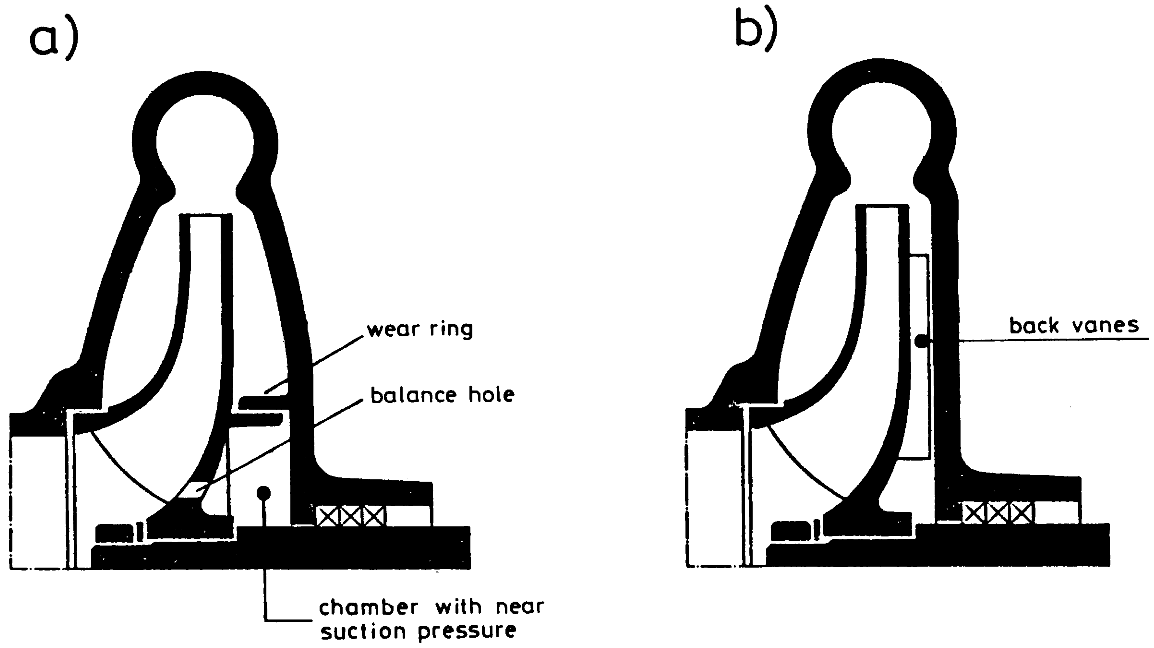


FIGURE 3.45 :
Balancing axial thrust by a) drilled holes through the back shroud of the impeller and b) radial ribs on the back shroud

Compensation of axial forces is usually not complete and axial bearings can be found in most pump designs (despite compensating devices) to carry a residual axial thrust. Balancing axial thrust becomes more important for multistage pumps because of the higher pressures involved and the combined thrust of several stages.

It has been found that axial thrusts **in turbine mode** are lower than in pump mode at similar operating conditions (according to YANG). Therefore, axial bearings of a PAT might not be overloaded despite the higher heads in turbine mode. When using a belt drive, the same bearing carrying the axial thrust will most probably be loaded by the additional radial forces of the belt tension. When reviewing the capacity of that bearing the axial thrust must be considered (see next section).

3.7.7 Shaft Resistance, Bearing Load and Critical Speed

a) Shaft Resistance and Bearing Load

A pump shaft is designed to transmit the power from the prime mover to the impeller. Power depends on the torque and the rotational speed ($P = T * \omega$) where torque constitutes the main parameter for shaft resistance. The nominal torque and thus nominal shaft resistance can therefore be calculated for the **pump mode** with:

$$\boxed{T_{np} = \frac{P_{np}}{\omega_{np}}} \quad \text{with} \quad \boxed{\omega_{np} = \frac{2\,\pi\, n_{np}}{60}} \qquad (3.39)$$

(n_{np} = nominal pump speed in rpm and P_n = nominal pump power; use the maximum permissible values for the pump model in question as indicated by the manufacturer's brochure; e.g 2900 - 3600 rpm even if the rated turbine speed will not be as high)

If the pump is operated as a turbine, the head and subsequently torque and power developed are generally higher (at the same nominal speed) than in the pump mode (see section 3.5.2, last paragraph). The shaft resistance should therefore be checked.

Calculate the nominal torque in turbine mode from the shaft power $P_{nt} = \rho * g * Q_{nt} * H_{nt} * \eta_{nt}$ and the assumed nominal turbine speed (nt) for the application which should generally be lower than the maximum permissible pump speed n_{pmax} (allowance for runaway speed in turbine mode):

$$\boxed{T_{nt} = \frac{P_{nt}}{\omega_{nt}} = \frac{\rho\ g\ \eta_{nt}\ Q_{nt}\ H_{nt}}{\omega_{nt}}} \quad \text{with} \quad \boxed{\omega_{nt} = \frac{2\,\pi\,n_{nt}}{60}}$$

(3.40)

If $T_{nt} < T_{np}$, the use of the specific pump as a turbine is unlikely to cause damage to the shaft or other internal damage due to insufficient shaft resistance.

If the torque in turbine mode is higher than the nominal torque in pump mode, it does not necessarily mean that the shaft would fail in turbine operation. Manufacturers often use a standard shaft design for a complete series of pumps which implies an overdesigned shaft and bearing assembly for the machines at the lower end of the series.
If the manufacturer cannot be contacted for further advice, the following formulae can be used to estimate the minimum shaft diameter and the bearing load of the pump in question. The shaft of a single stage pump represents a simple supported beam with a cantilever (see Figure 3.46).

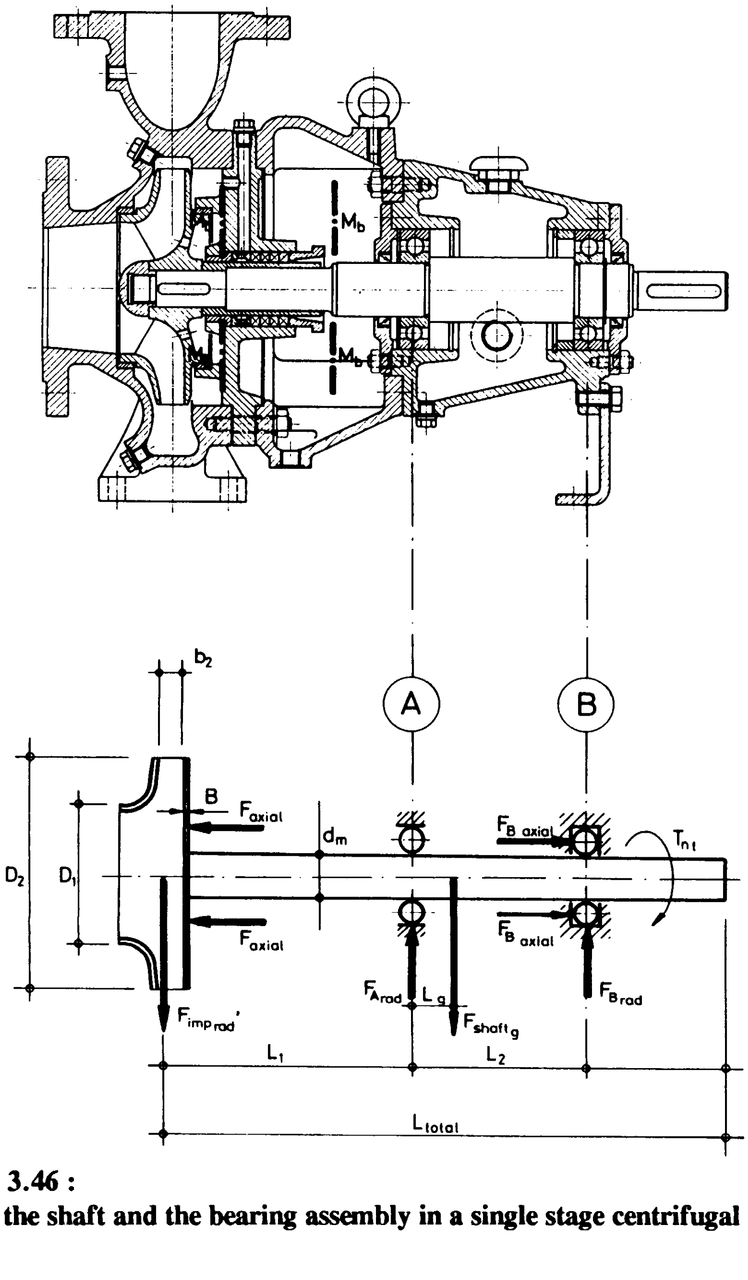


FIGURE 3.46 :
Loads on the shaft and the bearing assembly in a single stage centrifugal pump

Forces acting on the shaft are the following (note that it is usually sufficient to use rough estimates for the impeller and shaft masses because they do not contribute much to the shaft loads):

$$\boxed{F_{impeller_{radial}}' = F_{impeller_{rad}} + F_{impeller_g}} \qquad (3.41)$$

where $F_{impeller_{rad}} = 0.08\, g\, \rho_{water}\, D_2\, b_2\, H_{nt}$

$F_{impeller_g} = \frac{\pi}{4} D_2^{\,2}\ 3\, B\, g\, \rho_{steel}$ (assuming an equivalent thickness of the total impeller material of 3 times the back shroud thickness)

$F_{shaft_g} = \frac{\pi}{4} d_m^{\,2}\, L_{tot}\, g\, \rho_{steel}$

$$\boxed{F_{axial} = \frac{\pi}{4}(D_1^{\,2} - d_s^{\,2})(p_2 - p_1) = \frac{\pi}{4}(D_1^{\,2} - d_s^{\,2})\, g\, \rho_{water}\, H_{nt}} \quad (3.42)$$

(see section 3.7.6 above) Assuming $(p_2 - p_1)$ equal to the net turbine head: $H_{nt} * \rho_{water} * g$

The support reaction force acting on the bearing B is found by taking all moments about support A into equilibrium:

$$\boxed{F_{B_{radial}} = \frac{F_{impeller_{radial}}'\, L_1 - F_{shaft_g}\, L_g}{L_2}} \qquad (3.43)$$

From equilibrium in radial direction we obtain:

$$\boxed{F_{A\ radial} = \sum_0^i F_{i\ radial} - F_B}$$

From equilibrium in axial direction:

$$\boxed{F_{B_{axial}} = F_{axial}} \qquad (3.44)$$

(note that the axial thrust bearing could also be at bearing A, check the design of the selected PAT)

Bending moment (calculate the bending moment for each discontinuity of the shaft; see Figure 3.46 above)

$$\boxed{M_b = F_{impeller_{radial}}'\, L_x} \qquad 0 < L_x < L_1 \qquad (3.45)$$

Maximum torque (producing torsion in the shaft; see above)

$$\boxed{T_{nt} = \frac{P_{nt}}{\omega}}$$

allowance for transient conditions such as starting the PAT (see Section 4.4.4):

$$\boxed{T_{max\,t} = 1.5 \ldots\ldots 2\, T_{nt}} \qquad (3.46)$$

allowance for transient conditions such as starting the PAT (see Section 4.4.4):

$$\boxed{T_{max\,t} = 1.5 \ldots\ldots 2\, T_{nt}} \tag{3.46}$$

According to Niemann, the reference moment from the combined load (bending and torsion) writes as follows:
(to be calculated for each discontinuity along the shaft with the respective bending moments)

$$\boxed{M_r = \sqrt{M_b^2 + \left(0.6 * T_{max\,t}\right)^2}} \tag{3.47}$$

the minimum shaft diameter then becomes:

$$\boxed{d_{min} = 2.17 \sqrt[3]{\frac{M_r}{\sigma_{perm}}}} \tag{3.48}$$

where M_r = reference moment in Nmm

σ_{perm} = 60 N/mm^2 = permissible stress

(this value allows for a stress concentration at the shaft discontinuities)

The existing diameter should be larger than the calculated minimum diameter of the selected PAT at each section. If this condition is not fulfilled, the pump should not be used as a turbine unless the manufacturer guarantees the shaft resistance for the specific application.

Shaft key

The maximum torque must be transmitted from the impeller onto the shaft by means of the **shaft key.** Figure 3.47 shows shaft key and spline dimensions commonly used (according DIN). Knowing the actual dimensions, the permissible torque T is calculated as follows:
- for shaft keys:

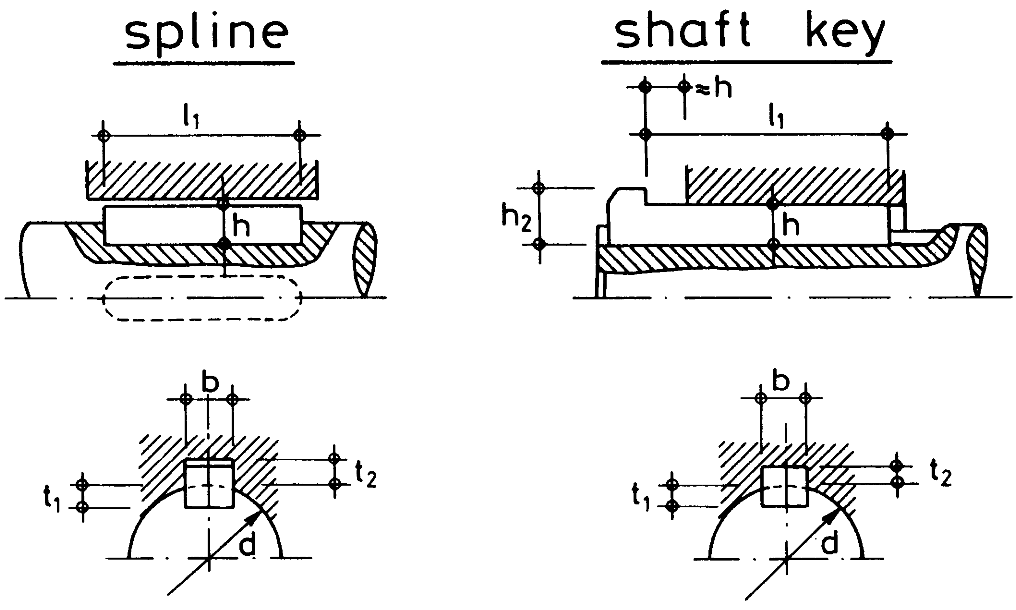
$$\boxed{T_{key} = (h - t_1) (l_1 - h) \frac{d}{2} p_{perm}} \tag{3.49}$$

- for splines (with rounded ends):

$$\boxed{T_{spline} = (h - t_1) (l_1 - b) \frac{d}{2} p_{perm}} \tag{3.50}$$

with p_{perm} = permissible contact pressure of the impeller material

p_{perm} = 50 N/mm^2 for cast iron impeller and $(l_1\text{-}b)/ d_W$ = 1.6.....2.1



shaft diameter d	b	h	t_1	t_2 key	t_2 spline	h_2 key
10 - 12	4	4	2.5	1.2	1.8	7
12 - 17	5	5	3.0	1.7	2.3	8
17 - 22	6	6	3.5	2.1	2.8	10
22 - 30	8	7	4.0	2.4	3.3	11
30 - 38	10	8	5.0	2.4	3.3	12
38 - 44	12	8	5.0	2.4	3.3	12
44 - 50	14	9	5.5	2.9	3.8	14
50 - 58	16	10	6.0	3.4	4.3	16
58 - 65	18	11	7.0	3.4	4.4	18
65 - 75	20	12	7.5	3.9	4.9	20
75 - 85	22	14	9.0	4.4	5.4	22
85 - 95	25	14	9.0	4.4	5.4	22
95 - 110	28	16	10.0	5.4	6.4	25
110 - 130	32	18	11.0	6.4	7.4	28
130 - 150	36	20	12.0	7.1	8.4	32
150 - 170	40	22	13.0	8.1	9.4	36
170 - 200	45	25	15.0	9.1	10.4	40

FIGURE 3.47 :
Dimensions of commonly used shaft key designs /Niemann/

b) Additional shaft and bearing stress due to a belt drive

Belt drives, especially flat belts must operate under relatively high tension to avoid slippage. This results in a an additional load on the shaft and the bearings. When checking the shaft resistance this additional load must be considered.

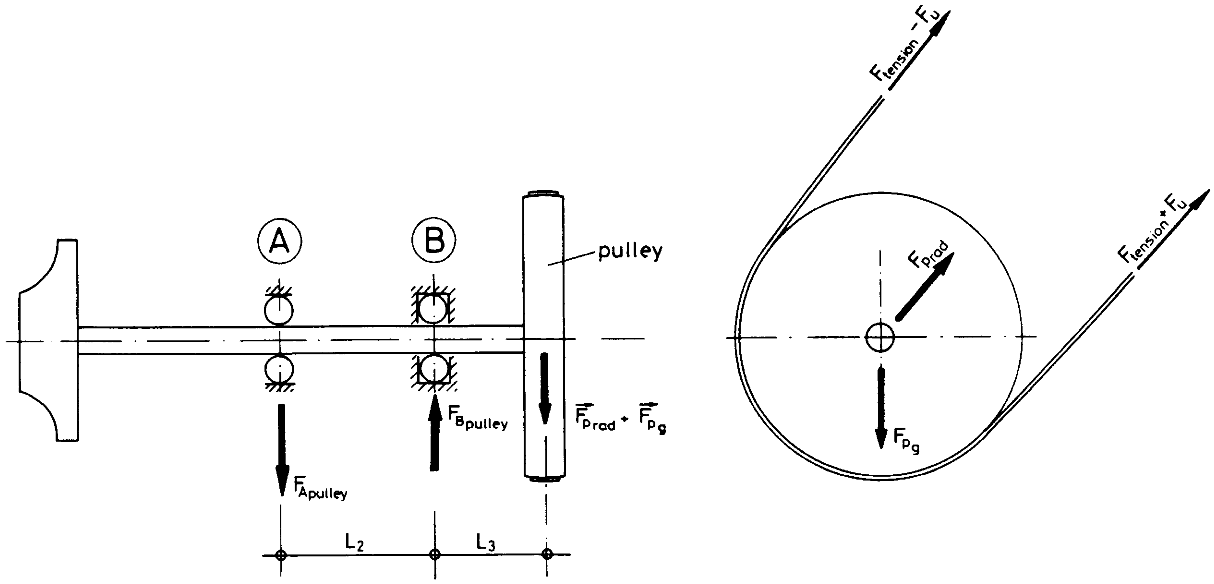


FIGURE 3.48 :
Overhanging load on the turbine shaft due to the use of a belt drive

Calculate the radial force acting on the turbine pulley due to the belt tension:

$$\boxed{F_{P\,rad} = k\ F_u} \quad \text{where} \quad \boxed{F_u = \frac{2\,T_{nt}}{d_1} = \frac{60\,P_{nt}}{\pi\,n_1 d_1}} \qquad (3.51)$$

$k = 4$ for flat belts
$k = 2$ .....2.5 for V- belts

d_1 = diameter of small pulley ; n_1 = rotational speed of small pulley

Add the weight of the pulley to $F_{p\,rad}$ (either vectorially or mathematically especially if the layout of the belt drive is not yet known)

$$\boxed{F_{P\,radial} = F_{P\,rad} + F_{g\,pulley}} \qquad (3.52)$$

This additional radial load $F_{p\,radial}$ from the pulley must be carried by the bearings A and B:

$$\boxed{F_{A radial\ total} = F_{A\ radial} + \frac{L_3}{L_2}\,F_{P\ radial}} \qquad (3.53)$$

$$\boxed{F_{B\,radial\ total} = F_{B\ radial} + \frac{L_2 + L_3}{L_2} F_{p\ radial}} \qquad (3.54)$$

maximum bending moment at bearing B: $M_b = F_{p\ radial} * L_3$

To check the shaft diameter at bearing B use the same equations as above for the impeller side.

Note that a similar additional load due to the belt drive acts on the generator or machine pulley. Shaft, bearing and mounting frame resistance should be checked in the same manner as for the turbine shaft.

c) Permissible bearing loads

The bearing loads calculated above ($F_{rad\ total}$, F_{axial}) must be compared with the permissible load of the bearings installed in the selected PAT (the calculation procedure is similar for both bearings A and B). Calculate the ratio between the axial force and the radial force on a particular bearing:

For bearing B: $F_{Baxial} / F_{B\ radial\ total}$

Check which type of bearing is installed in the selected PAT ; refer to Table 3. 49 below and enter into the corresponding line (for bearings not included in the list, general indications of load factors cannot be given as these depend on the specific bearing design; refer to manuals of bearing manufacturers)

Compare the ratio of $F_{Baxial} / F_{B\ radial\ total}$ with the value **e** according to the table; this will lead you either to the column on the left (if $F_{axial} / F_{radial\ total} > e$) or to the column on the right (if $F_{axial} / F_{radial\ total} < e$). Read the values for X and Y and calculate the reference load acting on the bearing:

$$\boxed{F_{Breference} = F = X\ F_{Bradial\ total} + Y\ F_{Baxial}} \qquad (3.55)$$

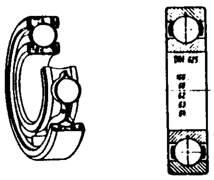
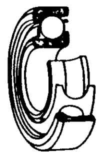
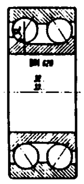
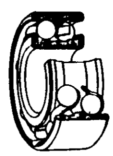
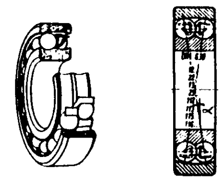
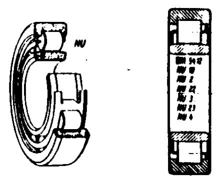
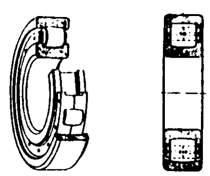
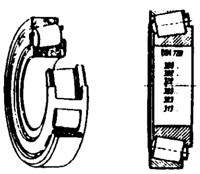
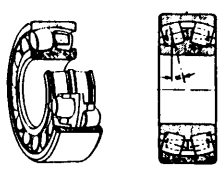
Type of bearing		e	Single row bearing				Double row bearing			
			Fa/Fr > e		Fa/Fr < e		Fa/Fr > e		Fa/Fr < e	
			X	Y	X	Y	X	Y	X	Y
Grooved ball bearing (Series 160, 60, 62, 63, 64)		0.27	0.56	1.6	1	0	same as single row bearings			
Ball bearing with oblique contact (Series 72 B, 73 B)	angle of contact: 40°  	1.14	0.35	0,57	1	0				
Ball bearing with oblique contact (Series 32, 33)	angle of contact: 32°  	0.86					0.62	1.17	1	0.73
Self-aligning ball bearings (Series 12, 23)		1.5 tanα					0.65	0.65 cotα	1	0.42 cotα
Roller bearings (Series NU10, NU49, NU2, NU22, NU3, NU4, NU23)						not designed for combined loads, radial loads only				
Barrel-shaped roller bearing (Series 202, 204)					1	9.5				
Tapered roller bearing (Series 302, 322, 332)		1.5 tanα	0.4	0.4 cotα	1	0				
Self-aligning roller bearing (Series 230, 231, 232)		1.5 tanα					0.67	0.67 cotα	1	0.45 cotα

FIGURE 3. 49 :
Load factors for bearings (source: G. Niemann: Maschinenelemente, Springer-Verlag)

Determine the dynamic load number C from Figure 3.50 according to the bearing type installed in the selected PAT (Note that the values given in Figure 3.50 are minimum values which are guaranteed by most types of bearings; if brochures from bearing manufacturers of your specific bearing type are available, use the values of C (in N) indicated there).

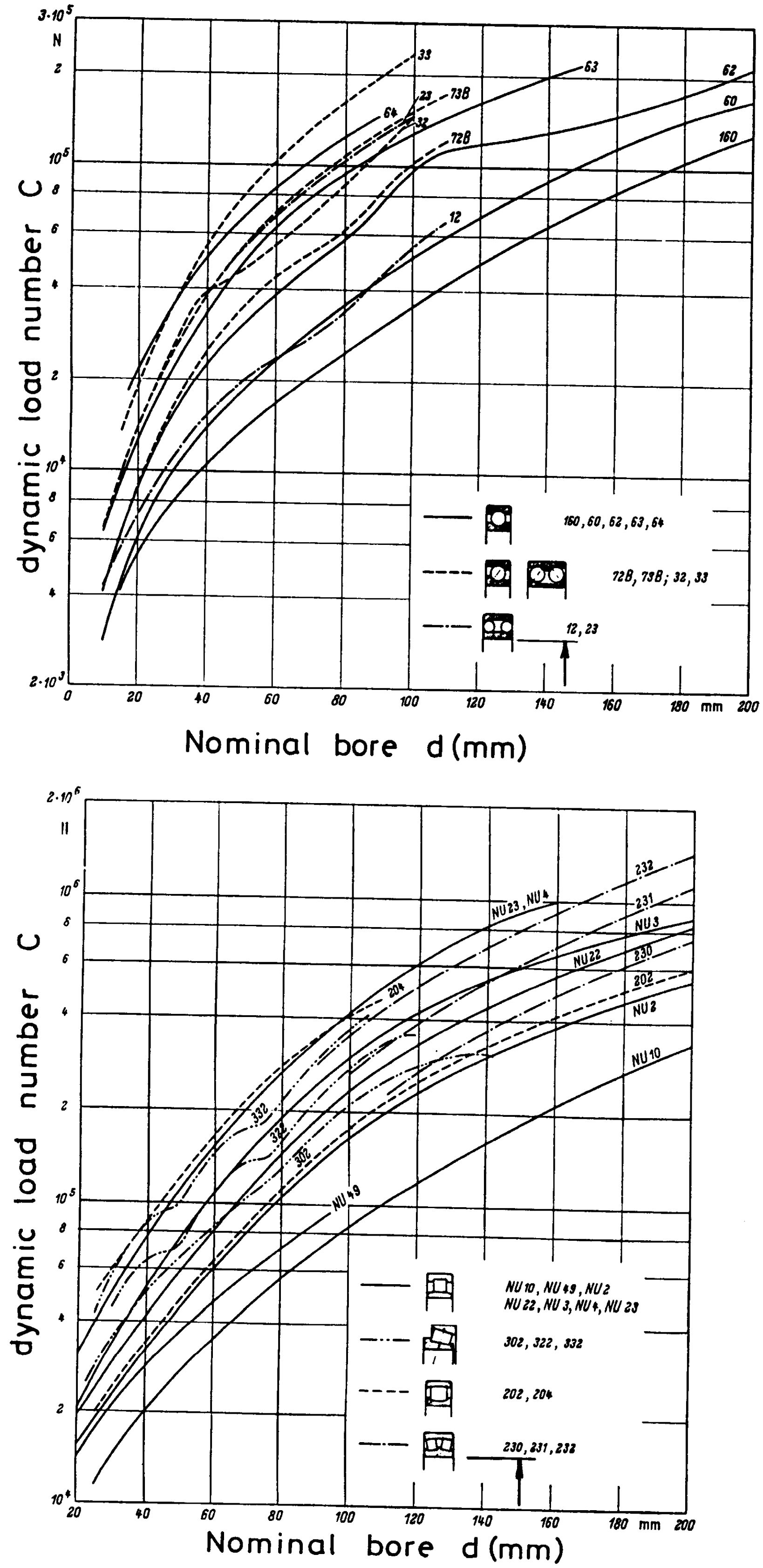


FIGURE 3.50 :
Dynamic load number C in Newton (N) for various types of bearings (minimum values according to DIN) (source: G.Niemann, Maschinenelemente, Springer-Verlag)

Enter with this dynamic load number C into the diagram 3.51 (abscissa) and find the intersection point with the reference bearing load F. Move horizontally to the left as far as the intersection with the turbine nominal speed n_{nt}. The hours corresponding to that point indicate the lifetime of your bearing.

If this lifetime is too short, replace the bearing in your selected PAT with a bearing type of a higher dynamic load number. It should be checked whether a stronger bearing fits into the bearing housing (For PATs, a bearing life of about 50 000 hours should be aimed at).

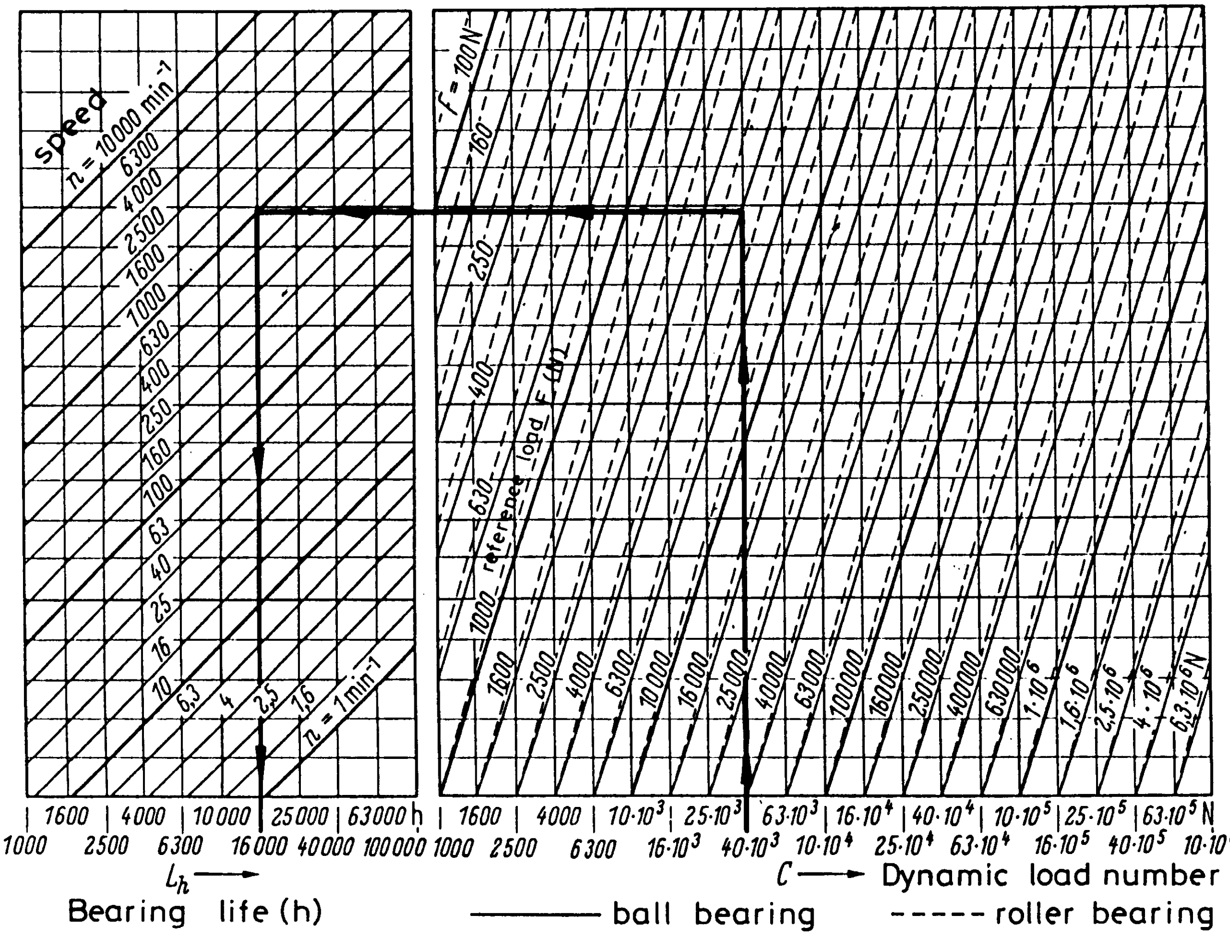


FIGURE 3.51 :

Bearing life in function of dynamic load number C, reference load F and turbine rotational speed n_{nt} (source: Tochtermann/Bodenstein, Konstruktionselemente des Maschinenbaues, Springer-Verlag)

d) Critical speed

The critical speed is defined as the speed at which the shaft of the pump or PAT operates under increased vibration and noise resulting from dynamic deflection of the shaft.

To visualize the concept of critical speed, the following simplified experiment is considered. The shaft of a pump (supported by the bearings) and its main masses (impellers) represents an elastic element which can be set into swinging or oscillating If the shaft (at standstill) is set into vibrating movement by a single stroke (see Figure 3.52a), it will continue to vibrate at well defined frequency, the so-called natural period of vibration or resonance frequency.

The rotating shaft will also vibrate due to unbalances of the rotating elements (shaft and impeller). Even with carefully balanced elements, there is always a residual unbalance which develops a centrifugal force causing increased deflection of the shaft. If the frequency of this deflection (i.e., the frequency or speed of rotation) corresponds to the resonance frequency of the shaft, the deflection/vibration will be continuously amplified [1)] until some internal parts are damaged or the shaft fails. This speed is called the critical speed. In practice, deflection is limited by the damping effect of the liquid and the stuffing box. Figure 3.51 shows the principle of the critical speed for a single-stage centrifugal pump.

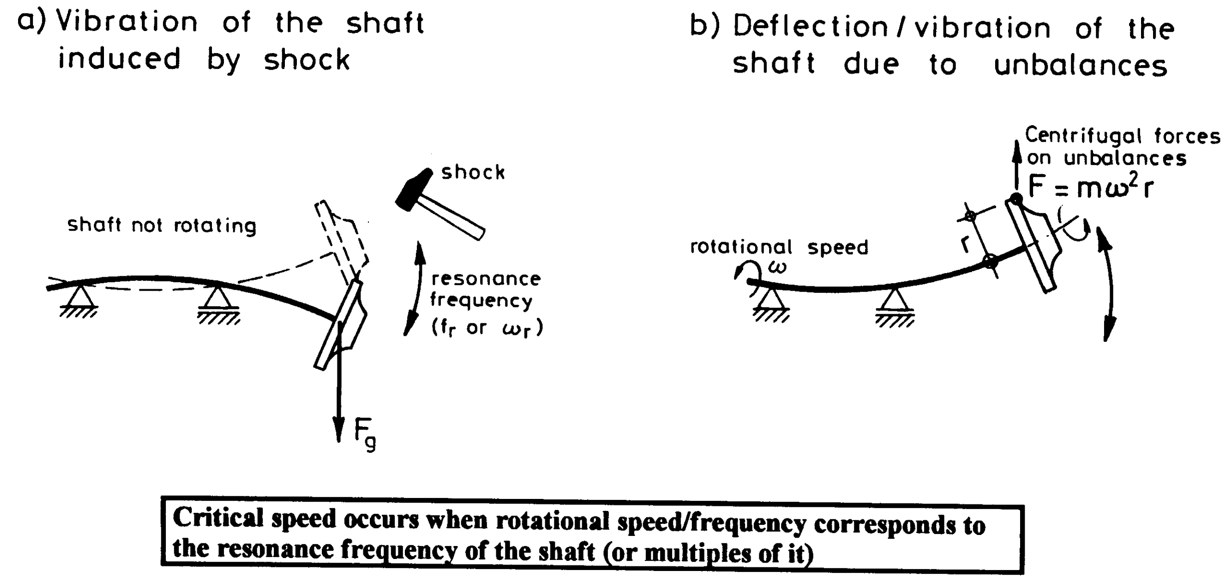


FIGURE 3.52 :
The principle of the critical speed for a single-stage centrifugal pump

There are other causes than pure imbalances of the shaft which may produce critical speeds of turbines and PATs. One source is for example tied to the number of impeller vanes or blades: every passing of the impeller vanes (number of vanes z) at the cut water induces a slight shock on the shaft (change of torque). This vibration can provoke a critical speed if synchronized with the resonance frequency of the shaft /impeller arrangement.

The critical speed will not be a problem for the selected PAT as long as the rotational speed coincides with one of the proposed speeds of the machine in pump mode (see manufacturer's brochure). Similarly, rotational speeds at fractions of the nominal pump speed (1/2, 1/3 1/4) are very unlikely to be a critical speed either and safe turbine operation is possible. Standard single-stage pumps are generally designed in such a way that up to the maximum pump speed no critical speeds are passed; thus, the machine can be operated as a PAT at any speed below maximum speed.
In the case of multi-stage pumps or pumps with long shaft arrangements used as turbines, we recommend to contact the manufacturer, especially if the turbine-mode design speed deviates for more than ±20 % from anyone of the pump-mode design speeds according to the manufacturer's handbook.

1) every downward swing of the shaft or impeller coincides with the occurrence of the centrifugal force acting in the same direction on the unbalanced mass of the impeller

3.8 Summary of the Selection Procedure - Check List

The following form may be photocopied and used as a working sheet (in conjunction with the diagrams of Appendix D) for practical PAT selection.

1. **Design head and flow**

From the preliminary design/layout, the following data are known:

- nominal turbine flow Q_{nt} ....m^3/s
- nominal or net turbine head H_{nt} ....m
- hydraulic power $P = \rho\ g\ Q_{nt}\ H_{nt} = 1.0 * 9.81$ ....=......................kW

Diagram D1 indicates, what type of PAT would probably suit the application

....

2. **Specific speed in turbine mode**

a) Assume a PAT nominal speed $n_{n\,t}$ = ...rpm

Considerations: 1500 rpm is usually a good compromise between PAT size and allowance for runaway speed; if an induction generator/induction motor as generator is to be driven, slip must be added (average values of slip speed see Appendix G)

b) Calculate the specific speed of the installation (turbine mode):

$$n_{qti} = n_t \frac{\sqrt{Q_{ti}}}{H_{ti}^{3/4}} = \ldots\ldots\ldots \frac{\sqrt{\qquad\qquad m^3/s}}{\ldots\ldots\ldots\ldots\ldots\ldots^{3/4}} = \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots$$

For multi-stage or double flow PATs, relate the design head and flow to a single-stage and single-entry impeller : (i_{st} = number of stages, i_{fl} = number of entries)

$$H_{ti} = \frac{H_{nt}}{i_{st}} = \frac{\qquad}{\qquad} = \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots m$$

$$Q_{ti} = \frac{Q_{nt}}{i_{fl}} = \frac{\qquad}{\qquad} = \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots m^3/s$$

3. **Pump-mode specific speed**

$$n_{qp} = \frac{n_{qt}}{0.89} = \frac{\qquad}{0.89} = \ldots\ldots\ldots\ldots\ldots$$

Verify that $n_{qp} > 15$ otherwise change assumed PAT speed or number of impeller stages/entries.

4. **Pump efficiency**

$Q_{np} \approx Q_{nt} / 1.3 =$ .... / 1.3 =........................m^3/s

Use diagram D3 to estimate the maximum efficiency of the suitable pumps.

maximum pump efficiency η_p = ...

5. **Converting turbine design conditions into pump design conditions**

Enter Diagram D2 with pump specific speed and efficiency; read the conversion factors for head C_H and flow C_Q

C_H = ... ; C_Q = ...

Pump design head and flow at proposed turbine speed n_t :

$$H_{np}\,(nt) = \frac{H_{nt}}{C_H} = \frac{\qquad}{\qquad} = \text{.........................m}$$

$$Q_{np}\,(nt) = \frac{Q_{nt}}{C_H} = \frac{\qquad}{\qquad} = \text{.........................}m^3/s$$

6. **Converting pump design conditions at turbine rated speed into pump rated speed**

Standard centrifugal pumps are usually driven by induction motors. Manufacturers therefore indicate the pump performance for speeds lower than synchronous speeds.Use manufactureres brochures to get an idea of what the rated speed of the pump would be.

assumption: np = ...rpm

The affinity laws apply (see also Appendix B):

nominal pump head at pump speed n_p

$$H_{np\,(n_p)} = H_{np\,(n_t)} \left(\frac{n_p}{n_t}\right)^2 = \text{.............} \left(\frac{\qquad}{\qquad}\right)^2 = \text{........... m}$$

nominal pump flow at pump speed n_p

$$Q_{np\,(n_p)} = Q_{np\,(n_t)} \left(\frac{n_p}{n_t}\right) = \text{..................} \left(\frac{\qquad}{\qquad}\right) = \text{................}m^3/s$$

Select a pump (if necessary recalculate from pint 6 with the correct pump speed np)

7. **PAT selection recommendations**

Select a pump with a rated flow (Q_{np}) slightly lower than the value required according to formula above. The PAT will then <u>**operate in the overload range**</u> (which will avoid an excessive drop in efficiency if the predicted bep does not correspond to the actual bep)

If no standard pump is available for these head/flow conditions at the proposed pump speed, recalculate from point 2 using a different turbine speed or assume a different number of impeller stages or entries.

Selected pump:

Type: ....

Rated pump head: H_{np} = ...m

Rated pump flow: Q_{np} = ...m^3/s

Maximum efficiency: $\eta_{p\,max}$ = ...

8. **Converting the best efficiency point of the selected pump into turbine mode**

a) **Calculate the specific speed of the selected pump (best efficiency head and flow related to single-stage and single-entry conditions i)**

$$H_{pi} = \frac{H_{np}}{i_{st}} = \frac{\quad}{\quad} = \text{............................m}$$

$$Q_{pi} = \frac{Q_{np}}{i_{fl}} = \frac{\quad}{\quad} = \text{............................m}^3/\text{s}$$

$$n_{qpi} = n_p \frac{\sqrt{Q_{pi}}}{H_{pi}^{\,3/4}} = \text{........} \frac{\sqrt{\qquad\qquad m^3/s}}{\text{.....................}\quad 3/4} = \text{........................}$$

b) **Read the conversion factors for head and flow from Diagram D2 using the maximum efficiency of the selected pump.**

C_H = ... ; C_Q = ...

Estimate a <u>performance range</u> by including the following scattering factors:

conversion factors for head:

$C_{H\,max} = 1.1 * C_H = 1.1$ .... = ...

$C_{H\,min} = 0.9 * C_H = 0.9$ .... = ...

conversion factors for flow:

$C_{Q\,max} = 1.075 * C_Q = 1.075$ .... = ...

$C_{Q\,min} = 0.925 * C_Q = 0.925$ .... = ...

c) Calculate the turbine best efficiency point (bep t) for both maximum and minimum conversion factors

Turbine design head and flow at rated pump speed:

$H_{nt}\max(n_p) = C_{H\max} H_{np} =$ .... = ... m

$H_{nt}\min(n_p) = C_{H\min} H_{np} =$ .... = ... m

$Q_{nt}\max(n_p) = C_{Q\max} Q_{np} =$ .... = ... m^3/s

$Q_{nt}\min(n_p) = C_{Q\min} Q_{np} =$ .... = ... m^3/s

d) Relate these turbine bep conditions to the nominal turbine speed n_t = ...rpm

Maximum value:

$$H_{nt\max}(n_t) = H_{nt\max}(n_p)\left(\frac{n_t}{n_p}\right)^2 = \ldots\ldots\ldots \left(\frac{\quad}{\quad}\right)^2 = \ldots\ldots\ldots\ldots\ m$$

$$Q_{nt\max}(n_t) = Q_{nt\max}(n_p)\left(\frac{n_t}{n_p}\right) = \ldots\ldots\ldots \left(\frac{\quad}{\quad}\right) = \ldots\ldots\ldots\ldots\ m^3/s$$

Minimum value:

$$H_{nt\min}(n_t) = H_{nt\min}(n_p)\left(\frac{n_t}{n_p}\right)^2 = \ldots\ldots\ldots \left(\frac{\quad}{\quad}\right)^2 = \ldots\ldots\ldots\ldots\ m$$

$$Q_{nt\min}(n_t) = Q_{nt\min}(n_p)\left(\frac{n_t}{n_p}\right) = \ldots\ldots\ldots \left(\frac{\quad}{\quad}\right) = \ldots\ldots\ldots\ldots\ m^3/s$$

9. **Power output in turbine mode at the best efficiency point**

$P_{nt\max} = \rho\, g\, Q_{nt\max} H_{nt\max}(\eta_{p\max} - 0.03) =$

$P_{nt\max} = 1.0 * 9.81$ ....*................. (........ - 0.03) = ...kW

$P_{nt\min} = \rho\, g\, Q_{nt\min} H_{nt\min}(\eta_{p\max} - 0.03) =$

$P_{nt\min} = 1.0 * 9.81$ ....*................. (........ - 0.03) = ...kW

10. **Predicting PAT performance away from the bep**

From Diagram D4 and D5: (with the specific speed of the selected PAT n_{qp} = ...)

For Head versus Flow Curves:
(Diagram D4)

Q_t / Q_{nt}	H_t / H_{nt} (1)
1.2	
1.1	
0.9	
0.8	

For Power versus Flow Curves:
(Diagram D5)

Q_t / Q_{nt}	P_t / P_{nt} (2)
1.2	
1.1	
0.9	
0.8	

For curves with **maximum** values:
Q_{ntmax} = ...m^3/s; H_{ntmax} = ...m; P_{ntmax} = ...kW

point	Q_t =	H_t = H_{ntmax}* factor (1)	P_t = P_{ntmax}*factor (2)
1.2 max	1.2 Q_{ntmax} =		
1.1 max	1.1 Q_{ntmax} =		
bep max	1.0 Q_{ntmax} =		
0.9 max	0.9 Q_{ntmax} =		
0.8 max	0.8 Q_{ntmax} =		

For curves with **minimum** values:
Q_{ntmin} = ...m^3/s; H_{ntmin} = ...m; P_{ntmin} = ...kW

point	Q_t =	H_t = H_{ntmin}* factor (1)	P_t = P_{ntmin}*factor (2)
1.2 min	1.2 Q_{ntmin} =		
1.1 min	1.1 Q_{ntmin} =		
bep min	1.0 Q_{ntmin} =		
0.9 min	0.9 Q_{ntmin} =		
0.8 min	0.8 Q_{ntmin} =		

11. **Plot the turbine performance curves for head and power**.

The intersection points of the system resistance curve(s) (see Appendix A for calculation) with the predicted PAT performance curves (maximum and minimum values) provide the possible PAT operating range:

Q_{efft} **max** =	H_{efft} **max** =	P_{efft} **max** =
Q_{efft} **min** =	H_{efft} **min** =	P_{efft} **min** =

Check whether these conditions are acceptable as regards to the available flow and the planned application.

12. **Runaway speed and flow**

From Diagram D6 and D7: (with the specific speed of the selected PAT n_{qp} = ...)

ε = ... K = ...

Calculate the steady state runaway flow for various turbine heads H_t using the rated pump-mode performance: H_{np}.............................. m; Q_{np}... m^3 /s;

point	H_{tR} =	$Q_{tR} = K\, Q_{np} \sqrt{\frac{H_{tR}}{H_{np}}}$
a)	1.2 H_{np} =	Q_{tR} =
b)	1.6 H_{np} =	Q_{tR} =
c)	2.0 H_{np} =	Q_{tR} =
d)	2.5 H_{np} =	Q_{tR} =

Plot the **no-load line** with these head and flow values (H_{tR} / Q_{tR}).
The intersection point with the system resistance curve (minimum friction) gives the
steady-state runaway conditions:

H_{tR} = ... m ; Q_{tR} = ...m^3/s

Calculate the corresponding **steady-state runaway speed**:

$$n_{tR} = \varepsilon\, n_{np} \sqrt{\frac{H_{tR}}{H_{np}}} = \text{.....................} * \text{.....................} \sqrt{\rule{3em}{0.4pt}} = \text{..................} \text{ rpm}$$

Read the maximum change of flow from your diagram:

$\Delta Q_{max} = Q_{tR} - Q_{eff_t\,max} =$ .... - .... = ... m^3/s

(= .... % of $Q_{eff_t\,max}$)

13. **Further checks and modifications on the PAT**

a) Runaway speed under transient conditions / waterhammer (see Section 4.4)

b) Cavitation / Setting of the PAT above tailrace water level (see Section 3.6.2)

c) Design a draft tube / diffusor throat (see Section 3.6.5)

d) Design coupling arrangement (direct coupling, belt or chain drives, gear box) according to the application of the PAT (see Section 3.7.5)

e) Check the bearing design & shaft resistance including axial thrust and additional load due to coupling arrangement (see Section 3.7.7)

f) Check critical speed (see Section 3.7.7 d)

14. **Design Governing Concept / Flow Regulation** including the selection of a suitable control valve (see Section 4.2 and 4.3)

15. **Compare economy of the PAT with a conventional turbine** (see Chapter 5.)

4. OPERATION AND CONTROL

4.1 General

In the previous chapter emphasis was given to the PAT mainly as an isolated machine for which a selection procedure was presented and whose design details were discussed.
Chapter 4 now deals with the PAT when installed and operating in an MHP plant and its interdependence with other system components (penstock, end-use of hydro-power).

4.2 Governing

4.2.1 Extent of Governing Required

The speed of a turbine or a PAT varies with the load put onto the shaft (see Figure 4.1 below). When directly driving mechanical equipment, the load on the turbine and consequently its speed varies with the operating conditions of the coupled machine (e.g. a mill fully, partly or not charged at all). A speed deviating from its nominal value might be critical for the operating condition of that machine.
When electricity is generated at an isolated power plant, its frequency (and voltage) are determined by the speed of the turbine/generator unit. If generated power and consumed power do not match, the speed of the turbine/generator unit increases or decreases and so do frequency (and voltage). However, all electric appliances are designed to operate at a distinct frequency and voltage (nominal values). Deviating from those might either affect the lifetime of the appliances or they might not work properly.
For these reasons, some form of speed control is necessary in most hydropower plants, be it for direct-drive of machinery or electricity generation.

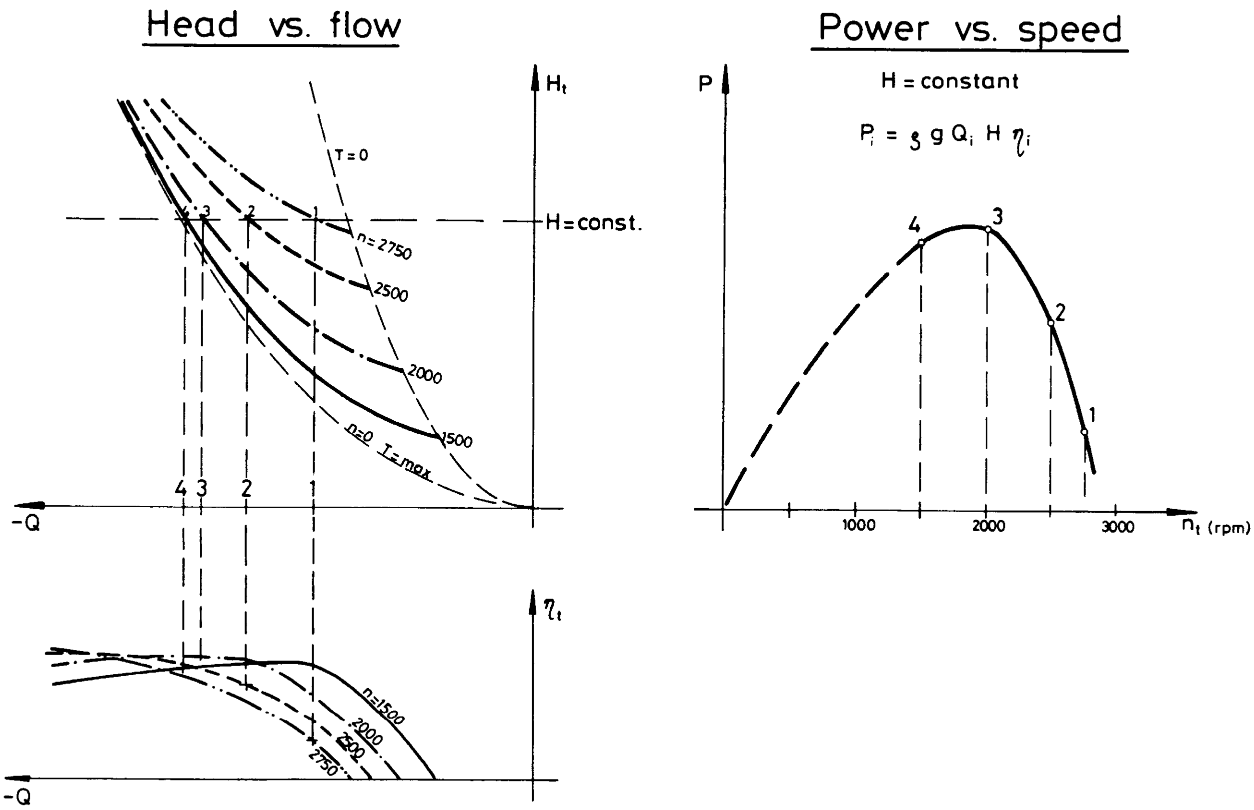


FIGURE 4.1 :
Graph of speed versus power, developed from the normal H - Q and η diagrams introduced in chapter 3

The governing system required for an MHP is mainly dictated by the type of end-user of the power generated. While direct-driven machinery is usually not very sensitive to speed variations, other loads, mainly electric appliances such as fluorescent lamps and motors are affected quite adversely by varying operating conditions (frequency and voltage). Therefore, the design engineer must first determine what variations in speed (for mechanically driven equipment) or frequency and voltage (for electricity generation) are permissible for the different loads planned.

To achieve control over speed, either of the following two options may be applied:

- <u>Flow control</u>: control of the water input by mechanical movement of valves or turbine gates;
- <u>Load control:</u> control of the electrical output through power electronics

When using a PAT , the choice of the governing system is considerably reduced since the PAT itself has no hydraulic control device such as the guide vanes of conventional turbines. Governing must therefore take place either in the hydraulic system (penstock valve) or on the consumer side (load control on machinery or generator). This lack of hydraulic control of the PAT need not necessarily be regarded as a disadvantage. Since a PAT does not require a sophisticated (and expensive) governor to operate guide vanes or nozzles, both investment costs and the know-how requirements of operators and mechanics are reduced. That is why PATs contribute favourably to low-cost solutions in micro-hydro also as regards to governing.
The following sections describe some of the possible approaches to governing using the same categories of end-use appliances as introduced in section 3.2.2 above.

4.2.2 Direct-driven Machinery and Pumps

- <u>Ungoverned Systems:</u> A PAT direct-coupled to machinery and pumps can in many cases be operated without any governing device. For most mechanical uses, speed variations are not critical. Furthermore, experienced operators of machines in agro-processing factories (mills, oil expellers, rice hullers) can load the machines in such a way that the PAT operates smoothly close to its design point.

  When driving a centrifugal pump in parallel with other processing machinery of varying loads, the pump acts like a self-regulating device. If some of the loads are disconnected, PAT speed cannot rise excessively since the torque absorbed by the centrifugal pump increases with the second power of speed and stabilizes the system very effectively; i.e., the pump takes over some of the power from the disconnected load without incurring a large increase of speed.

- <u>Manual Control using the Control Valve:</u> The penstock of an MHP installation is usually equipped with a control valve to start and stop the turbine/PAT. This valve can also be used to regulate the flow through the PAT and subsequently its power output. With this approach, the operator of machinery maintains constant turbine speed by matching the flow through the PAT (and its power output) with the power demand of the driven machines. When using only a limited number of machines which require less power than the PAT was design for, slight throttling of turbine flow will therefore keep the system at the designed speed. As mentioned earlier, the control valve dissipates hydraulic energy when used as a throttling device (reduction of head) and is therefore not very effective. Efficiency is however of no importance when exclusively driving machinery, provided that sufficient stream flow is available throughout the year. (For more details see section 4.2.4 and section 4.3)

4.2.3 Grid-linked Electricity Generation

When generating electric power using an induction generator connected to a large grid, no governor is required since the grid itself regulates both frequency and voltage. It is the grid which dictates the speed of the PAT automatically and therefore releases the operators of an MHP from the responsibility of speed and voltage/frequency control. In this respect a PAT behaves just like a conventional turbine.

In case streamflow falls short of the design flow of the turbine or PAT during the dry season, a non-governed approach is not possible since the turbine would absorb too large a flow and would partly empty the penstock.
To maintain head and thus prevent power output to drop sharply, the conventional approach (purpose-built turbine) uses a water level control. The governor senses the water level in the forebay and closes the guide vanes of the turbine as soon as the water level drops. In this way, all available water can be used under the nominal turbine head which ensures optimum energy production despite a reduced streamflow (no air is entrained into the penstock and turbine).

When using a PAT which lacks adjustable guide vanes, a similar approach is only possible if the water level control governor acts on the control valve of the penstock (see section 3.4.3) or if the valve is throttled manually during the dry season . As explained earlier, the control valve is an inefficient means of governing since it does not only reduce flow but simultaneously dissipates pressure head. Thus, efficiency drops sharply and the range of flow which can be accommodated with the control valve as a governing device is very limited. The use of a PAT in conjunction with water level control, (i.e. where optimum energy production around the year is required), is not recommended. Figure 4.2 shows the H/Q and efficiency curve of a PAT operated at part flow (throttled flow using the control valve). The efficiency graph of the MHP using a PAT (η_{system}) is put in comparison with a flow regulated Francis turbine. Assuming that both may yield the same output at the nominal point (Q/Qn = 100 %), the PAT shows a performance highly inferior to the Francis turbine under part flow. While the PAT would most probably stop operating at about 50 % of the nominal flow, the Francis turbine still runs at maximum efficiency due to efficient flow control with the adjustable guide vanes.
This example shows that maximum electricity generation in grid-linked systems exploiting a varying streamflow cannot be achieved with PATs.

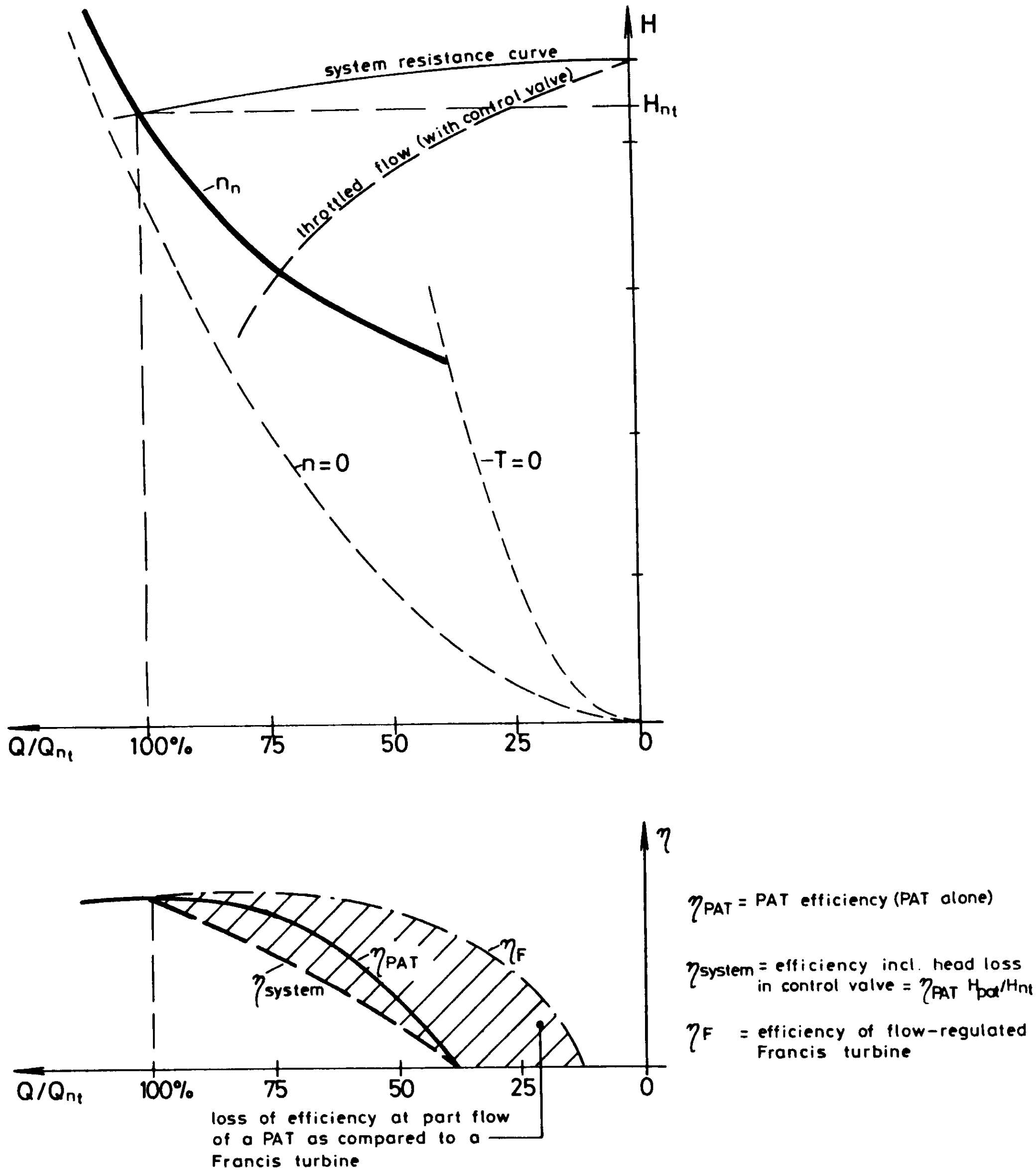


FIGURE 4.2 :
Inefficient operation of a PAT under part flow

4.2.4 Stand-alone Electricity Generation

- <u>**Ungoverned:**</u> Isolated electricity generation can only remain ungoverned if the load on the generator is kept constant. Though being very straightforward and most suitable for the application of a PAT, the ungoverned approach reduces the choice of end-use appliances to village lighting and heating devices.

  The system is designed to operate without switches, i.e., electricity is simultaneously distributed to all consumers at once by opening the control valve of the PAT. Consumers cannot vary appliances except for the use of two-way switches where switching one load off automatically connects a second load of the same rating.

- **<u>Manual Control:</u>** Manual control requires an operator supervising the speed/frequency of the turbine/generator unit; it can take two forms: the operator can either adjust the flow through the PAT by closing the control valve (if streamflow falls below design discharge of the PAT) or he can add or remove load on the generator (e.g. by switching on or off a heating device or lamps) to maintain constant frequency and voltage.

  Note that ungoverned or manual control does not mean that no electrical devices are needed between the generator terminals and the power lines to the end-users. The usual protection equipment such as overload/short-circuit protection, overvoltage relays or frequency trip for synchronous generators must be incorporated into the system in order that the generator is automatically disconnected from the external loads when a fault occurs. This avoids damage to the end-use appliances and the generator itself.

- **<u>Dissipative Base Load:</u>** Electricity is generated in parallel to a relatively large mechanical drive such as a centrifugal pump, a large fan or a mechanical heat generator. Simultaneous operation of the generator and the pump or the mechanical heat generator provides a system with relatively small speed deviations due to the stabilizing feature of the mechanical drive. Figure 4.3 shows the power characteristics of the turbine and the mechanical drive (dotted lines). The solid line represents the power available for the generator , i.e turbine output power minus power demand of the dissipative load. Note that the mechanical drive must have a power characteristic as indicated, otherwise the self-regulation feature of the mechanical drive is not given (e.g. for a centrifugal pump, power increases approximately with the third power of speed; see affinity laws, Appendix B).

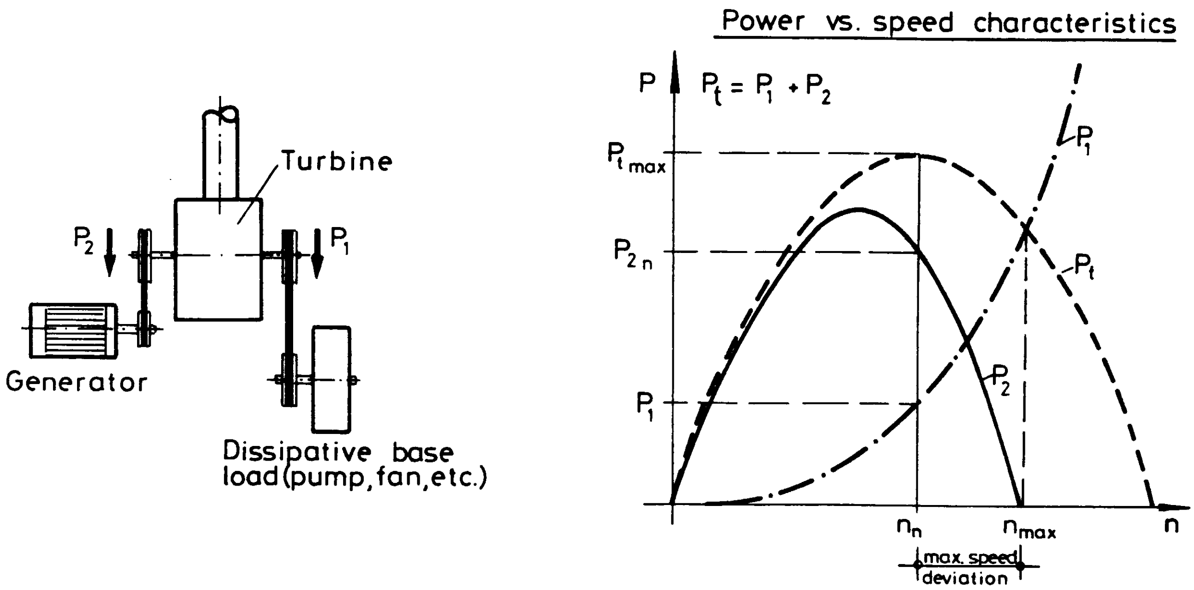


FIGURE 4.3 :
Layout and power distribution of a stand-alone electricity generating unit working in parallel with a dissipative load

Figure 4.3 shows the maximum speed variations of the plant. Minimum speed (n_{min}) of the turbine/generator unit is attained when all electric appliances are switched on simultaneously; this value is determined by the total electric consumer load allowed to be connected to the scheme. When all electrical loads are withdrawn, the unit reaches its maximum speed (n_{max}). Its magnitude depends on the size of the mechanical drive. The use of a belt drive instead of a direct coupling offers fine tuning of the power dissipated by the pump or heat generator by changing the speed reduction ratio (pulleys).

□ **Electronic load controller (ELC):**

1. *Conventional approach using a synchronous generator with in-built automatic voltage regulator (AVR):* The load controller (ELC) is an electronic device which maintains constant load (hence constant frequency) on the generator despite varying end-user loads. Basically, it does the same job as the operator in a manually controlled system; i.e. switching ballast loads onto the generator as soon as the frequency deviates from the nominal value. Figure 4.4. below presents a simplified diagram of a load controller.

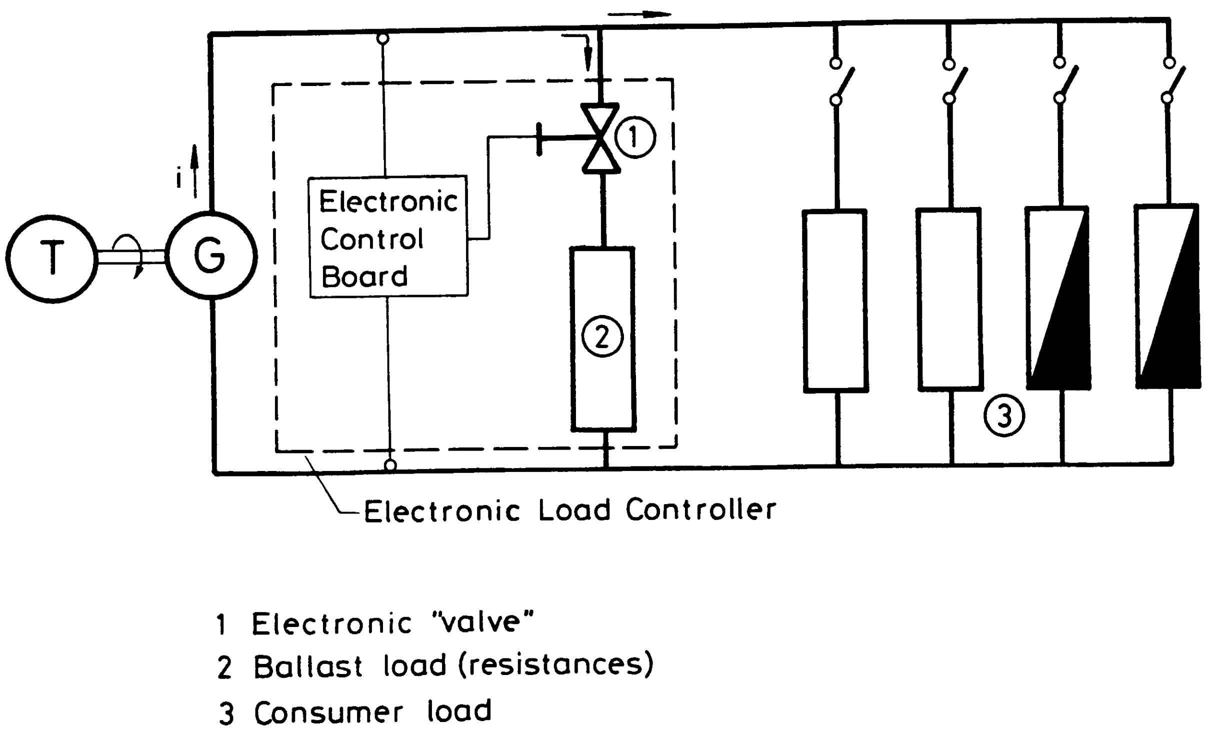


FIGURE 4.4 :
Simplified diagram of a load controller

The electronic control board senses the generator frequency and compares it with a preset nominal value. If some of the consumer loads are switched off, electrical power used by the consumers is less than the power output of the turbine/PAT and the speed of the generator/turbine unit and the generated frequency begin to rise. Sensing this increase of frequency, the electronic control board opens the electronic "valve" (switch) and passes the surplus power to one or several ballast loads which dissipate the surplus energy. Despite the change of consumer load, the total load on the generator is the same and speed and frequency remain approximately constant.
The electronic "valve" can be a number of thyristors or another form of solid-state (semi-conductor) switch or even power relays. These electronic components cannot normally be repaired locally, but as they are of the plug-in type for most of the commercially available load controllers, they can easily be replaced by spare units kept in stock. This feature makes load controllers also fairly appropriate and user friendly despite their relatively advanced technology. Additionally, they are less expensive both in initial investment as well as O&M costs than conventional oil-pressure governors for the power range of up to 50 - 80 kVA.
Care must be taken when selecting the resistances for the ballast load. Commercial heating elements for air (< 3 kW without ventilator) or for water (boiler elements) are usually applied but their reliability is sometimes low. A common causes of failure for water heating elements are overheating due to insufficient submergence or corrosion.

As electronic load controllers do not require a hydraulic control on the turbine side, the use of PATs in conjunction with a load controller is a very favourable solution for stand-alone electricity generation.
Note that in case of failure of the load controller, an emergency shut-down on the hydraulic side must be provided to avoid that the unit runs at runaway speed for a long period. This usually requires a control valve in the penstock operated either automatically or by an operator who is always near the plant.

A list of manufacturers of ELCs is given in Appendix J.

2. *Induction (asynchronous) generators or induction motors as generators*

The use of ELCs is not confined to synchronous generators with AVR. Recently, the development of a load controller for induction generators (IGC) both in the UK and in Switzerland has been reported. Therefore, the advantages of induction generators for sizes up to 50 kW (less expensive than synchronous ones, more reliable) can now be used also for stand-alone electricity generation. The installation of an induction generator in free- standing electricity generation contributes very favourably to the low-cost philosophy of PATs. A further cost reduction is possible if **standard induction motors are used as generators**; thus, standard pump-motor sets may be run very cost-effectively as turbine(PAT) - generator sets (a special manual is available for more details on the use of induction motors as generators; see bibliography).

The induction generator controller (IGC) combines frequency and voltage control in one unit, hence, no additional and costly voltage regulator is required. The IGC senses voltage rather than frequency and, similar to the ELC for synchronous generators, diverts surplus power to a dissipative load if voltage rises. The IGC system is most suitable for smaller MHPs (up to 50 kW) and those with total motor loads on the consumer side not exceeding 20 to 25 % of nominal generator output (starting too large a motor will cause self-excitation to collapse). Appendix J provides the addresses of the institutes developing the IGC.

- **Speed-flow governor**

The disadvantage of regulating speed / frequency of a generator by means of adding ballast loads or dissipative loads both manually or electronically , is that they do not economize on the use of water. The ballast or dissipative load, if not used productively, is wasted energy. This is only of importance where there is water storage and does not concern the run-of-river schemes.

In conjunction with a storage pond, PATs usually operate on an on-off basis (intermittent operation; e.g. day on - night off) due to their lack of hydraulic control device. Since complete stoppage of supply is not wanted in isolated electricity generation, a speed-flow governor might be used on the PAT/generator unit acting on the control valve. Similar to a conventional governor, speed deviations of the PAT are sensed and the necessary adjustments made on the control valve.

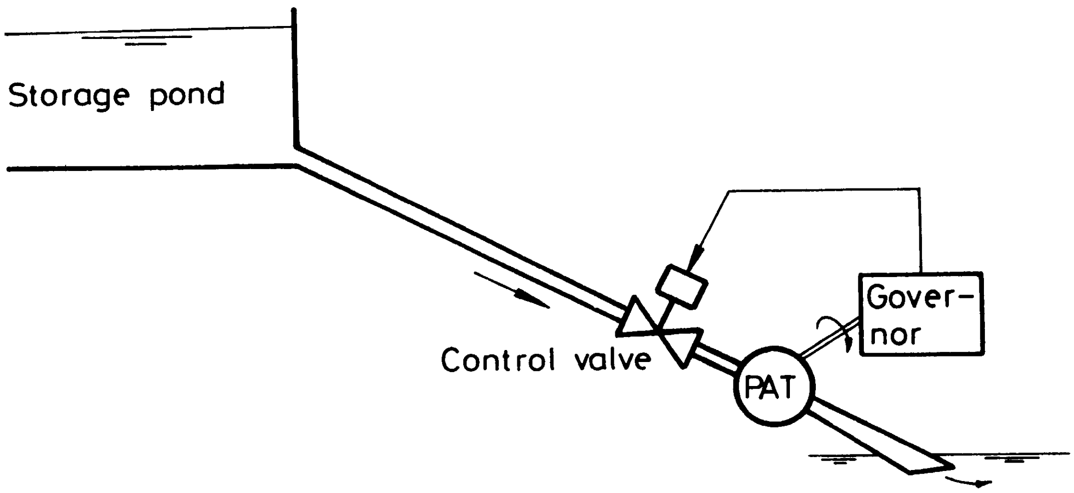


FIGURE 4.5 :
Speed-flow governor in conjunction with a PAT

To the authors' knowledge, this approach has not been tried so far. Regulation with a nonlinear characteristic of the control valve (opening versus flow) results in a fairly demanding job for a governor of whatever type. As shown in Figure 4.2, a control valve is generally an inefficient means of governing. For these reasons, this solution should not be attempted without further investigation and tests.

TABLE 4.6 :Overview of possible governing systems for PAT-driven MHPs of different applications

	Application		
Governing system suitable for MHPs using a PAT	**Direct drive of machinery**	**Grid-linked electricity generation**	**Stand-alone electricity generation**
a) Ungoverned	suitable	suitable	possible, but limited choice for the end-use appliances
b) Manually controlled valve	suitable	-	possible but not efficient
c) Dissipative load / switched manually	suitable	-	possible
d) Dissipative base load	suitable	-	possible
e) Water level control acting on the control valve	-	possible but inefficient	possible but not sufficient for regulating speed under varying loads
f) Electronic load controller (ELC)	-	-	highly recommended in conjunction with synchronous generators (50 - 80 kVA)
g) Induction generator controller (IGC)	-	-	recommended in conjunction with induction generator / motor up to 50 kW
f) Speed-flow governor	possible if water storage available but not recommended	-	possible if water storage available but not recommended

4.3 Starting and Stopping PATs - Control Valve

4.3.1 Procedure

Starting up a PAT depends on the application of the hydropower but is similar to the procedure of conventional turbines except that the control valve in the penstock instead of guide vanes must be used. In addition to starting and stopping a PAT, the control valve serves as a PAT isolation valve to permit working on the machine (replacement of gland packing, etc.). In any case, the valve must be operated slowly to avoid pressure surges and waterhammer problems in the penstock (see also section 4.4).
For grid-linked electricity generation using a synchronous generator, synchronization with the grid is required at every start-up. Additional equipment (synchronoscope or lamps) and a fairly good understanding of electricity is needed if carried out manually. When using an induction generator, the procedure is much less demanding (after acceleration of the turbine-generator unit to about synchronous speed, the main circuit breaker is simply closed and the control valve further opened according to the available water).

4.3.2 Position of the Control Valve

Basically, there are two alternative positions of the control valve in the penstock, either upstream or downstream of the PAT. If the valve is used exclusively in **on-off mode**, the upstream position is preferable: the PAT is always drained after valve closure and maintenance on the machine is possible without emptying the penstock. It should, however, be placed at some distance from the PAT inlet flange (about 10 * nominal bore of the penstock pipe) since the valve disc and seats might create an irregular flow pattern approaching the PAT, thereby affecting the efficiency of the machine.

If the design concept foresees **permanent adjustment of flow** using the control valve, both the downstream and the upstream position have advantages; the final decision will depend on the local conditions and valves available.

- **Upstream position**:

- PAT can be isolated for maintenance without emptying the penstock

- reduced risk of cavitation at the valve when throttled but

- risk of cavitation on PAT at full flow (check with the values given in section 3.6.2 above)

- valve in throttled position induces turbulence on PAT inlet [may be reduced by a sufficiently long distance between valve and PAT or by a flow straightener (but which may clog)].

- **Downstream position**

- reduced danger of cavitation on the PAT because the valve increases the head loss between tail race and machine and subsequently the backpressure on the PAT but

- the valve may cavitate when used as throttling device; the minimum backpressure required by a valve of a particular design must be checked with the manufacturer

- the PAT is under full static pressure if not in operation; the shaft seal (e.g. a stuffing box) might not be designed for this pressure. During operation, full penstock pressure on the seal might be compensated by wearing rings, balance holes or back vanes but these devices do not work at standstill (see also section 3.7.6 Axial Thrust);

- when closing or opening a valve located downstream of the PAT, pressure surges might develop due to rapid change of flow. Thus, transient peak pressure (waterhammer) will also act on the PAT which must be design to stand the resulting stress;

- the outlet flange of a PAT is usually larger than the inlet side which requires either a valve of a larger size or tapper pieces.

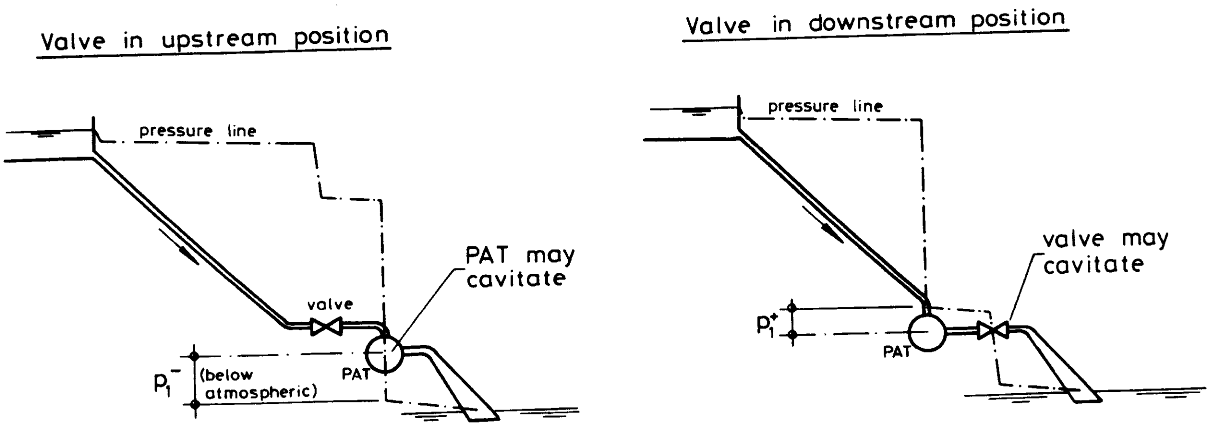


FIGURE 4.7 :
Position of the control valve of an MHP using a PAT

4.3.3 Type of Valve to Be Used in Conjunction with PATs

There is a distinct difference between a control valve for pure on-off purposes and a valve with flow regulation features. While the former should have a very low loss coefficient (zeta, see below) when fully opened, the latter requires a certain resistance to flow (conveniently proportional to the opening) to achieve flow regulation. These two requirements cannot normally be combined in one valve design.
For small hydropower installations using a PAT, flow regulation with a valve is required in many applications, be it to correct predicted to actual PAT performance (see section 3.6.3) or to adjust power output to demand (see section 4.2). However, simple stop valves create irregular flow when operated at partial opening. This might lead to cavitation and subsequent damage on the valve and the pipe especially if the valve is used as a permanent throttling device.

From the above, the following indications on appropriate valve selection can be deduced:

- **Valve for on - off operation**

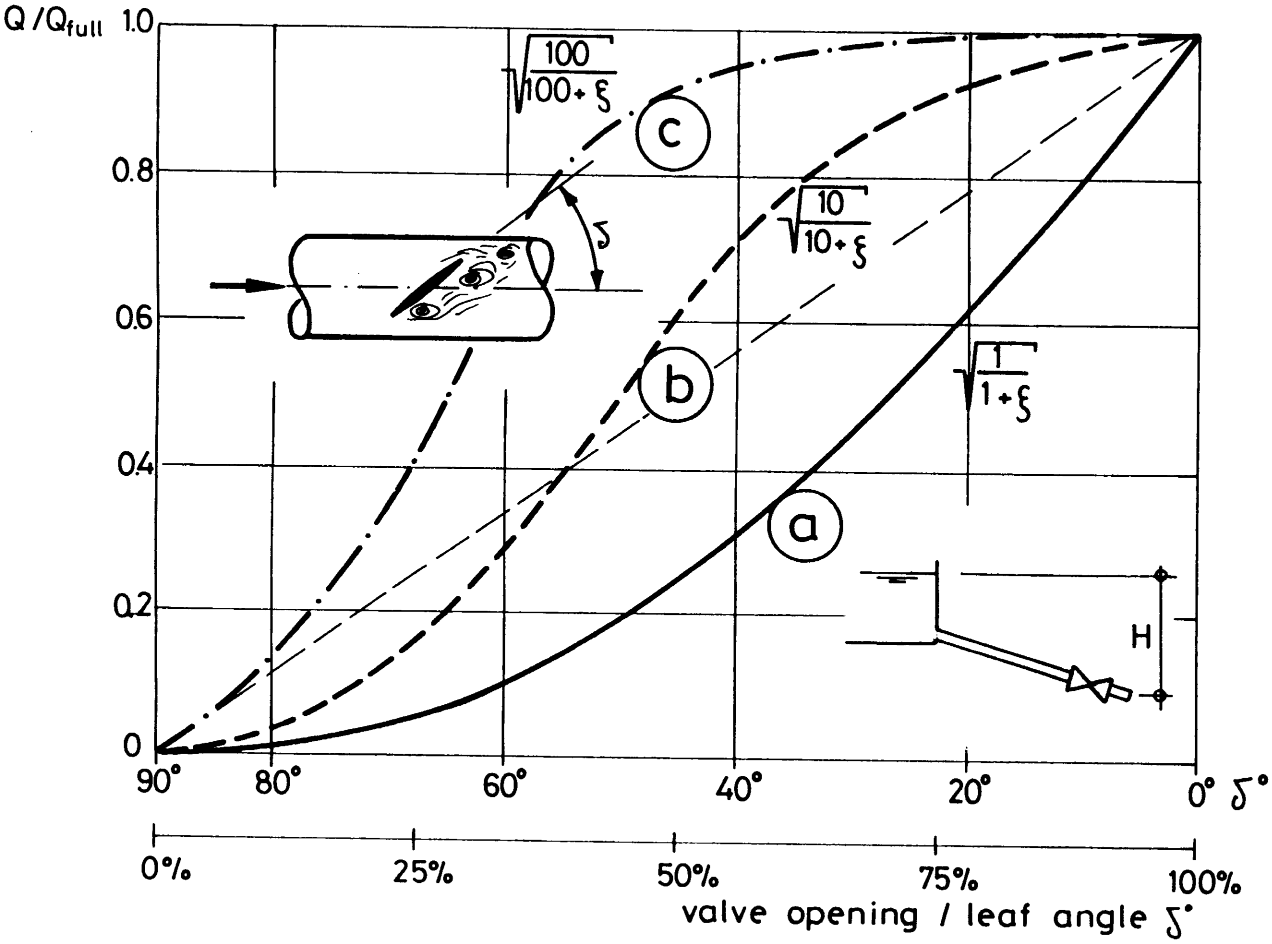
- valve nominal bore should be chosen to **minimize head loss** caused by the valve; usually the same nominal bore as the penstock pipe or the PAT inlet flange is selected, i.e the valve should be as large as possible

- **Valve for flow regulation**

The **nominal bore of the valve should be as small as possible**. Valve manufacturers indicate a permissible maximum velocity through the fully opened valve (nominal flow) of between 2 and 4 m/s. If the valve is too large, resistance to flow and thus the throttling effect becomes only appreciable near the closed gate or leaf position (i.e. the valve characteristic is nonlinear). This makes adjustment fairly difficult and may also cause cavitation and vibration.

Figure 4.8 below shows the characteristics of a butterfly valve. Curve "a" represents the relationship between flow and valve opening if the head loss of the system is negligible in relation to the loss created by the valve (small valve in a large penstock). The characteristics "b" and "c" on the other hand represent valves with relatively small head losses in relation to the system head loss (large valve compared to penstock). While minimizing head loss at fully opened valve may be favourable for maximum power generation, it becomes rather inconvenient when flow should be regulated: closing the valve will change almost nothing at the beginning while all of a sudden, flow diminishes rapidly over a small change of valve opening. This nonlinearity of the characteristic makes flow regulation difficult not only for operators (hand-regulation) but more so for electronic or mechanical governors acting on the valve (unstable operation due to constant adjustment and over-reaction by the governor).

Additionally, the characteristic curve "c" (the large valve) achieves the same throttling effect as curve "a" at a valve position closer to complete closure, i.e the leaf of valve "c" is much more oblique in the flow than valve "a" which is more likely to cause separation of flow and cavitation. **<u>That is why a small valve should be chosen when permanent regulation of flow is required.</u>**



for a penstock with valve (without PAT or turbine):

$$H_{static} = \frac{v^2}{2g}\left[\lambda\frac{L}{d} + \Sigma\zeta_1 + 1 + \zeta\left(\frac{d_0}{d}\right)^4\right]$$ where penstock losses (friction + local + outlet velocity head) = $v^2/2g\ (\lambda\frac{L}{d} + \Sigma\zeta_1 + 1)$ =

$v^2/2g\ \zeta_{piping}$ (with v = pipe velocity) and

head loss in valve (related to pipe velocity) = $v^2/2g\ \zeta\left(\frac{d_0}{d}\right)^4 = v^2/2g\ \zeta_{valve}$

(with d_0 = valve nominal bore, d = pipe diameter)

$$Q = \frac{\pi d^2}{4}\sqrt{2gH\frac{1}{\zeta_{piping} + \zeta_{valve}}}$$

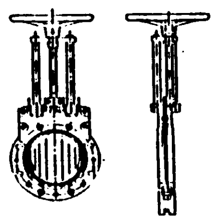
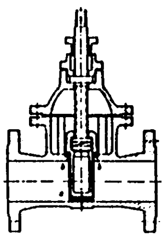
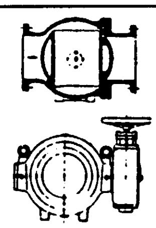
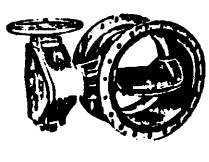
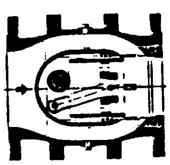
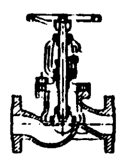
$$\boxed{Q/Q_{full} = \sqrt{\frac{\zeta_{piping}}{\zeta_{piping} + \zeta_{valve}}}}$$ (with ζ_{valve} fully opened = 0)

curve a : head loss penstock is negligibly small (ζ_{piping} = 1)
curve b : head loss penstock is increasing (ζ_{piping} = 10)
curve c : head loss penstock is large (ζ_{piping} = 100)

FIGURE 4.8 :
Typical valve characteristics (flow versus opening) for a butterfly valve at constant head (without PAT)

The following table gives an overview of valves generally available on the market in sizes required for micro-hydropower installations (NB 100 - 500 mm) and their suitability as flow regulating devices.

TABLE 4.9 : Overview of valves used in micro-hydropower installations and possible application as throttling device

Type of valve **pressure range**		**Advantage**	**Disadvantage**	**Suitable for flow regulation**
GATE VALVE SINGLE DISC pmax = 6bar		- readily available - simple design	- only for low pressure (25 to 60m or sometimes 100m maximum pressure)	**possible for low head installations and maximum flow velocity 3 m/s**
Wedge valve (double disc) pmax = 16 bar		- readily available	- significant forces to actuate the valve for large diameters (might require a small by-pass valve)	not recommended (damage on seals if permanently used as a throttling device; disc tends to vibrate at partial opening)
Ball valve (sphere, cock, plug valves) pmax = 64 bar		- compact design (small height) - can be actuated with little force - straight through flow when fully opened, no losses	- not readily available for large size diameter - larger diameters expensive	not recommended (flow separation at edges of the sphere, cavitation)
BUTTERFLY VALVE pmax = 25 bar		- can be actuated with little force	- not yet available everywhere at reasonable price	**may be used for flow velocities up to 3 m/s and low pressure installations**
NEEDLE VALVE pmax = 40 bar		- actuated with little force - smooth flow even when throttled - free of cavitation at partial opening (- used in water supply systems)	- very expensive - not everywhere available at reasonable price	recommended for all velocities but does not suit the low-cost philosophy of PATs
GLOBE VALVE pmax = 6 - 160 bar		- characteristics individually variable - watertight - models for very high pressures available	- only for small sizes - high head loss in fully opened position	**recommended for flow regulation (max velocity of water 2 - 4 m/s)**

4.3.4 Valve Characteristics

When dealing with pressure transients induced by valve closure/opening, knowledge of the valve characteristics is required (see next section). Suppliers of valves usually provide these graphs (head loss versus valve opening curves) or might help to select suitable valves for larger installations. For micro-hydropower installations, this kind of support is rarely available; Appendix E therefore provides general characteristics for a number of valve types which might be used in micro-hydro. The transformation of the head loss coefficients into actual discharge through the valve at partial opening (for a specific site) is also shown.

4.4 Waterhammer / Pressure Surges

4.4.1 General

In hydropower installations, pressure surges occur as a result of changes in velocity of flow in the penstock. Caused by either rapid closure or opening of the control valve or by a sudden load rejection of the turbine / PAT (failure of coupling, transmission belt , grid failure in electricity generation), these pressure surges can reach several times the static pressure of the installation. Precautionary measures must be taken to avoid damage to pipelines and machines.
Waterhammer and pressure surges have been introduced in Appendix A for the case of sudden and gradual valve closure. This chapter deals mainly with load rejection of the PAT and the subsequent pressure transients induced in the penstock.

4.4.2 Gradual Closure/Opening for Nonlinear Valve Characteristics

Figure 4.8 above has shown that valve characteristics are seldom linear over the whole range of valve stroke. For many valve/penstock configurations the rate of change of flow is low at the beginning of valve closure and will increase rapidly towards complete closure. The magnitude of possible pressure surges (waterhammer) induced by valve closure will therefore depend on the **maximum rate of change** of flow rather than on the mean value. For practical purposes an **equivalent linear valve characteristic** is defined as shown in Figure 4.10 below. The corresponding closure time should be used for the waterhammer computations as shown in Appendix A for gradual valve closure. Note that the valve opening (angle δ) might not always correspond to the valve stroke (revolutions of wheel) because of the gate mechanism; a diagram of flow versus stroke rather than versus opening should be employed to define the equivalent valve closure time.
Generally, a lever-actuated valve as sometimes offered for small butterfly or ball valves should not be used on PAT installations since very short closure times and subsequently large pressure surges can be produced with this mechanism.

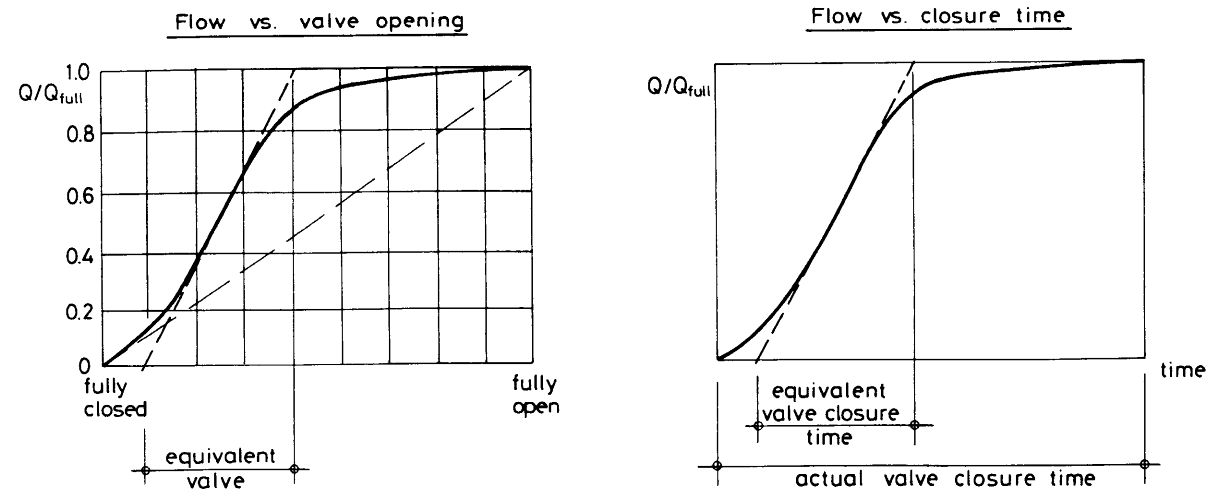


FIGURE 4.10 :
Equivalent gate closure time for waterhammer computations

4.4.3 Pressure Transients due to Load Rejection of the PAT

In section 3.6.1, the problem of the runaway speed of a turbine or PAT has been introduced. It was shown that the PAT is accelerated to runaway speed when all external loads are removed. The values given for runaway speed of different machines corresponded to the final steady state conditions of the PAT. It was also shown that a change of speed is in most cases accompanied by a change of flow (except for turbines/PATs of $nq \approx 80$) which, as stated above, basically bears the danger of inducing pressure surges into the pipeline if the rate of change ($\Delta Q/\Delta t$) is high. These pressure surges or pressure transients might drive the PAT temporarily to a higher speed than according to the steady state runaway speed. Figure 4.11 below presents this transient behaviour of a radial flow PAT and its interrelation with the penstock system in a head-flow (H/Q) diagram (note the difference to Figure 3.31, where the change of flow was assumed to take place very slowly (along the system resistance curve) without inducing pressure transients).

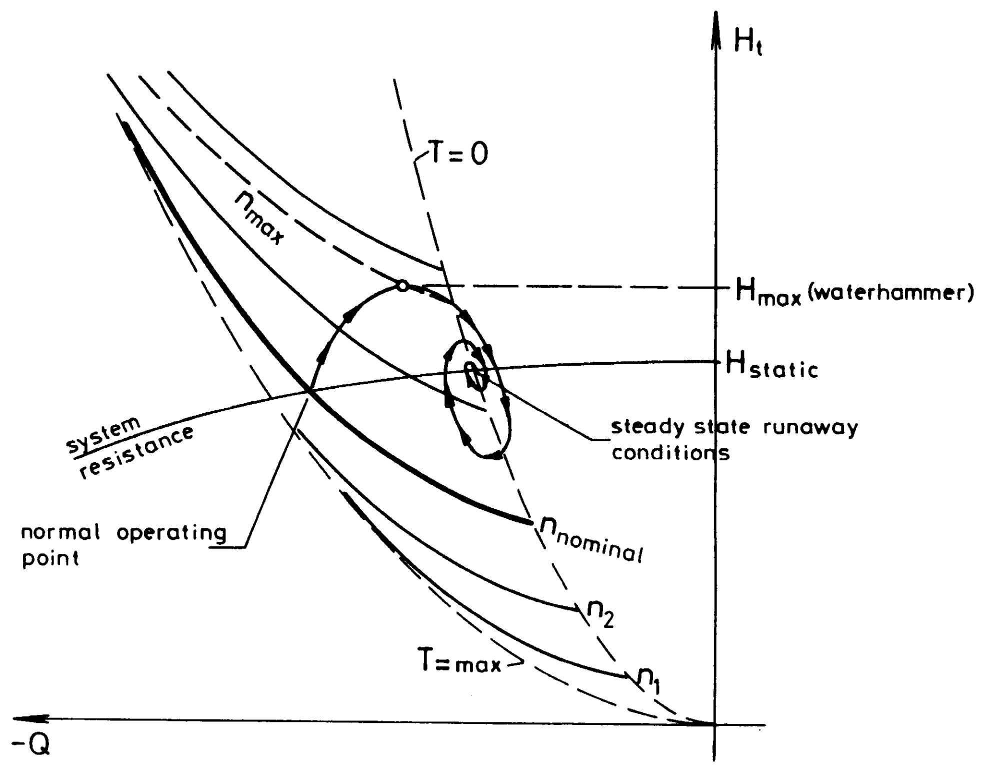


FIGURE 4.11 :
Transient pressure development and runaway speed after total load rejection of a radial flow PAT

As can be imagined, the determination of the maximum PAT speed and pressure head after full load rejection including waterhammer is rather involved. Somewhat more analysis is required than for the computation of the waterhammer due to valve closure/opening where a simple formula could be given for a rough estimate (see Appendix A). While the gate movement was determined externally by an operator, PAT speed/flow variations after load rejection is itself affected by the development of the waterhammer; this requires an iterative solution of the problem.

In the following, we will present a simplified graphical method to estimate maximum waterhammer due to load rejection of a PAT. Further development of this method would lead to the general solution of transient problems in hydraulics according to Schnyder / Bergeron.

4.4.4 Machine Parameters

When the load on the PAT shaft is suddenly removed, the torque produced by the water flowing through the impeller is no longer in equilibrium with load torque. This torque difference therefore accelerates the PAT until a new equilibrium between the driving hydraulic and the resistive forces (friction) is reached. Figure 4.12 b) shows the development of the torque from nominal to runaway speed under constant head. As speed changes so does the flow through the PAT. Figure 4.12 c) shows the speed versus flow relationship for a radial flow PAT. Note that the flow through an axial flow machine might be increasing with rising speed. Both curves T/n and Q/n are developed from the PAT performance curve calculated in the course of the PAT selection procedure discussed in Chapter 3. (Figure 4.12 a) To simplify calculations, a linear relationship between speed and torque, and flow respectively has been assumed.

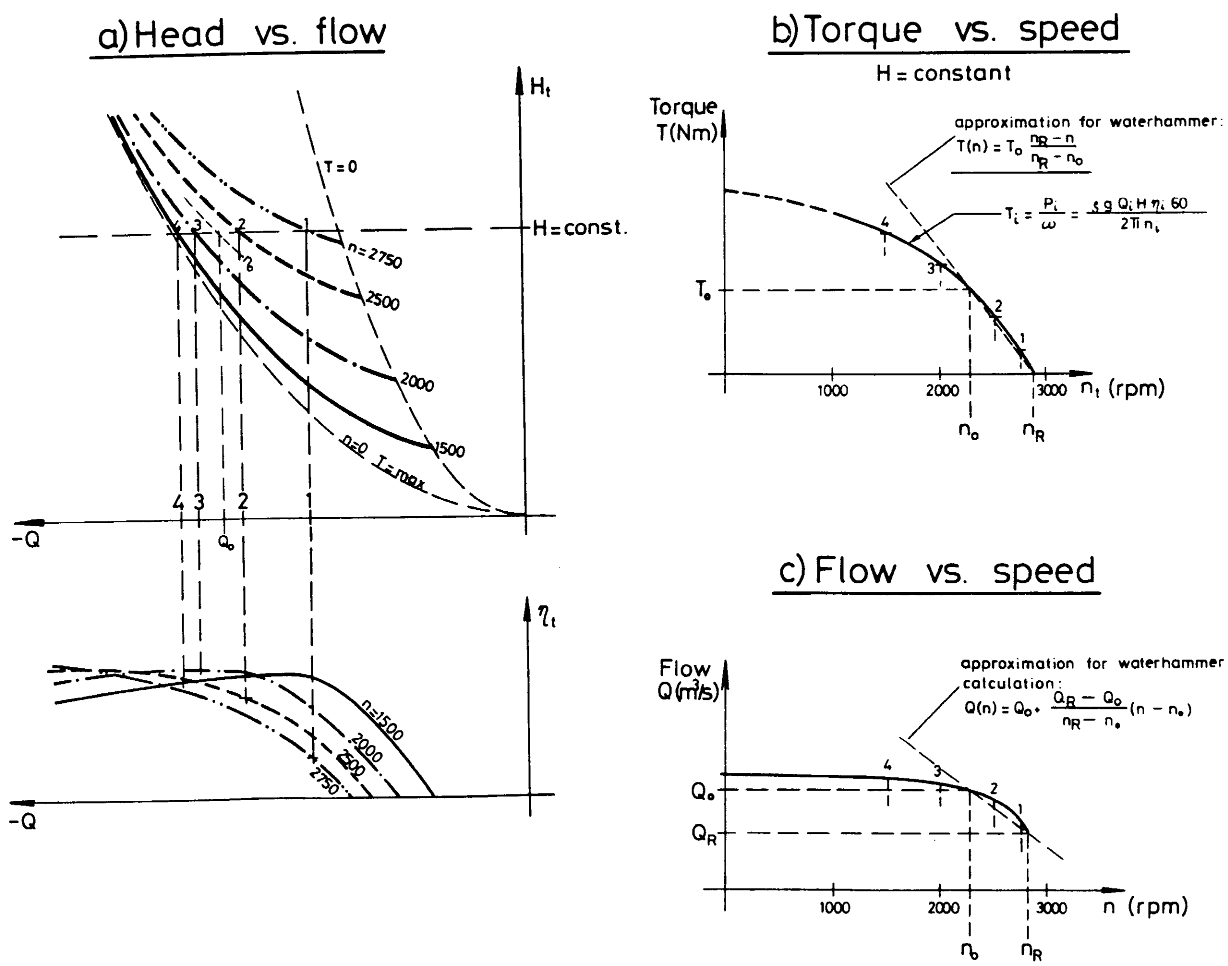


FIGURE 4.12 :
Development of torque and flow versus speed for a radial flow PAT under constant head

It is now decisive for the magnitude of the waterhammer, **in what time** the PAT would accelerate from the operating point (Q_o) to runaway conditions (Q_R). The faster the change of flow (Δ Q), the more powerful the induced waterhammer will be.
Acceleration of the PAT depends on the driving torque and on the rotating masses (PAT impeller, pulleys, flywheel and generator rotor or machine assemblies). Applying Newton's second law of motion on rotating elements we will find:

$$\boxed{\text{ACCELERATION}: \quad \frac{d\omega}{dt} = \frac{T_{(\omega)}}{J}} \qquad (4.1)$$

where ω = rotational (angular) speed (rad/s)
T (ω) = accelerating torque (Nm) as a function of rotational speed and
J = moment of inertia (kgm2) of all rotating masses (details of calculation see Appendix F)

As accelerating torque T(ω) we may use the speed - torque relationship developed for constant head in Figure 4.12 a), knowing that this will not be quite correct in many cases since head increases due to pressure surges. Thus, acceleration may be expressed with the following differential equation:

$$\boxed{\frac{d\omega}{dt} = \frac{T_0}{J} \; \frac{\omega_R - \omega}{\omega_R - \omega_0}} \qquad (4.2)$$

Integrating this, leads to an exponential law for rotational speed ω versus time. Figure 4.13 below indicates that the runaway speed ω_R would be reached at infinity. For our first estimate, we will use a straight line with the slope of the curve at the starting point (ω_o). Thus, the accelerating time becomes:

$$\boxed{T_{a\,eff} = \frac{\omega_R - \omega_0}{\omega_0} \, T_a} \quad \text{where} \quad \boxed{T_a = \frac{J\,\omega_0}{T_0}} \qquad (4.3)$$

T_a is called the unit accelerating time.

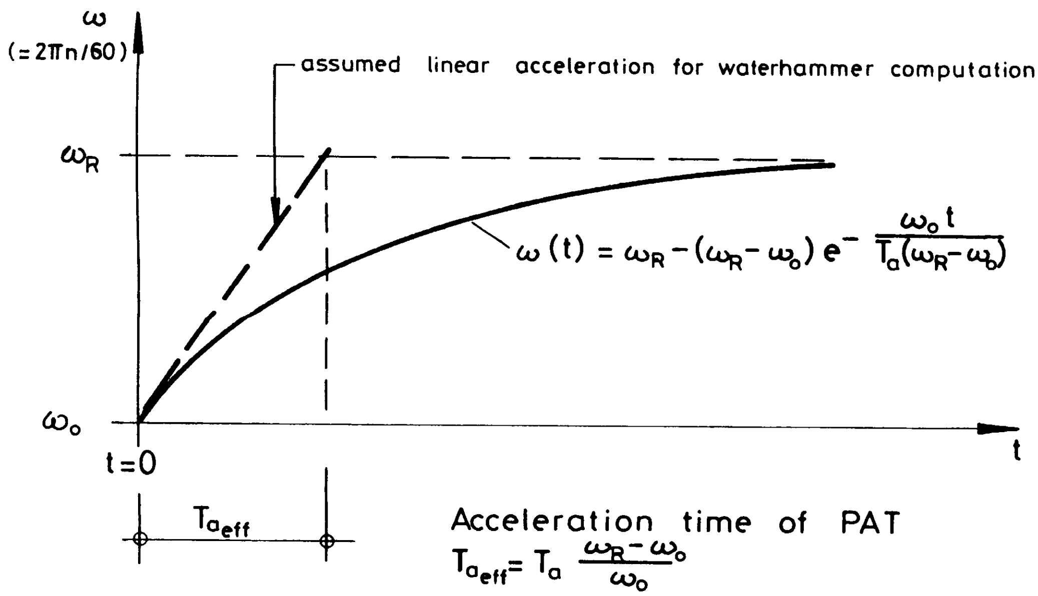


FIGURE 4.13 :
Development of PAT speed versus time under a linearly diminishing torque T

4.4.5 Penstock Parameters

The main parameter of the penstock is the reflexion time or period

$T_r = 2L/a$

where a = propagation velocity of the pressure wave along the penstock and L = penstock length (see also Appendix A). It represents the time required by the pressure wave to travel from the PAT to the forebay and back. For compound penstocks (varying diameters and/or pipe materials), use the equivalent wave velocity and the equivalent pipe cross-sectional area defined as follows (limits of application see below):

$$\boxed{a_{equi} = \frac{L}{\frac{L_1}{a_1} + \frac{L_2}{a_2} + \frac{L_3}{a_3} + \ldots\ldots + \frac{L_n}{a_n}}} \qquad (4.5)$$

where L = total penstock length, a_{equi} = equivalent wave velocity, L_i and a_i are the length and wave propagation velocities of the individual sections.

$$\boxed{A_{equi} = \frac{L}{\frac{L_1}{A_1} + \frac{L_2}{A_2} + \frac{L_3}{A_3} + \ldots\ldots + \frac{L_n}{A_n}}} \qquad (4.6)$$

where A_{equi} = equivalent cross sectional area of the penstock and L_i and A_i are the length and cross sections of the individual pipe sections of different diameter.

4.4.6 Development of the Graphical Method

From the waterhammer computations for valve closure/opening (see Appendix A), we know that the reflexion time T_r must be compared with the closing time of the valve which, in the case of load rejection, corresponds to the effective acceleration time of the PAT (formula 4.3). If acceleration of the PAT/generator unit is completed within one period, i.e before the pressure wave arrives back at the PAT, the full pressure rise according to Joukowsky's law occurs:

$$\boxed{T_{a\,eff} < T_r} \qquad \boxed{\Delta h_{max} = a\frac{\Delta v}{g} = a\frac{\Delta Q}{g\,A}} \qquad (4.7)$$

Note that in the case of $T_{aeff} < T_r$, the wave velocity a and cross-sectional area A corresponding to the pipe at the PAT must be used. **The equivalent values a_{equi} and A_{equi} should only be used for PAT acceleration times T_{aeff} appreciably greater than one reflection time T_r (over four times T_r).**

Maximum speed and pressure for a given installation are found graphically by plotting the straight line of the Joukowsky's law into the performance curve of the selected PAT (see Figure 4.14 below).

Proceed as follows:

1. Assume a convenient change of flow ΔQ and calculate the corresponding increase/decrease of pressure: $\Delta h = a * \Delta Q / (g * A)$. Note that the change of flow could either be negative or positive, depending on whether flow will increase or decrease when the PAT accelerates to runaway speed (see also section 3.6.1, Figure 3.31).

2. Plot the resulting triangle (ΔQ and Δh) into the performance curve. The intersection with the no-load line of the PAT performance curve (T = 0) indicates the maximum pressure head H_{max} after full load rejection;

3. Applying the affinity laws on two points of the no-load line, we obtain the corresponding maximum PAT speed n_{Rmax}. Since we know the speed (n_R) for the steady state runaway point (H_R / Q_R) from Formula 3.28 above, we can write:

$$\boxed{n_{Rmax} = \sqrt{\frac{H_{max}}{H_R}}\; n_R} \qquad (4.8)$$

Note that the affinity laws in this form can only be applied on two operating conditions having the same torque. Starting from the initial operating point (H_o / Q_o) to calculate the speed corresponding to the maximum head (H_{max}) would be wrong.

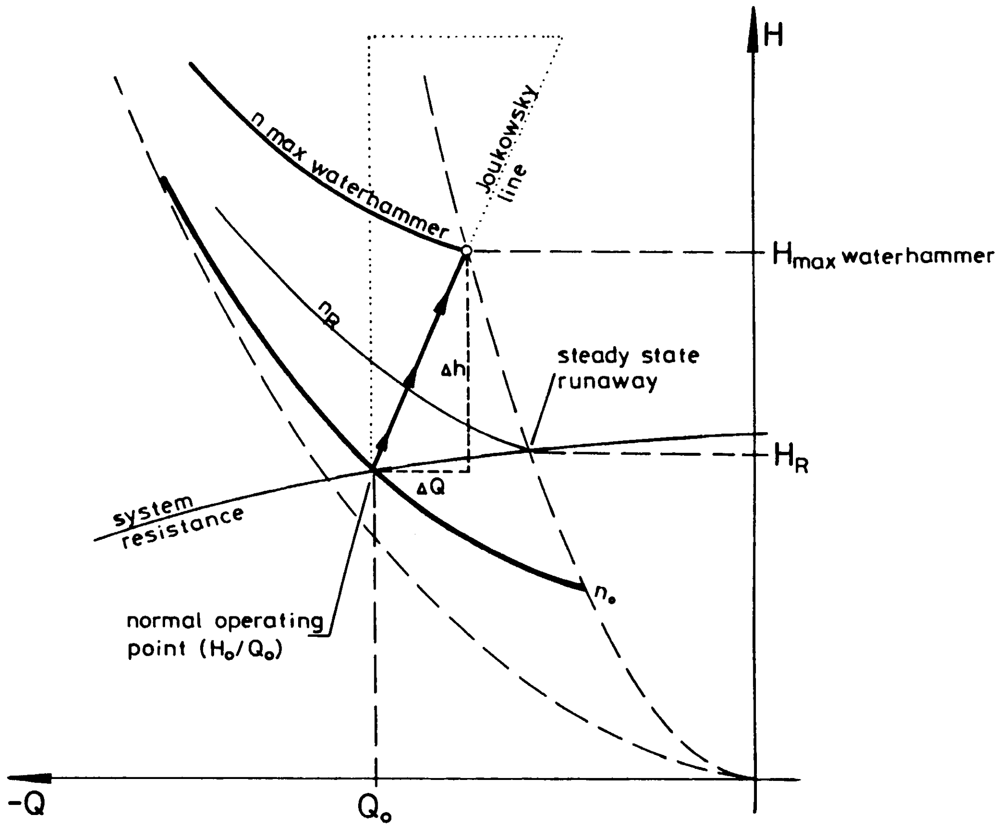


FIGURE 4.14 :
Maximum waterhammer for a PAT acceleration time smaller than the reflexion time of the penstock pressure wave.

This worst case only occurs if the penstock is very long or if the inertia of the rotating masses of the generating unit are very small (PAT and induction motor as generator).

If the acceleration time of the PAT is greater than one period or reflexion time Tr of the penstock, the pressure rise at the machine does not attain the maximum values as found in Figure 4.14 (Joukowsky's equation). The influence of the pressure wave arriving from the forebay/surge tank after one period inhibits maximum pressure build-up at the PAT. As a rough estimate, we may use the same approach as presented in Appendix A for the gradual gate closure/opening of a valve: maximum pressure according to

Joukowsky's equation is proportionally reduced by the ratio of the PAT acceleration time and the penstock reflexion time:

$$\boxed{T_{a\,eff} > T_r} \qquad \boxed{\Delta h = \Delta h_{max} \frac{T_r}{T_{a\,eff}}} \tag{4.9}$$

where $\Delta h_{max} = H_{max} - H_R$ (see Figure 4.15)

$$\boxed{H_{max}' = H_R + \Delta h} \tag{4.10}$$

$$\boxed{n_{R max}' = \sqrt{\frac{H_{max}'}{H_R}}\; n_R} \tag{4.11}$$

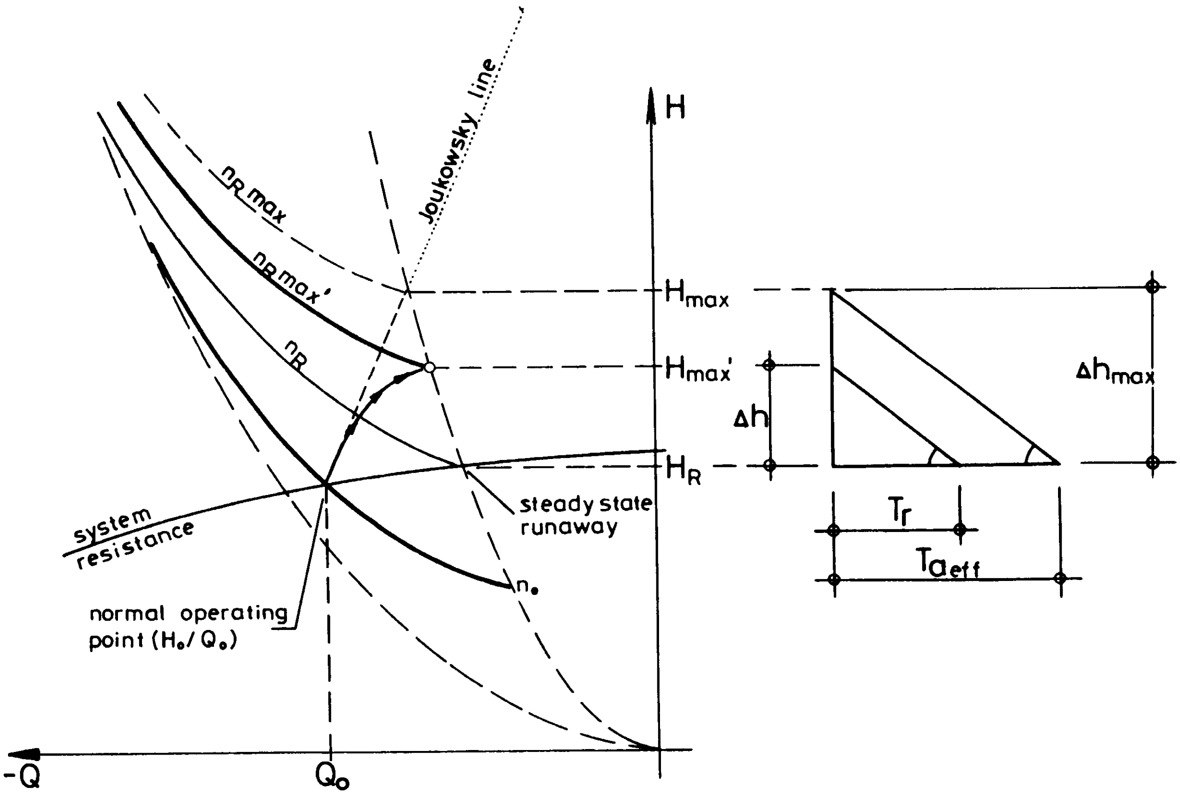


FIGURE 4.15 :
Waterhammer due to load rejection with PAT acceleration time longer than wave reflexion time T_r

This estimate is usually sufficiently accurate to verify whether waterhammer and maximum PAT speed is likely to be a problem. Considering the uncertainties involved in the conversion of pump data into turbine-mode performance (see Chapter 3), which forms the basis of this graphical method, a more detailed analysis does in many cases not make sense: no method can be more accurate than the data which the method was based upon.

If a measured turbine-mode characteristic of the selected PAT is available, the graphical method briefly introduced here may be further developed to obtain not only a more accurate maximum pressure at the PAT but also pressure values at intermediate points of the penstock which allows to determine minimum wall thickness of the penstock pipes at each section (see e.g. Stepanoff, Centrifugal and Axial Flow Pumps, John Wiley and Sons).

4.4.7 Means to Reduce Waterhammer

The pressure rise due to valve closure and load rejection of a PAT may be reduced by one or several of the following means (see also Figure 4.16 below):

□ **Penstock parameters**

1. Designing pipe systems with low original velocities i.e. increase nominal bore (NB) of the pipe (refer to Figure A7 in Appendix A to remain within economically acceptable limits).

2. Using another pipe material to reduce wave propagation speed "a" (plastic instead of steel).

3. Inserting a surge tank or stand pipe on the penstock to reduce penstock length and thus reflexion time T_r. The source (intake or storage basin) is connected to the surge tank (or stand pipe for micro-hydro) by a nearly horizontal low-pressure, low velocity pipe. For the waterhammer calculations, this stand pipe may be considered as a constant pressure basin such as the forebay before inserting the tank. Note that the maximum and minimum water elevations in the stand pipe and its horizontal cross-sectional area are determined by the conduit <u>upstream</u> of the stand pipe and not by the penstock. For stand pipes of uniform cross-sectional area, the following formulae may be used to check whether a stand pipe or surge tank might be a valid option to reduce waterhammer:

For instantaneous valve closure/opening according to Thoma:

maximum (+) and minimum (-) water levels in stand pipe:

$$+z = v_0 \sqrt{\frac{L_0 \; A_0}{g \; A_S}} - \frac{h_0}{2} \qquad -z = -v_0 \sqrt{\frac{L_0 \; A_0}{g \; A_S}} - \frac{h_0}{2} \tag{4.12}$$

minimum horizontal cross-sectional area of stand pipe / surge tank:

$$A_{s\,min} = L_0 \; A_0 \; \frac{v_0^{\;2}}{2g \; h_0 \; (H_{gross} - h_0)} \tag{4.13}$$

H_{gross} = gross head in m
L_o = length of power conduit in m
A_o = cross-sectional area of power conduit in m^2
A_s = horizontal cross-sectional area of stand pipe / surge tank in m^2
v_o = velocity in power conduit at full flow in m/s
h_o = head loss in power conduit at nominal flow in m (approx. $h_o = h_{friction} + v_o^2/(2g)$)
see also Figure 4.16 below

□ **Machine and equipment parameters**

4. Increasing the PAT acceleration time T_a by installing a flywheel (increase of inertia of rotating elements (details of calculation see Appendix F)

o **Waterhammer due to valve closure/opening**

5. Longer closing/opening times of control valve either by compulsory instructions for operators or adjustment of the spindle gearing of the valve.

o **Special measures**

(more sophisticated and therefore sensitive to the quality of maintenance affecting the reliability of the system)

6. Installation of an air vessel to damp pressure surges (high pressure air tank bears some danger)

7. Installation of a by-pass around the PAT incorporating a pressure relief valve (Clayton-type or similar water actuated servo-valve). At a certain pressure increase caused by waterhammer, the pressure relief valve opens the by-pass conduit and the flowing water is not further retarded but led around the obstacle (PAT impeller accelerating to run-away speed).

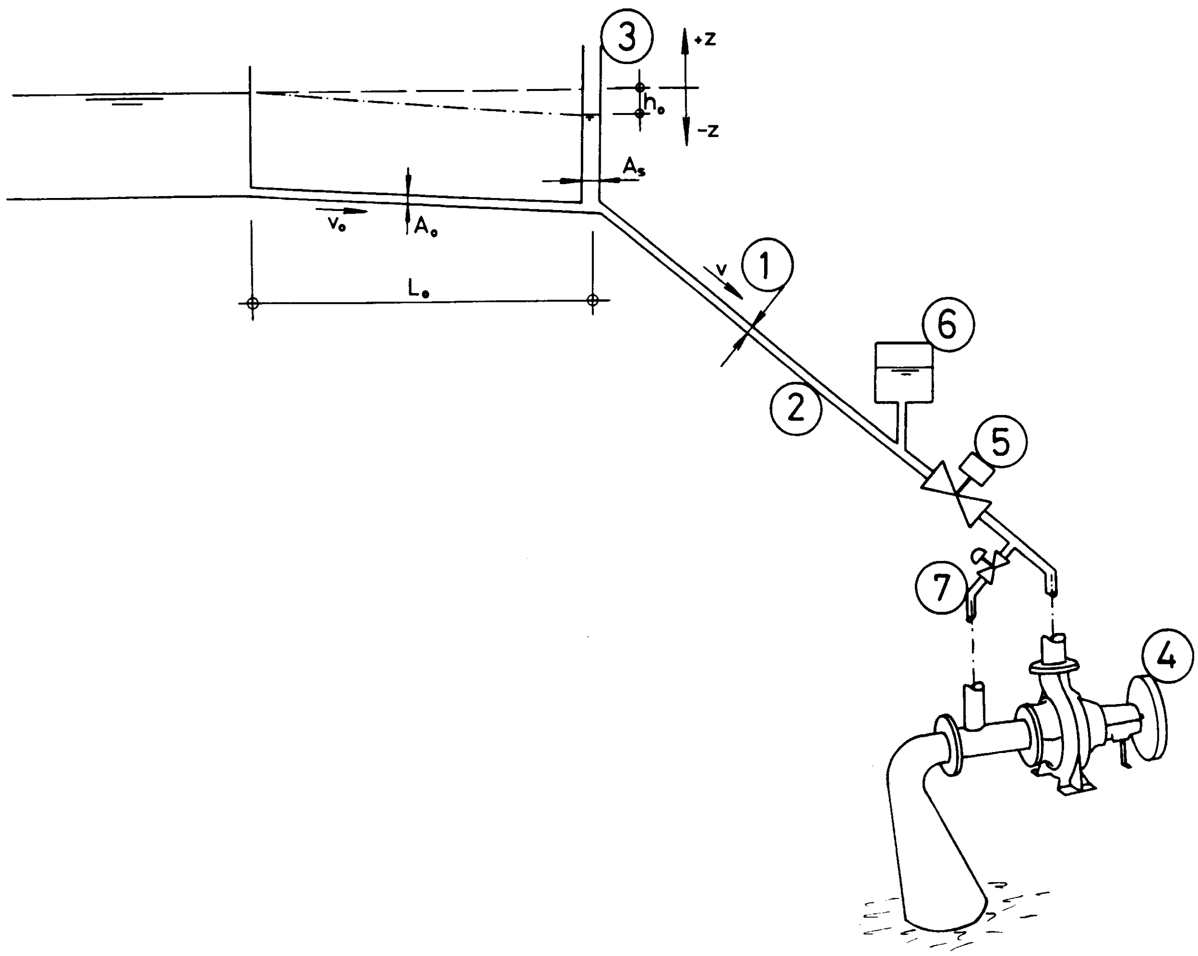


FIGURE 4.16 :
Means to reduce waterhammer

4.5 Measuring PAT Performance in the Field

4.5.1 General

Since the selection of a PAT is not as easy as selecting a pump or other standard machines (see Chapter 3 above: State of the Art) measurement of PAT performance in the field (or in a laboratory if more convenient) should be foreseen for all installations using PATs. In some cases, it might even be required by contract. The results of the measurements may help both designer/supplier as well as purchaser of the PAT to arrive at a more objective judgement in case the installation does not perform as expected. However, it need not necessarily be a faulty installation which demands the measurement of PAT performance. Three general cases may be broadly distinguished:

- **Commissioning of a PAT:**

  Even if no contractual guarantees are involved, the actual performance of a PAT should be determined soon after putting it into operation:

  - to verify whether PAT power output meets the requirements of the planned end users or whether adaptations must be made;

  - to provide the (local) design engineer with a feedback on the accuracy of his design and selection calculations especially in view of further installations using the same or a similar machine.

- **Field Acceptance Test:**

  The supply contract between the purchaser and the PAT manufacturer might include a guaranteed performance of the machine (power output, efficiency, flow). The fulfilment of these guarantees can be verified at the actual site (in situ test) after the installation of the PAT.

- **Trouble Shooting:**

  The PAT may have been in operation for some time when suddenly or gradually the required power is no longer delivered.

All three occasions require **field measurements** of varying accuracy and procedures. Rules, methods and equipment for acceptance tests of hydraulic turbines are described in the IEC (International Electrotechnical Commission) Code, Publication No. 41. Publication No. 41 was written mainly in view of field acceptance tests of large turbines; it may be used for micro-hydro only as a general guide-line.

The ISO Standard No. 2548 (Acceptance tests for pumps, Class C) allowing an overall error of measurement of 5 % on pump efficiency, might also be used as a reference for micro-hydropower installations using a PAT.
In the following, we will give some general hints on suitable measuring equipment required to conduct field tests for MHPs. In any case, it is recommended that **the measuring equipment and devices be planned already during the design stage.**

4.5.2 Parameters to be Measured and Analysis of Tests

To determine PAT performance, the following parameters should be measured:

- net head H_t at the PAT (m)

- flow Q (m3/s)

- shaft power P (W)

- rotational speed (rpm)

If all these values can be measured, the efficiency of the PAT may be calculated as the ratio of shaft power to hydraulic power:

$$\boxed{\eta_t = \frac{P_t}{\rho\, g\, H_t\, Q_t}} \qquad (4.14)$$

Rotational speed indicates whether the PAT operates at nominal speed; if test speed deviates from nominal speed, head, flow and power shall be translated to the specified speed using affinity laws (see Appendix A).

The calculated efficiency is compared with either:

- PAT efficiency (turbine mode) indicated by the manufacturer based on actual tests in turbine mode, or

- efficiency in pump mode from pump performance curves; the difference between turbine and pump-mode efficiency is usually in the order of ± 5% but may reach -10 % and more (see Figure 3.23 above).

When fixing a guaranteed PAT efficiency, allowance should be made for the inevitable measuring errors which are usually higher in the field than in a laboratory.

For direct-driven machinery it is difficult to determine PAT shaft power in the field. In such cases, the PAT output is usually considered to comply with the requirements stated in the supply contract if the machine operates smoothly at its nominal throughput (e.g. kg of grains per hour etc.). Measuring flow and head of the PAT is then used to detect possible faults (power not delivered; flow absorbed too high, etc) and to indicate possible remedies (see trouble shooting below).

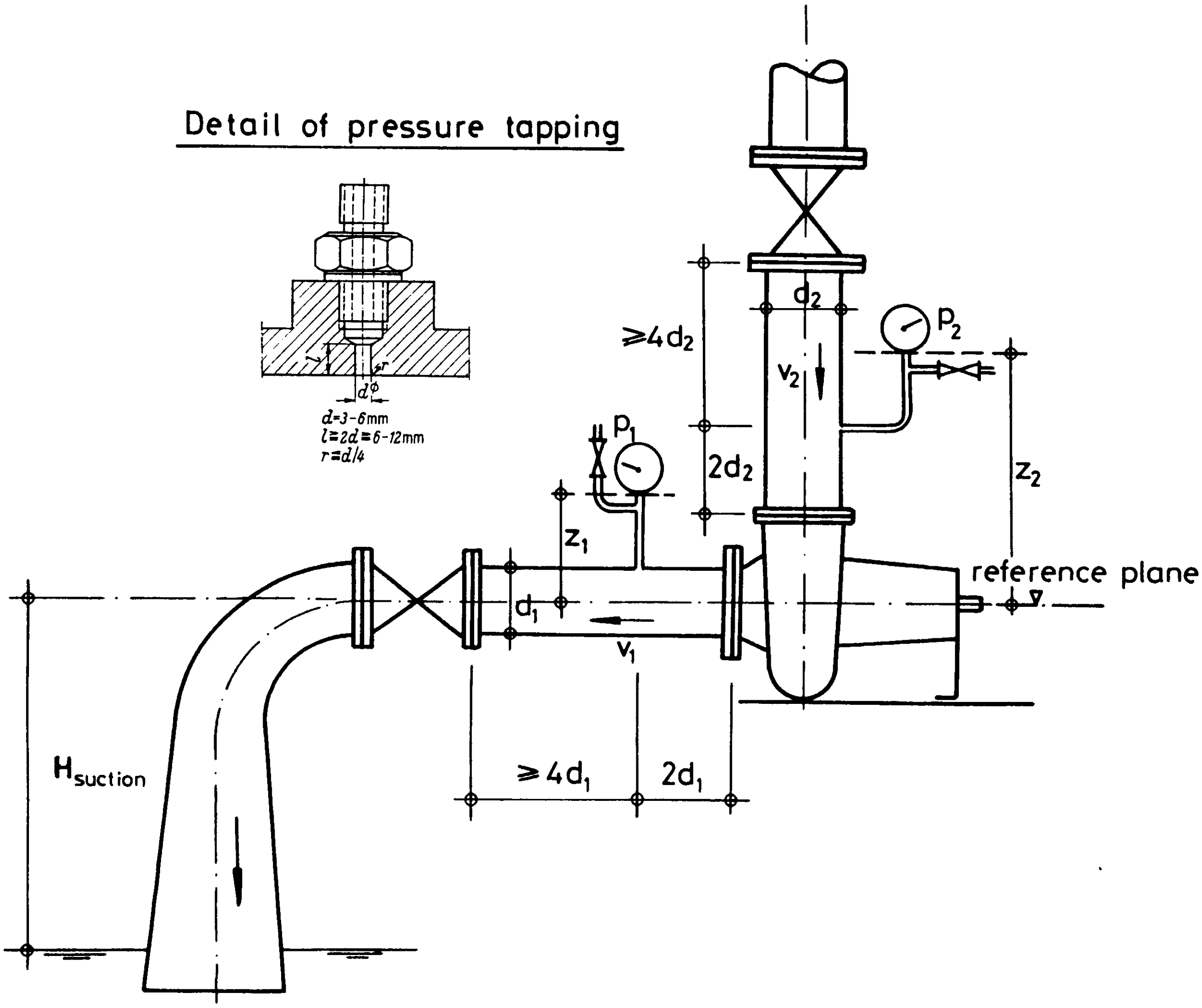
4.5.3 Measuring Techniques and Instruments Required

□ **Pressure Head**

Measuring net head on the machine should be done with calibrated pressure gauges whenever possible. Topographical measurement of gross head (difference in elevation of free water levels of forebay and tail race) and head loss calculation is possible but does not serve as a permanent control device which might especially be useful to detect pressure drops due to obstacles blocking the penstock or PAT inlet.

Bourdon-type pressure gauges have proven to be most convenient for field measurements though recalibration (e.g. after exposure to excess pressure) is necessary from time to time. Although the concept of MHPs using PATs aims at keeping costs as low as possible, the economic advantage of using cheap pressure gauges is seldom justified. Even large measuring efforts will fail to arrive at a valid statement on actual PAT performance if instruments of insufficient precision and reliability are used [accuracy should not be inferior to class 1 (absolute error of 1 % of full scale)]. Therefore, pressure gauges with glycerine filling and/or corrosion-free materials should be chosen which will ensure long-term trouble-free measurements.

Figure 4. 17 shows the arrangement of Bourdon-type pressure gauges on a PAT. Note that the position of the pressure tappings should be two pipe diameters away from the PAT inlet and outlet flanges and no closer than four pipe diameters to a valve or other disturbances (however, at right angles of the plane of a bend is allowed). Pressure tappings should conform to the detail given below and must not be chamfered or contain burrs at the liquid side. In some installations, the gauge on the low pressure side might be omitted and the suction head be measured directly as a difference of elevation (and accounting for losses in the draft tube analytically).



a) Pressure reading p_1 < atmospheric (negative reading, pressure gauge pipe vented - containing air): Turbine net head $H_t = z_2 - d_1/2 + (p_2 - p_1)/(\rho *g) + (v_2^2 - v_1^2)/(2g)$

p_1 and p_2 in N/m2 (1 bar = 10^5 N/m2 ; 1 kp/cm^2 = 98100 N/m2)

b) Pressure reading p_2 > atmospheric (pressure gauge pipe containing liquid):

Turbine net head $H_t = z_2 - z_1 + (p_2 - p_1)/(\rho *g) + (v_2^2 - v_1^2)/(2g)$

FIGURE 4.17 :
Determining net turbine head by means of pressure gauges

□ **Flow**

From the point of view of accuracy, flow is the most difficult of all parameters to be measured. For conventional micro-hydropower installations, only one option is considered to be appropriate and to give sufficiently accurate results: standard rectangular or V-notch weirs built in the tail race or at the forebay/sand trap are reliable, inexpensive structures which can be built everywhere. It is however **important to plan these devices already in the early design stage of the civil works.**

Such measuring weirs are usually permanent structures but need not necessarily incur an additional head loss since in many installations, the tail race requires a sill in any case to keep the draft tube outlet submerged at low river discharge. Thus, this sill can easily be transformed into a measuring weir. Details of construction of sharp crested weirs, limits of application and rating tables suitable for MHPs are given in Appendix H.

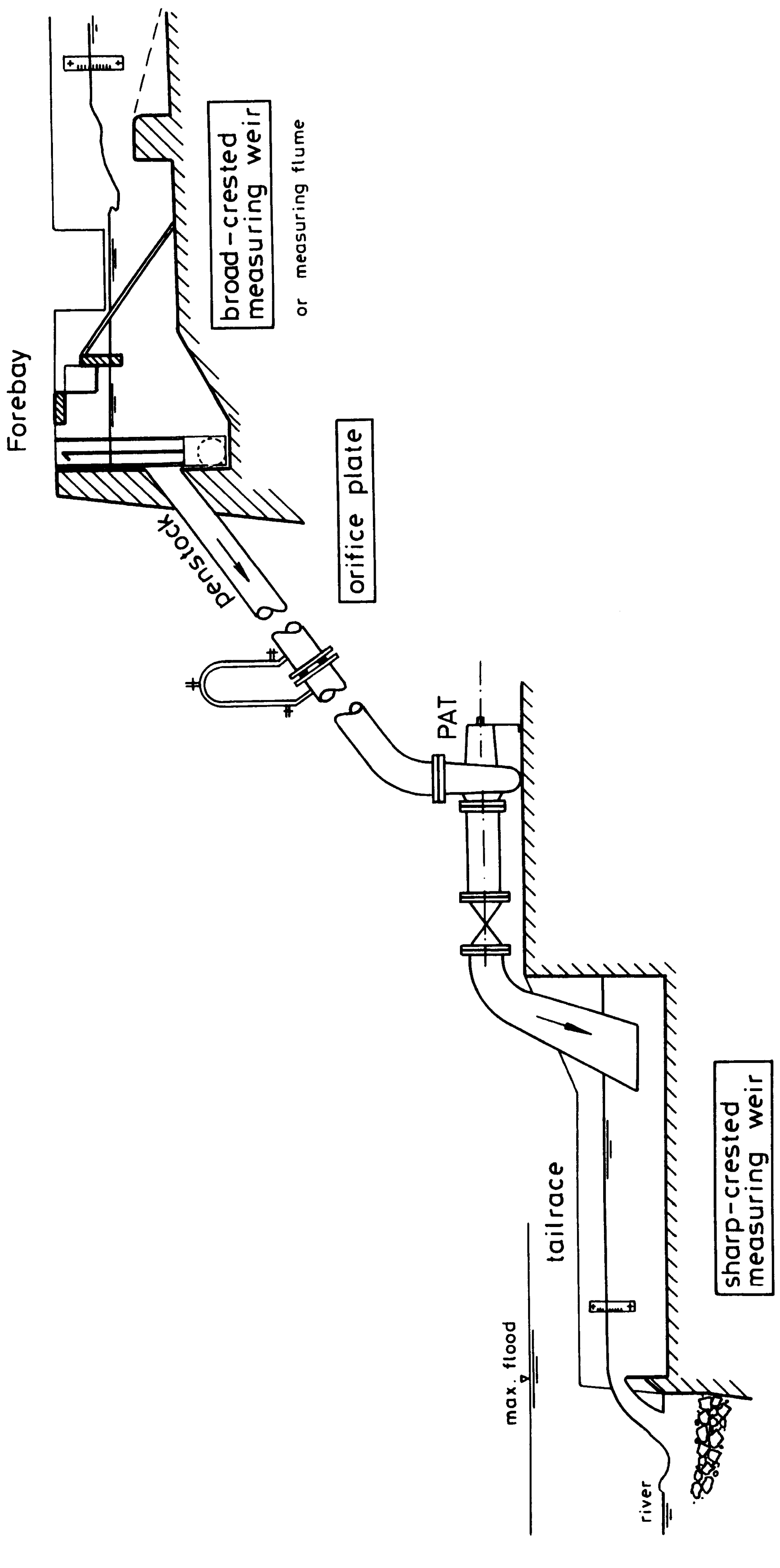


FIGURE 4.18 :
Flow measurement using standard measuring weirs in MHP installations

In installations (e.g. existing plants) where weirs are not possible, an **orifice plate** in the penstock may be used as a measuring device. Flow is measured indirectly; the orifice plate consists of a constriction in the penstock involving head losses which are measured with a differential pressure gauge and then related to the flow through the orifice. Since head losses in the penstock are not desired, the orifice should not be installed permanently but only during the measurements. This means that the flow measured with the orifice does not correspond to the nominal flow for normal PAT operation (another system resistance curve) and a further calculation is necessary. Additionally, the limits of application, such as the minimum length of free straight pipe required up- and downstream of the orifice (up to 20 times pipe diameter), make the use of orifices for micro-hydropower rather impractical. For details of design, layout and calculation of orifice plates see ISO standard No. 5167.

Other methods of flow measurement such as velocity measurement with **current meters** or the **salt gulp method** do not yield accuracies better than ±10% in normal field application and are therefore not suitable for efficiency computations of PATs despite their straightforward application. Small flows in water supply systems incorporating a PAT might be measured using (existing) **water meters** or the **volumetric method** using temporary (bucket, barrel) or permanent volumes (storage tank, reservoir) and a simple stop-watch.

- **Rotational Speed**

Permanent measurement of speed is seldom required. Portable tachometers are usually sufficiently accurate. In electricity generation, frequency meters may be used to measure the (synchronous) speed of the generator:

$n_g = 60\, f / p$ where f = frequency in Hz and p = number of pole pairs, n_g = rotational speed of the generator (synchronous) in rpm

- **Shaft Power**

Electricity generating units usually comprise all the necessary meters and devices to calculate power output of the generator. Shaft power can then be determined indirectly. Either of two methods may be applied:

- ***kW-hour meter available***: Determine the meter constant X (= revolutions of the disc per kWh)

Measure the time T in seconds for N revolutions of the disc

Calculate the power output of the generator:

$$\boxed{P_g = \frac{3600\,N}{X\;T}} \qquad P_g \text{ in kW}$$

-Voltage, current and Watt meters available (e.g. portable instruments):

grid-linked electricity generation: reading voltage U, current I and active power P off the instrument leads to the following relationship (for single phase system):

Apparent Power S = U * I

$$\text{Power factor } \cos\phi = \frac{P}{S} = \frac{P}{U\,I}$$

(Note that power factor indications from instruments or the nameplate of the generator are not reliable enough to calculate the power output P accurately from voltage and current measurements only)

in isolated systems: measuring voltage and current on the ballast resistances of the load controller (after disconnecting consumers) or on resistances especially taken to site for measuring purposes:

$P_g = U * I$ (cos ϕ becomes unity, since ballast loads of the load controller are pure resistors)

From generator power we can calculate PAT shaft power using the generator efficiency according to the table in Appendix G or from the specifications of the supplier.

$P_{PAT} = P_g / \eta_g$ and comparing this with hydraulic power $P = \rho * g * Q_t * H_t$ leads to an (approximate) value of PAT efficiency (see also above).

4.6 Trouble Shooting

When a PAT fails to operate or when the power delivered drops, the cause of trouble should be investigated and steps taken to eliminate it. Since a PAT involves a number of uncertainties resulting from the fact that the machine is used in a way it was not explicitly designed for, the field of possible causes might be vaster than for a standard pump. Trouble shooting should therefore be conducted in a systematic approach. Three general fields of causes may be distinguished:

- **trouble or faults in the hydraulic system** (penstock, forebay, control valve)

- **faulty operation of the PAT**

  This may either become apparent on commissioning of the PAT, i.e be related to the hydraulic design and selection of the PAT (section 3.6.3 above deals with this issue and proposes remedial measures) or

  may be caused by mechanical failure or excessive wear (bearing or wearing rings worn etc.) and faulty O&M (stuffing box too tight). The operating instructions of the pump design for pump mode operation should be followed. For specific issues related to PATs (reverse rotation etc.) refer to section 3. 7 above.

- **trouble caused by the driven machinery, generator or end-use appliances**

Investigations should always start with visual inspections and checks of machine temperature (bearing, stuffing box) and lubrication levels. Furthermore, measuring the main operating parameters such as head, flow, rotational speed and power output as indicated above might cut down the possible causes of faults and should be tried before starting to dismantle the machine or auxiliary equipment.

5. ECONOMIC CONSIDERATIONS

5.1 General

A major advantage of a standard pump used as turbine (PAT) is certainly its lower price as compared to a purpose-built conventional turbine. However, the lack of hydraulic controls (adjustable guide vanes) may reduce the annual energy production in many applications. The design engineer must therefore verify whether the price advantage is not offset by the lower yearly energy output of the PAT.
It would be beyond the scope of this manual to depart on a full-fledged financial or economic analysis of a micro-hydropower plant. On the basis of a simple comparison of annual costs and returns we will show, especially to those not being familiar with economic calculations, how the selection criterion ECONOMY may be established approximately. For more detailed studies we recommend the GTZ publication No. 163 by *H. Fink and G. Oelert:* A Guide to the Financial Evaluation of Investment Projects in Energy Supply.

5.2 Assumptions

We assume that project implementation has been justified previously in a feasibility study, i.e. it has been shown that hydropower is the most economic solution to supply the energy for a specific purpose (agro-processing, electrification, etc.). Thus, the design engineer we are addressing here is not required to choose between hydropower and an alternative source of energy (e.g. diesel gen-set) but to select the most suitable hydraulic machine (technical optimisation), in our case a PAT or a conventional turbine. It is therefore acceptable to use an economic calculation which does not take into account indirect (or secondary) costs and benefits created outside the project itself such as socio-economic (e.g., changes on the local labour and financial market due to the occurrence of the new energy source) and environmental effects (e.g. reduced deforestation due to substituting firewood or changing river discharge regimes). In short, our objective is the **financial analysis** or the direct benefit of the plant owner (electricity producer, mill owner, etc) and not the national benefit (usually referred to as economic analysis).

In the following, we propose a **static** method of financial viability calculation which does not consider WHEN investments or returns occur but expresses all costs and benefits in **annual average values**. This method provides sufficiently accurate results as long as the service lives of the project components to be compared are similar, which is certainly the case for PATs and turbines. Another precondition of this method is that the returns on produced energy should remain constant over the project duration.

5.3 Identifying Project Costs and Benefits

5.3.1 Costs

Although some of the components of an MHP might remain the same (both layout and costs) regardless of the choice of the hydraulic machine, we nevertheless recommend that **all direct costs** of the project be established and included in the comparison of the two alternatives PAT - conventional turbine. In this way, the calculated unit production costs (e.g. currency/kWh) are real values (and not only relative machine costs) and may be compared with the figures of the feasibility study (if available), masterplan estimates or actual energy prices of the region (e.g., from national electricity board).

- **Capital costs / Investment costs**

In a first step, all capital cost items of a project must be identified and grouped together according to their expected service lives. Service life is the time during which an investment facility can be used economically and it is usually equal or shorter than the technical lifetime. Depreciation (periodic reduction of value due to wear and tear not covered by maintenance) of each item is calculated based on the service life which then becomes equal to the recovery period, i.e. the time during which the plant must generate the income to pay back the investments made. Services such as project planning and engineering design must be included. The following table gives some general indications on service lives used in micro-hydropower installations. Financing agencies sometimes demand shorter recovery periods corresponding to the credit duration. This will increase unit production costs and will consequently affect the feasibility of a plant.

Cost Item	**Service Life (years)**
Civil works (intake, canal, power-house)	15 to 25
Generating equipment (PAT, turbine, generator, governor)	10 to 20
Penstock	15 to 25
Distribution network	
Medium to high voltage incl. step-down transformer	20 to 25
Low voltage system	10 to 20
Project planning, engineering design, land acquisition, survey	15 to 25
Other fixed assets	10

Investment costs should be calculated according to the selected machines (locally manufactured or imported), construction methods and materials to be used and the price level (wages) prevailing in the region. To arrive at relatively reliable results of the financial analysis, capital costs must be determined using the usual methods (bill of quantities, contractors' and manufacturers' offers). General indications on the distribution of costs as are sometimes found in literature (e.g. machine costs in % of total investment costs and as a function of power or head) are not precise enough to decide whether a PAT or a conventional turbine should be used.

Each item usually contains a contingency allowance (physical contingency) which allows for adverse conditions during construction adding to the baseline costs.

Subsidies or grants obtained from the government or development agencies will reduce the investment costs. [Note, if grants are available, i.e means without incurring debt services or lost income (due to the tied capital of the plant owner), a yearly depreciation allowance must be considered which will eventually allow reconstruction of the scheme at the end of its lifetime, see O&M costs below].

Each item of investment is converted into equal annual payments (annuities) using the so-called capital recovery factor. This factor permits calculating the equal instalments necessary to repay (amortize) a loan over a given period (usually the service life) at the current market interest rate. The yearly payment is a varying combination of both interest and repayment of capital. The recovery factor (RF) is defined as follows:

$$RF = \frac{i\,(i+1)^n}{(i+1)^n - 1} \qquad (5.1)$$

where i = interest rate (%/100) and n = period of loan (years) or service life

(The recovery factor may also be obtained from the table in Appendix I)

If the project is not financed through loans (external financing) but by internal capital of the owner/organization of the MHP, the same approach is used, i.e. the current interest rate may be applied. This assumes that the investor could invest his capital on an alternative (the next best) use thereby achieving the same (market) interest .

The economic calculation assumes that today's prices will remain constant. This seems not to be true, certainly not in many developing countries where inflation rates are high. Additionally, the market interest rate is determined not only in view of a return on capital and the risks involved in borrowing but also allows for the loss of purchasing power due to **inflation**. If we correct the market interest rate with the inflation rate, the increase of the general price level is eliminated and a constant price level over the whole project period may be applied. The corrected or actual interest rate writes as follows:

$$i^* = \frac{(i+1)}{(a+1)} - 1 \qquad (5.2)$$

where i = market interest rate (%/100) and a = inflation rate (%/100)

(Due to the uncertainties involved in the determination of these rates, many projects use the simpler formula $i^* = i - a$, which is not far from the correct value for low interest and inflation rates). Note, if the actual interest rate i^* is used, no price contingencies for inflation should be made on the capital costs.

- **Operation and Maintenance (O&M)**

Operation and Maintenance usually comprises the following items:

Cost Item	*Description*	*Yearly Costs in % of capital costs*
Personnel	<u>**Operator**</u>: Supervision of operation, preventive maintenance and inspection <u>**Technicians, engineers**</u>: General supervision, inspection and repairs in case of failures <u>**Administration**</u> (General manager, secretary) <u>**Fitters & Labourers:**</u> Inspection and preventive maintenance of distribution network, Replacement of faulty equipment <u>**Labourers:**</u> Repainting steel structures Desilting canals, forebay, storage pond	according to local conditions and wage levels
Office rents, office supplies and administrative costs	(without manpower)	according to local conditions
Electro-mechanical Equipment (PAT/Turbine, generator, governor)	Spare parts for the equipment, filters, lubricants, etc. (costs of material only)	0.6 - 2 % of electro-mechanical investment costs (covering costs of material only)
Civil Works (Intake, sand trap, canal, penstock, gates, trashrack, power-house)	Material for refilling of cavities, concrete for relining canals and intake works, paint for all steel structures, etc.	0.2 - 1 % of capital costs of civil works (covering costs of material only)
Distribution Network	Fuses, insulators, etc.	
Contingencies for major breakdowns	Flooding of power-house, intake etc.	in industrialized countries usually covered by insurance (yearly premium to be included)
Reserve for upgrading equipment	Replacing manual control with governor, etc.	depending on the technological level of the initial design
Depreciation allowance (only if investment costs fully or partly covered by subsidies and grants)		
Taxes and water rights, other duties		according to local rules and regulations

As a general rule, total costs of O&M without major replacements account for approximately 3 to 4 % of capital costs for micro-hydropower installations (using low wage levels without foreign personnel / mechanics involved).

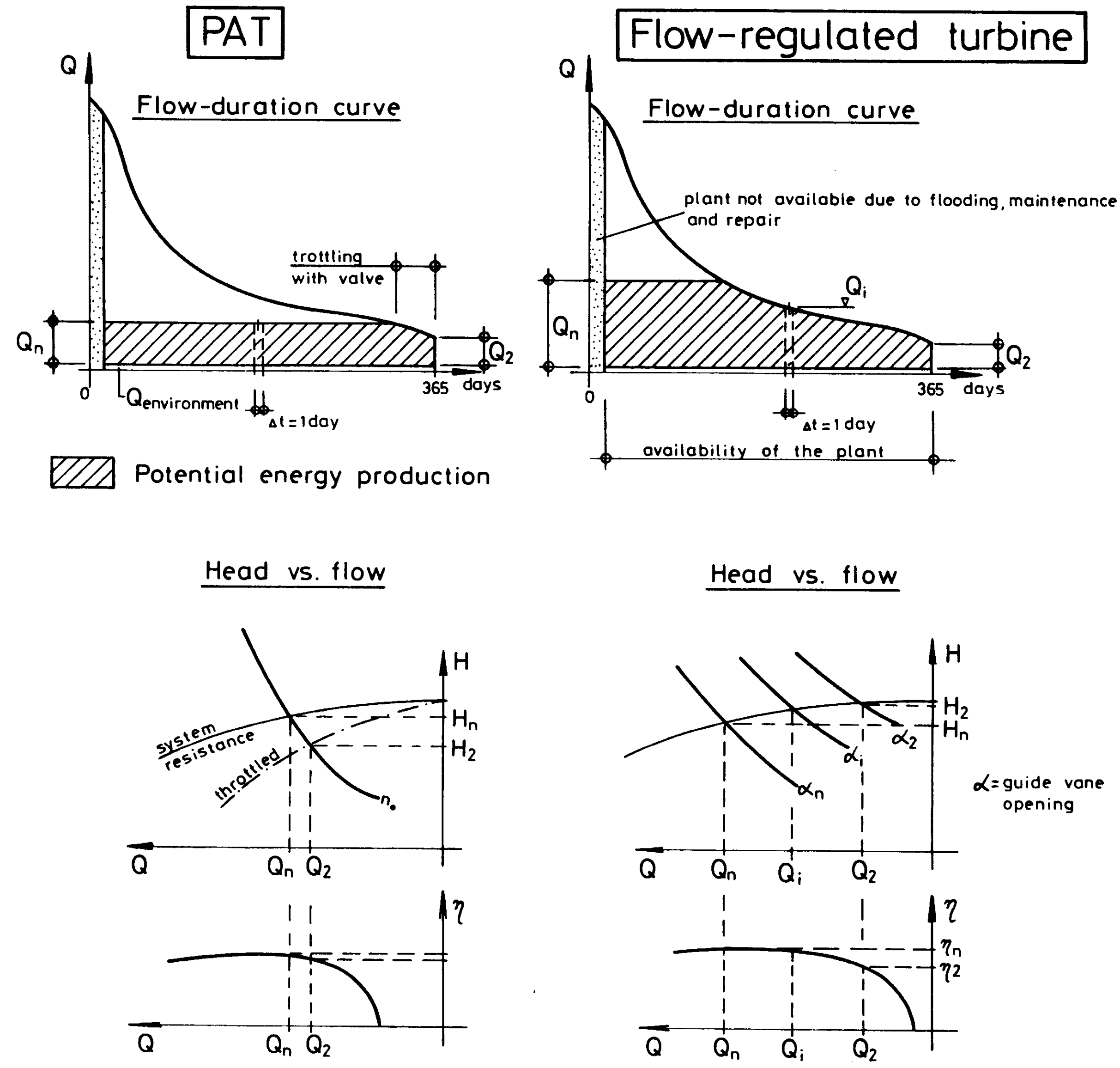
O&M costs are also estimated on a yearly basis (yearly average over the whole service life or recovery period). Together with the annual instalments for capital costs, <u>**total annual costs**</u> of the MHP is obtained.

$$\boxed{Ak = K_o + (I_o - L)\ RF + L\ i^*} \qquad (5.3)$$

Ak = total annual costs
K_o = O&M costs per year
I_o = Investment costs
L = Liquidation value of installations at the end of the assumed service life (recovery period); for micro-hydropower components with service lives over 15 years usually zero.
RF = recovery factor according to Appendix I or Formula 5.1 (as a function of the recovery period and the interest rate i^*)
i^* = actual interest rate (market interest rate corrected with inflation rate according to formula (5.2) above)

5.3.2 Income

The potential yearly energy production may be determined using the available streamflow (duration curve) and the nominal flow (layout flow) of the PAT or the conventional turbine (see also section 3.3.1).



$$E_{pot_t} = \rho g \sum^{year} [(Q_i - Q_{env.}) H_i \eta_i \Delta t]$$

$$E_{pot_{plant}} = E_{pot_t} \eta_{gen} \eta_{trans}$$

t = PAT / turbine
gen = generator
trans = transmission

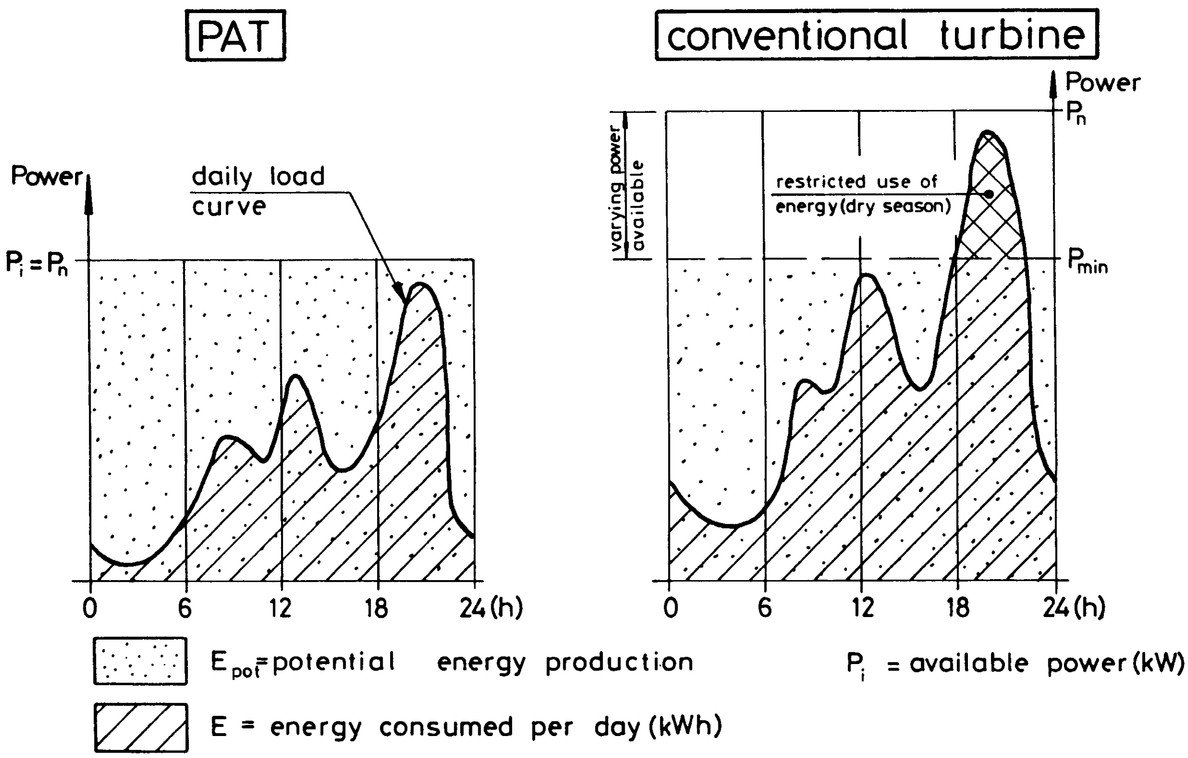
FIGURE 5.1 :
Potential energy production per year for PAT and flow regulated turbine

For the calculation of the part-flow energy production, it is important to take account of the part load efficiency which is relatively low for the PAT (using a throttle valve); on the other hand, a good turbine (Francis, Pelton) might yield good efficiencies down to 50 % or less of nominal flow.

Not all the energy that can be produced will serve a productive purpose (except in grid-linked electricity generation). End-use of hydropower, be it direct-driven machinery or stand-alone electricity generation, will always fluctuate between a minimum and a peak demand. Since income is only generated from the productive energy, the cost-benefit comparison must be carried out with the actual consumer load. The relationship between actual and potential energy production/utilization is expressed by the station factor, also known as plant factor (or load factor in interconnected grids of electricity generation) which reads as follows:

$$\text{Stf} = \frac{\text{actual (measured) energy output of the plant}}{\text{potential energy output}} \quad \text{(during one and the same period of time)}$$

The station factor may be calculated from the load curves used for the determination of the nominal output of the plant (see also section 3.3.3).



Daily station factor $StF_d = \frac{E}{P_i \; 24}$

Annual mean station factor $StF_a = \frac{1}{365} \sum^{\text{year}} StF_d$

FIGURE 5.2 :
Load curves for a stand-alone electricity generating unit and determination of the station factor using a PAT or a conventional turbine

Station factors for small hydropower installations are usually in the order of 10 to 30 %, i.e. the plant capacity is not well utilized; this is because the capacity of a plant is designed to accommodate peak demand which might occur only during a very short period of time (see also below).

Actual energy production per year therefore becomes:

$$\boxed{E_{output} = E_{pot} * StF} \quad (5.4)$$

5.3.2 Comparing Yearly Income and Costs

The last stage of calculation is now the comparison of costs and incomes. This may be done in two different ways:

a) **Calculating the unit production costs:**

$$\boxed{\text{Unit production costs} = \frac{Ak}{E_{output}}} \quad (5.5)$$

where A_k = total annual costs (see formula (5.3) above

E_{output} = energy output per year

If the PAT or the conventional turbine drives machinery directly, the total yearly throughput of the processed products (amount of grains, timber, pulp, etc.) may be used instead of output of energy like in electricity generation.

The unit production costs represent the minimum price per kWh or per unit of processed goods which the plant owner must charge to its customers if costs and benefits are to break even at the end of the service life of the installation, or in other words, if the MHP installation should be economically viable. From this general statement we may deduce that **the economically most favourable solution would be the alternative (PAT or conventional turbine) yielding the lowest unit production costs.** Note that the criterion concerning the economy of an alternative solution, is only **one parameter among others** which must be considered when selecting a PAT or a conventional turbine (see also section 2.1).

b) **Using the current price of energy** ($ / kWh or $ / unit of processed goods) and calculating annual income from energy sales:

$$\boxed{\text{Annual income} = (E_{output} / a) * \text{current price}} \quad (5.6)$$

Comparing annual costs and income leads to the annual return generated by the plant :

$$\boxed{\text{Annual return} = (\text{Annual Income}) - Ak} \quad (5.7)$$

The alternative yielding the higher annual return should be chosen. A negative return means that the plant is economically not viable.

5.4. Improving Economic Viability by Energy Management

From the above, it becomes obvious that any improvement of the economic viability of an MHP plant, be it equipped with a PAT or a conventional turbine, must first aim at increasing the station factor (load factor). This is usually referred to as energy management and means in practical terms that the load curve is flattened, i.e., lower peak demand relative to average demand. (Note that these considerations do not apply to grid-linked electricity generation.)

Several methods might be considered:

- For plants with direct-driven machinery running only during the working hours, electricity generation during the evening hours for domestic use or the neighbouring houses might improve the station factor; however, additional costs are involved for generating equipment and distribution lines.

- Promoting the use of energy (electricity in stand-alone generation) during the off-peak hours by differentiating the tariff:

  - decreasing rates for off-peak energy use (day - night tariff); this requires metering in many cases which involves additional costs

  - differentiating the tariff according to different end-uses: industries guaranteeing continuously high demand are favoured as compared to domestic uses with short peaks

  - introducing a double tariff: one for actual energy consumption and a second rate for peak power demand (rate increasing with power); in this way, consumers avoid the simultaneous use of many appliances, hence they try to flatten the load curve themselves.

- Cutting out appliances during peak hours to provide the necessary power for short-time users: e.g. refrigerators, boilers are cut off in the evening peak hours in favour of TV sets, radios, lighting; this may be achieved by separating the lines to different users, double switches (switching on one appliance cuts the other) or controlled by a time switch or manually by the consumers themselves.

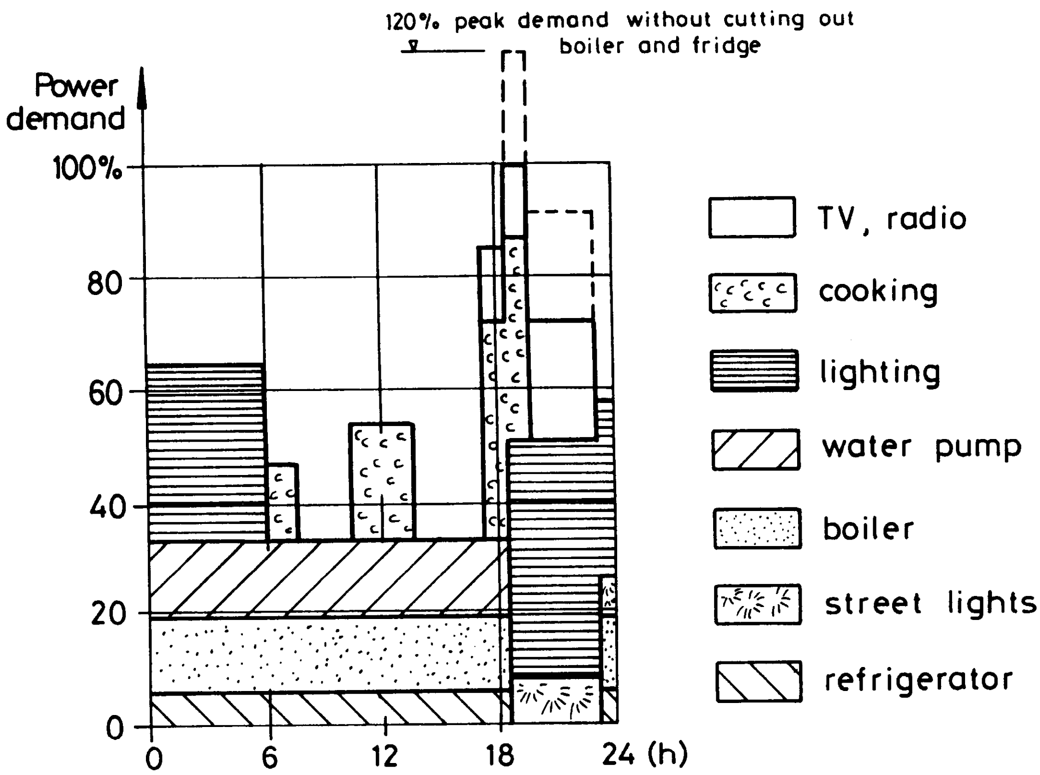


FIGURE 5.3 :
Avoiding peak demand by energy management (cutting out boilers, refrigerators during short peak hours)

6. Enquiry Information

As mentioned in Chapter 3, pump manufacturers will generally charge higher prices for PATs than for a pump of the equivalent design. These higher prices reflect the costs incurred for developing the manufacturer's know-how on PATs and, in case of larger machines, testing and/or modifying the PAT. This may be deemed necessary by the manufacturer to offer the same warranties as for a pump.
It might therefore be more economical to order a standard pump instead of a PAT. In this case, it is, however, the MHP design engineer who will have to carry the risk of faulty operation or damage due to the use of the standard machine in reverse, i.e. as a turbine. Nevertheless, it is essential that the design engineer tries to acquire as much information as possible on the selected machine though not directly on turbine mode but at least on the design particulars which will indicate the limits of a specific machine as a PAT. The following form has been compiled to serve as a check list when making enquiries for a pump to be used as a turbine.
When ordering the machine later on, the same items as indicated on the enquiry form should be guaranteed by the manufacturer.

A) Data to be provided by the MHP design engineer

Selected Pump and Site Specific Details

(assuming that the PAT has been selected on the basis of the pump performance curve and the selection procedure presented in this manual)

1. The Pump Selected: Type: ....

   Size: ....Stages :..

   Impeller (trim diameter): ....

2. Liquid to be handled: Water at temperature: ....°C

   Density at pumping temperature : ....kg/m³

   Solid contents : ....low, medium, high

   max. size of particles : ....mm

3. Required Duty Rate of Flow (Q_{np}): ....m³/s

   Total dynamic head (H_{np}): ....m

4. Nominal pump speed required (n_{np}): ....rpm

B) Details on the pump design to be provided by pump manufacturer

5. Max. permissible pressure on the pump:

   a) for continuous operation — suction side: ....bar

   pressure side: ....bar

   b) during transient conditions — suction side: ....bar

   pressure side : ....bar

6. Max. permissible pump speed for continuous operation: ....rpm

7. Runaway speed and corresponding flow of the pump in reverse operation (e.g. after power failure, no check valve between reservoir and pump):

n_R : ....rpm (at nominal pump head H_{np})
Q_R : ....m3/s (at nominal pump head H_{np})

8. Cavitation NPSH$_{req.}$ for the selected impeller trim: ....m at best efficiency point (or enclose curve with NPSH$_{req.}$ versus flow)

9. Reverse rotation (e.g. due to wrong connection of motor terminals)

The selected pump design <u>**excludes**</u> loosening of threads of the rotating parts mounted on the shaft (impeller, shaft sleeves, etc)

Yes.............No.................(Comments):..

10. Design Details: Bearing type: impeller side : ....

motor side : ....

sensitive to reverse rotation: Yes .... No.....................

max. permissible bearing loads:

- impeller side: axial...N

  radial...N

- motor side : axial...N

  radial...N

belt drive possible? No..

Yes:....................max tension : ....N

Could a larger bearing type be incorporated into the bearing housing?

No..

Yes....................what type:...

11. Shaft Seals Type :...

sensitive to reverse rotation No. .... Yes ....(comments)

12. Max. permissible torque on the shaft :...Nm

13. Is the pump suitable for operation with full speed regulation from 0 to max permissible speed (no danger of operation at critical speed)?

No:............. Yes:....................................(comments)

14. Please, provide sectional drawings with full details on dimensions and materials of the pump (including flange dimensions, impeller passages, mounting frames incl. motors etc.)

7. WORKED EXAMPLE

For the layout shown in Section 6.2, Appendix A, a PAT is to be selected.

1. **Design head and flow**

From the preliminary design/layout, the following data are known:

- nominal turbine flow Q_{nt} ....0.100.................... m^3/s
- nominal or net turbine head H_{nt} ....12.60........................ m
- hydraulic power $P = \rho\, g\, Q_{nt}\, H_{nt} = 1.0 * 9.81$ x 0.100 x 12.60......... = ...12.4........ kW

Diagram D1 indicates, what type of PAT would probably suit the application

........mixed-flow PAT...

2. **Specific speed in turbine mode**

a) Assume a PAT nominal speed n_{n_t} = ...1540.................... rpm

Considerations: 1500 rpm is usually a good compromise between PAT size and allowance for runaway speed; if an induction generator/induction motor as generator is to be driven, slip must be added (average values of slip speed see Appendix G)

b) Calculate the specific speed of the installation (turbine mode):

$$n_{qti} = n_t \frac{\sqrt{Q_{ti}}}{H_{ti}^{3/4}} = 1540\ \frac{\sqrt{0.100\ \mathrm{m^3/s}}}{12.60^{\ 3/4}} = ...72.8..........$$

For multi-stage or double flow PATs, relate the design head and flow to a single-stage and single-entry impeller : (i_{st} = number of stages, i_{fl} = number of entries)

$$H_{ti} = \frac{H_{nt}}{i_{st}} = \text{——} = \text{..........—..........}\ \mathrm{m}$$ (single stage)

$$Q_{ti} = \frac{Q_{nt}}{i_{fl}} = \text{——} = \text{..........—..........}\ \mathrm{m^3/s}$$ (single entry)

3. **Pump-mode specific speed**

$$n_{qp} = \frac{n_{qt}}{0.89} = \frac{72.8}{0.89} = ...81.8.....$$

Verify that $n_{q_p} > 15$ otherwise change assumed PAT speed or number of impeller stages/entries.

4. **Pump efficiency**

$Q_{np} \approx Q_{nt} / 1.3 =$ ....0.100....... / 1.3 =.....0.077.........m^3/s

Use diagram D3 to estimate the maximum efficiency of the suitable pumps.

maximum pump efficiency η_p = ...0.80...............

5. **Converting turbine design conditions into pump design conditions**

Enter Diagram D2 with pump specific speed and efficiency; read the conversion factors for head C_H and flow C_Q

C_H = ...1.50.................. ; C_Q = ...1.37....................

Pump design head and flow at proposed turbine speed n_t :

$$H_{np}\,(nt) = \frac{H_{nt}}{C_H} = \frac{12.60}{1.50} = \text{........8.40.......m}$$

$$Q_{np}\,(nt) = \frac{Q_{nt}}{C_H} = \frac{0.100}{1.37} = \text{.......0.073.......}m^3/s$$

6. **Converting pump design conditions at turbine rated speed into pump rated speed**

Standard centrifugal pumps are usually driven by induction motors. Manufacturers therefore indicate the pump performance for speeds lower than synchronous speeds.Use manufacturers brochures to get an idea of what the rated speed of the pump would be.

assumption: np = ...1450................rpm

The affinity laws apply (see also Appendix B):

nominal pump head
at pump speed n_p

$$H_{np\,(n_p)} = H_{np\,(n_t)} \left[\frac{n_p}{n_t}\right]^2 = \text{8.40} \left[\frac{1450}{1540}\right]^2 = \text{7.45 m}$$

nominal pump flow
at pump speed n_p

$$Q_{np\,(n_p)} = Q_{np\,(n_t)} \left[\frac{n_p}{n_t}\right] = \text{0.073} \left[\frac{1450}{1540}\right] = \text{0.069 } m^3/s$$

Select a pump (if necessary recalculate from point 6 with the correct pump speed np)

7. **PAT selection recommendations**

Select a pump with a rated flow (Q_{np}) slightly lower than the value required according to formula above. The PAT will then __operate in the overload range__ (which will avoid an excessive drop in efficiency if the predicted bep does not correspond to the actual bep)

If no standard pump is available for these head/flow conditions at the proposed pump speed, recalculate from point 2 using a different turbine speed or assume a different number of impeller stages or entries.

Selected pump:

Type: ......Mixed-flow pump TS 7-150......

Rated pump head: H_{np} = ...6.65...... m

Rated pump flow: Q_{np} = ...0.075...... m^3/s

Maximum efficiency: $\eta_{p\,max}$ = ...0.76......

8. **Converting the best efficiency point of the selected pump into turbine mode**

a) Calculate the specific speed of the selected pump (best efficiency head and flow related to single-stage and single-entry conditions i)

$$H_{pi} = \frac{H_{np}}{i_{st}} = \frac{\quad}{\quad} = \text{....../......} \, m$$ (single stage)

$$Q_{pi} = \frac{Q_{np}}{i_{fl}} = \frac{\quad}{\quad} = \text{....../......} \, m^3/s$$ (single entry)

$$n_{qpi} = n_p \frac{\sqrt{Q_{pi}}}{H_{pi}^{3/4}} = 1450 \frac{\sqrt{0.075 \; m^3/s}}{6.65^{\;3/4}} = \text{......95.9......}$$

b) Read the conversion factors for head and flow from Diagram D2 using the maximum efficiency of the selected pump.

C_H = ...1.60...... ; C_Q = ...1.43......

Estimate a <u>performance range</u> by including the following scattering factors:

conversion factors for head:

$C_{H\,max} = 1.1 * C_H = 1.1$ x 1.60 ...... = ...1.76......

$C_{H\,min} = 0.9 * C_H = 0.9$ x 1.60 ...... = ...1.44......

conversion factors for flow:

$C_{Q\,max} = 1.075 * C_Q = 1.075$ x 1.43 ...... = ...1.54......

$C_{Q\,min} = 0.925 * C_Q = 0.925$ x 1.43 ...... = ...1.32......

c) Calculate the turbine best efficiency point (bep t) for both maximum and minimum conversion factors

Turbine design head and flow at rated pump speed:

$$H_{nt}\max(n_p) = C_{H\max} H_{np} = 1.76 \times 6.65 = 11.70 \text{ m}$$

$$H_{nt}\min(n_p) = C_{H\min} H_{np} = 1.44 \times 6.65 = 9.58 \text{ m}$$

$$Q_{nt}\max(n_p) = C_{Q\max} Q_{np} = 1.54 \times 0.077 = 0.119 \text{ m}^3/\text{s}$$

$$Q_{nt}\min(n_p) = C_{Q\min} Q_{np} = 1.32 \times 0.077 = 0.102 \text{ m}^3/\text{s}$$

d) Relate these turbine bep conditions to the nominal turbine speed n_t = 1540 rpm

Maximum value:

$$H_{nt\max}(n_t) = H_{nt\max}(n_p)\left[\frac{n_t}{n_p}\right]^2 = 11.70 \left[\frac{1540}{1450}\right]^2 = 13.20 \text{ m}$$

$$Q_{nt\max}(n_t) = Q_{nt\max}(n_p)\left[\frac{n_t}{n_p}\right] = 0.119 \left[\frac{1540}{1450}\right] = 0.126 \text{ m}^3/\text{s}$$

Minimum value:

$$H_{nt\min}(n_t) = H_{nt\min}(n_p)\left[\frac{n_t}{n_p}\right]^2 = 9.58 \left[\frac{1540}{1450}\right]^2 = 10.81 \text{ m}$$

$$Q_{nt\min}(n_t) = Q_{nt\min}(n_p)\left[\frac{n_t}{n_p}\right] = 0.102 \left[\frac{1540}{1450}\right] = 0.108 \text{ m}^3/\text{s}$$

9. **Power output in turbine mode at the best efficiency point**

$$P_{nt\max} = \rho\, g\, Q_{nt\max} H_{nt\max}(\eta_{p\max} - 0.03) =$$

$$P_{nt\max} = 1.0 * 9.81 \times 0.126 * 13.20 \; (0.76 - 0.03) = 11.91 \text{ kW}$$

$$P_{nt\min} = \rho\, g\, Q_{nt\min} H_{nt\min}(\eta_{p\max} - 0.03) =$$

$$P_{nt\min} = 1.0 * 9.81 \times 0.108 * 10.81 \; (0.76 - 0.03) = 8.36 \text{ kW}$$

10. **Predicting PAT performance away from the bep**

From Diagram D4 and D5: (with the specific speed of the selected PAT n_{qp} = ...95.9.............)

For Head versus Flow Curves:
(Diagram D4)

Q_t / Q_{nt}	H_t / H_{nt} (1)
1.2	1.45
1.1	1.22
0.9	0.82
0.8	0.65

For Power versus Flow Curves:
(Diagram D5)

Q_t / Q_{nt}	P_t / P_{nt} (2)
1.2	1.64
1.1	1.32
0.9	0.72
0.8	0.45

For curves with **maximum** values:
Q_{ntmax} = ...0.126........m³/s; H_{ntmax} = ..13.20...........m; P_{ntmax} = ...11.91........kW

point	Q_t =	H_t = H_{ntmax}* factor (1)	P_t = P_{ntmax}*factor (2)
1.2 max	1.2 Q_{ntmax} = 0.151	13.20 x 1.45 = 19.14	11.91 x 1.64 = 19.53
1.1 max	1.1 Q_{ntmax} = 0.139	13.20 x 1.22 = 16.10	11.91 x 1.32 = 15.72
bep max	1.0 Q_{ntmax} = 0.126	13.20 x 1.00 = 13.20	11.91 x 1.00 = 11.91
0.9 max	0.9 Q_{ntmax} = 0.113	13.20 x 0.82 = 10.82	11.91 x 0.72 = 8.58
0.8 max	0.8 Q_{ntmax} = 0.101	13.20 x 0.65 = 8.58	11.91 x 0.45 = 5.36

For curves with **minimum** values:
Q_{ntmin} = ...0.108.........m³/s; H_{ntmin} = ...10.81.............m; P_{ntmin} = ...8.36...........kW

point	Q_t =	H_t = H_{ntmin}* factor (1)	P_t = P_{ntmin}*factor (2)
1.2 min	1.2 Q_{ntmin} = 0.130	10.81 x 1.45 = 15.67	8.36 x 1.64 = 13.71
1.1 min	1.1 Q_{ntmin} = 0.119	10.81 x 1.22 = 13.19	8.36 x 1.32 = 11.04
bep min	1.0 Q_{ntmin} = 0.108	10.81 x 1.00 = 10.81	8.36 x 1.00 = 8.36
0.9 min	0.9 Q_{ntmin} = 0.097	10.81 x 0.82 = 8.86	8.36 x 0.72 = 6.02
0.8 min	0.8 Q_{ntmin} = 0.086	10.81 x 0.65 = 7.03	8.36 x 0.45 = 3.76

11. **Plot the turbine performance curves for head and power** .

The intersection points of the system resistance curve(s) (see Appendix A for calculation) with the predicted PAT performance curves (maximum and minimum values) provide the possible PAT operating range:

Q_{efft} max = 0.119 m^3/s	H_{efft} max = 12.0 m	P_{efft} max = 10.2 kW
Q_{efft} min = 0.110 m^3/s	H_{efft} min = 11.25 m	P_{efft} min = 8.8 kW

Check whether these conditions are acceptable as regards to the available flow and the planned application.

12. **Runaway speed and flow**

From Diagram D6 and D7: (with the specific speed of the selected PAT n_{qp} = 95.9)

ε = 1.42 K = 1.00

Calculate the steady state runaway flow for various turbine heads H_t using the rated pump-mode performance: H_{np} 6.65 m; Q_{np} 0.075 m^3/s;

point	H_{tR} =	$Q_{tR} = K\, Q_{np} \sqrt{\frac{H_{tR}}{H_{np}}}$
a)	1.2 H_{np} = 7.98 m	Q_{tR} = 1.00 x 0.075 x $\sqrt{1.2}$ = 0.082 m^3/s
b)	1.6 H_{np} = 10.64 m	Q_{tR} = 1.00 x 0.075 x $\sqrt{1.6}$ = 0.095 m^3/s
c)	2.0 H_{np} = 13.30 m	Q_{tR} = 1.00 x 0.075 x $\sqrt{2.0}$ = 0.106 m^3/s
d)	2.5 H_{np} = 16.63 m	Q_{tR} = 1.00 x 0.075 x $\sqrt{2.5}$ = 0.119 m^3/s

Plot the **no-load line** with these head and flow values (H_{tR} / Q_{tR}).
The intersection point with the system resistance curve (minimum friction) gives the
steady-state runaway conditions:

H_{tR} = 12.80 m ; Q_{tR} = 0.103 m^3/s

Calculate the corresponding **steady-state runaway speed**:

$$n_{tR} = \varepsilon\, n_{np} \sqrt{\frac{H_{tR}}{H_{np}}} = 1.42 * 1450 \sqrt{\frac{12.80}{6.65}} = 2857 \text{ rpm}$$

Read the maximum change of flow from your diagram:

$\Delta Q_{max} = Q_{tR} - Q_{eff\ t\ max}$ = ...0.103....... - ....0.119......... = ...-0.016....... m^3/s
(= ..13.... % of $Q_{eff_t\ max}$)

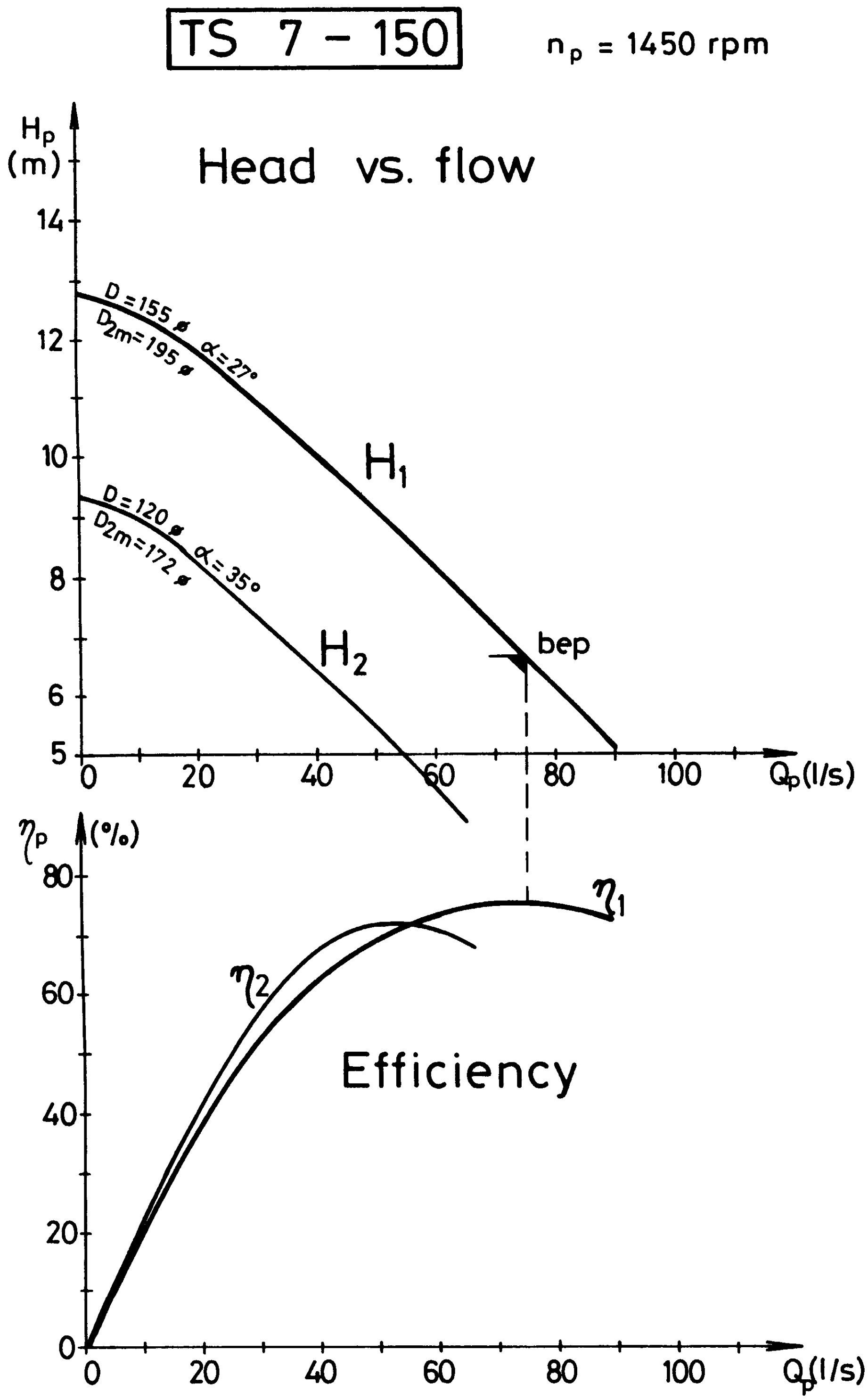


FIGURE 7.1 :
Performance curves of the selected pump

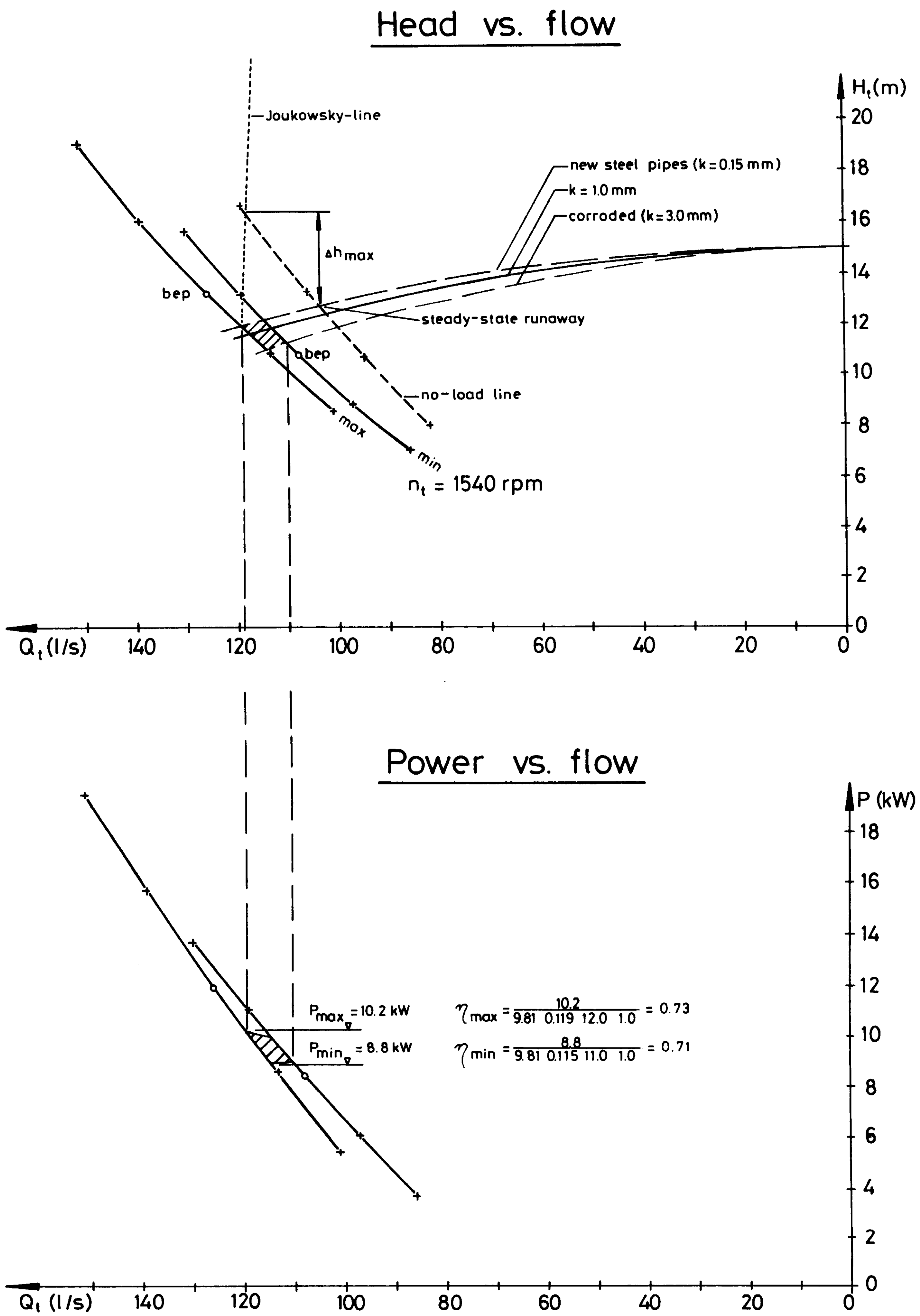


FIGURE 7.2 :
PAT performance curves plotted from calculation as shown above

13. Further checks and modifications on the PAT

a) Runaway speed under transient conditions (see also Section 4.4)

Penstock parameters (from Sections 6.2 and 7.2, Appendix A)

$a = 1214$ m/s $\qquad T_r = 2L / a = 0.044$ s

Joukowsky-line for a change of flow $\Delta Q = 0.010$ m³/s (10 l/s) :

$$\Delta h = a \frac{\Delta Q}{g\ A} = 1214 \frac{0.010}{9.81\ 0.225^2\ \pi/4} = 31.12 \text{ m}$$

Plot the Joukowsky-line in the H-Q diagram (p. 138) starting from the point Q_{eff_t} max / H_{eff_t}
From the intersection point with the no-load line:

$H_{max} = 16.4$ m ; $\Delta h_{max} = H_{max} - H_R = 16.4 - 12.8 = 3.6$ m

Inertia of the direct-coupled induction generator + PAT (see also Appendix G) $\quad J \approx 0.05$ kgm²

Torque at P_{eff_t} max:

$$T_o = \frac{P_{eff_t} \max}{2\pi\ n_o} = \frac{10\ 200}{2\ \pi\ 1540 / 60} = 63.2 \text{ Nm}$$

Unit acceleration time

$$T_a = \frac{J\ \omega_o}{T_o} = \frac{0.05\ 2\pi\ 1540/60}{63.2} = 0.127 \text{ s}$$

$$T_a \text{ eff} = \frac{n_R - n_o}{n_o} T_a = \frac{2857 - 1540}{1540} 0.127 = 0.11 \text{ s}$$

$T_a \text{ eff} > T_r \qquad \Delta h = \Delta h_{max}\ T_r / T_a \text{eff} = 3.6 \text{ m}\ 0.044 / 0.11 = 1.44$ m

$H_{max}' = H_R + \Delta h = 12.8 + 1.44 = 14.24$ m

corresponding maximum speed

$$n_{R\ max}' = n_{tR} \sqrt{\frac{H_{max}'}{H_R}} = 2857 \sqrt{\frac{14.24}{12.80}} = 3013 \text{ rpm}$$

The manufacturer must guarantee that the selected pump can stand 3013 rpm for a short duration and 2857 rpm for a longer period (several hours)

(If the same pump is available for 2900 rpm rated speed applications, runaway speed as calculated above is unlikely to be a problem.)

b) Cavitation / Setting of the PAT above tailrace water level (see also Section 3.6.2)

(layout and notations see Figure A11, Appendix A)

$$NPSH_{available} = \frac{p_{atm}}{\rho\, g} - (H_{stat\,suction} + D/2) + H_{loss} - v_s^2 / 2g - \frac{p_v}{\rho\, g}$$

with: $p_{atm} = 0.97$ bar $= 97000$ N/m^2 (from Table p. A12, for an altitude of 360 m a.s. l.)

$H_{stat\ suction} + D/2 = 2.0 + 0.195/2 = 2.10$ m

$H_{loss} = 0.91$ m

velocity head at suction branch: $\frac{v_s^2}{2g} = \frac{Q^2}{A^2} = \frac{0.119^2}{(0.25^2\pi/4)^2\, 2g} = 0.30$ m/s

$p_{vapour\ 20\,°C} = 2338$ N/m^2 ; density $\rho_{20\,°C} = 998.2$ kg/m^3 (from p. A12, Appendix A)

$$NPSH_{available} = \frac{97000}{998.2\ \ 9.81} - 2.10 + 0.91 - 0.30 - \frac{2338}{998.2\ \ 9.81} = 8.18 \text{ m}$$

from Figure 3.32: with $n_{qp} = 95.9$ $\sigma = 0.55$; $H_{n_t} = 13.2$ m

TREH $= \sigma * H_{n_t} = 0.55 * 13.2 = 7.26$ m

$NPSH_{available} = 8.18$ m > TREH $= 7.26$ m ; safety margin $= 0.92$ m (okay)

APPENDIX A:

ESSENTIAL THEORY OF ENGINEERING HYDRAULICS

1. STEADY AND UNSTEADY STATE FLOW

The flow parameters such as velocity, pressure, and density of a flow for each point are independent of time in a steady flow whereas they depend on time in unsteady flow.

Example for steady flow: flow through a pipe of variable diameter under constant pressure head (e.g. reservoir or tank).

Example for unsteady flow: flow through a pipe of variable diameter under variable pressure due to an increasing/decreasing water level of the reservoir or opening or closure of a valve or stopping/starting hydraulic machines connected to the pipe.

2. EQUATION OF CONTINUITY

For practical purposes one can say that the masses in a defined system do not change (no transformation of mass into energy according to Einstein). Considering an element of a pipeline, one can therefore state that the mass/second entering the tube must be equal to the mass/second leaving it (principle of mass conservation) since there are no losses across the tube wall.

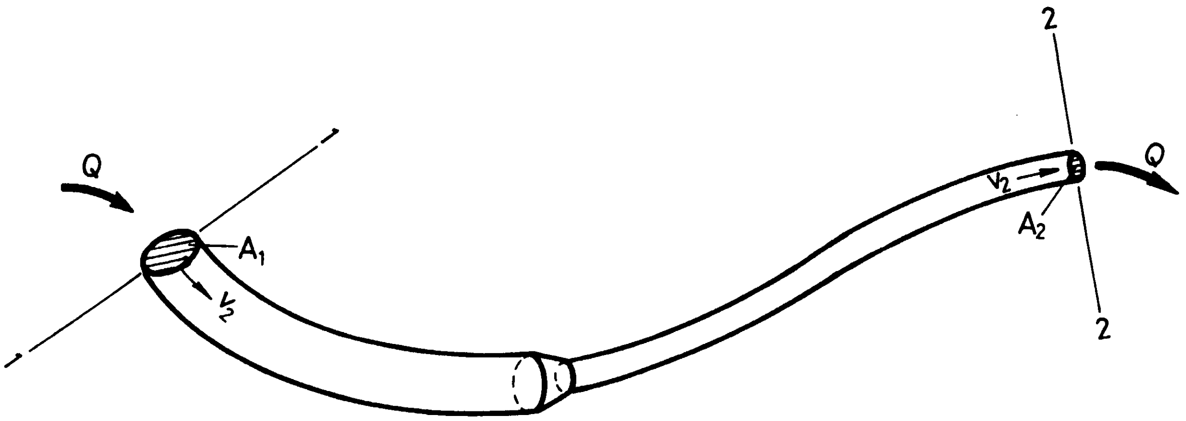


FIGURE A .1 :
Equation of continuity

$$\boxed{\rho_1 v_1 A_1 = \rho_2 v_2 A_2} \qquad (A.1)$$

where v are the steady average velocities at the entrance and exit, A the cross-sectional areas at entrance and exit (perpendicular to the centre-line of the tube) and ρ the corresponding densities of the fluid.

For <u>incompressible steady flow</u>, the above equation reduces to the one dimensional continuity equation:

$$\boxed{A_1 v_1 = A_2 v_2 = Q = \text{constant}} \qquad (A.2)$$

where Q is the volumetric rate of flow called discharge expressed in m^3/s.

3. CONSERVATION OF ENERGY: BERNOULLI'S EQUATION:

Energy can neither be produced nor destroyed but only transformed. The potential energy of water stored in a reservoir on top of a hill is transformed into kinetic energy (and heat due to friction losses) on its way down the hill (open channel flow). At the foot of the hill, kinetic energy is maximum while the potential energy is zero; the total energy, if friction losses are not considered, is the same at the top and the bottom of the hill and at all points in between.

$$\boxed{\text{- potential energy + kinetic energy = constant}} \qquad \text{(A .3)}$$

(only for frictionless, open channel flow)

Considering flow in closed conduits, a third form of energy in fluid flow must be defined, namely the energy derived from the action of a force or pressure (e.g. the work done on the water by a piston displacing a quantity of fluid a certain distance).
Applying the principle of conservation of energy to these three forms of energy leads to the Bernoulli Equation, which applies only to steady flow.

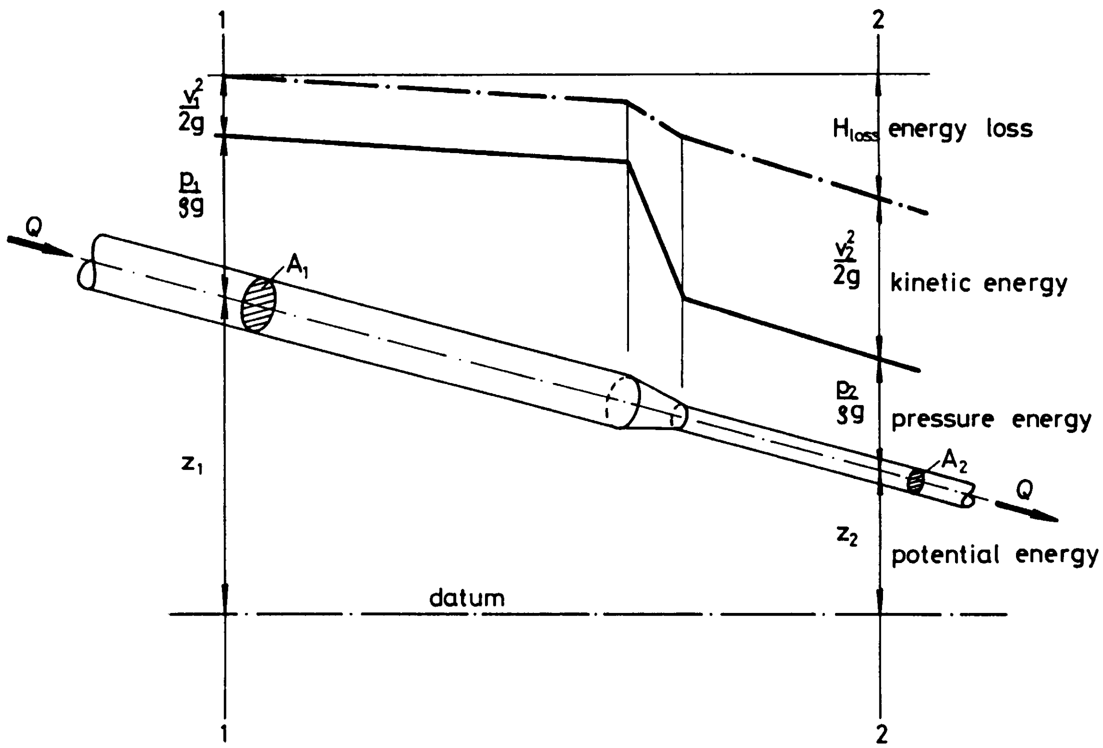


FIGURE A .2 :
Bernoulli's Equation applied to a pipe section

$$\boxed{\frac{p_1}{\rho g} + z_1 + \frac{v_1^2}{2g} = \frac{p_2}{\rho g} + z_2 + \frac{v_2^2}{2g} + H_L}$$

(A .4)

where

- $p/(\rho * g)$ **pressure head**, with p = pressure in N/m2 and ρ = density of the fluid in kg/m^3,

- z = elevation or **potential head** in m

- $v^2/(2g)$ = **velocity or kinetic head**, with v = velocity in m/s and g = gravitational acceleration (9.81 m/s^2)

- H_L = energy lost due to friction and the formation of eddies expressed in m fluid column

Note that in this form each term of the equation has the dimension of length, hence the name "head".

4. ENERGY AND PRESSURE LINE

The three forms of energy in the Bernoulli Equation, pressure, potential and kinetic head, can be shown graphically. This is a very convenient method in pipeline (hydropower and water supply) design because it shows the available pressure at each point of the network with more clarity than tables. Note that the datum (reference level) can be chosen arbitrary since energy is not an absolute quantity and can therefore be measured from any convenient datum.

The distance between this datum and the centre-line of the pipe represents the potential energy at each point (see Figure 3 below). The energy line for the water in the reservoir is the free surface (velocity is near zero, pressure is atmospheric which is usually taken as the reference pressure). In an ideal fluid flow (no losses) the energy line would be a horizontal line. However, due to friction and local losses, it falls steadily (friction) or abruptly (local losses) from the reservoir to the pipe outlet.

The pressure line is drawn everywhere at a distance of the velocity head $v^2/(2g)$ below the energy line. The distance between the centre-line and the pressure line is a measure to scale for the pressure energy. If stand-pipes were mounted onto the pipeline at various points the water level in each would rise to the pressure line.

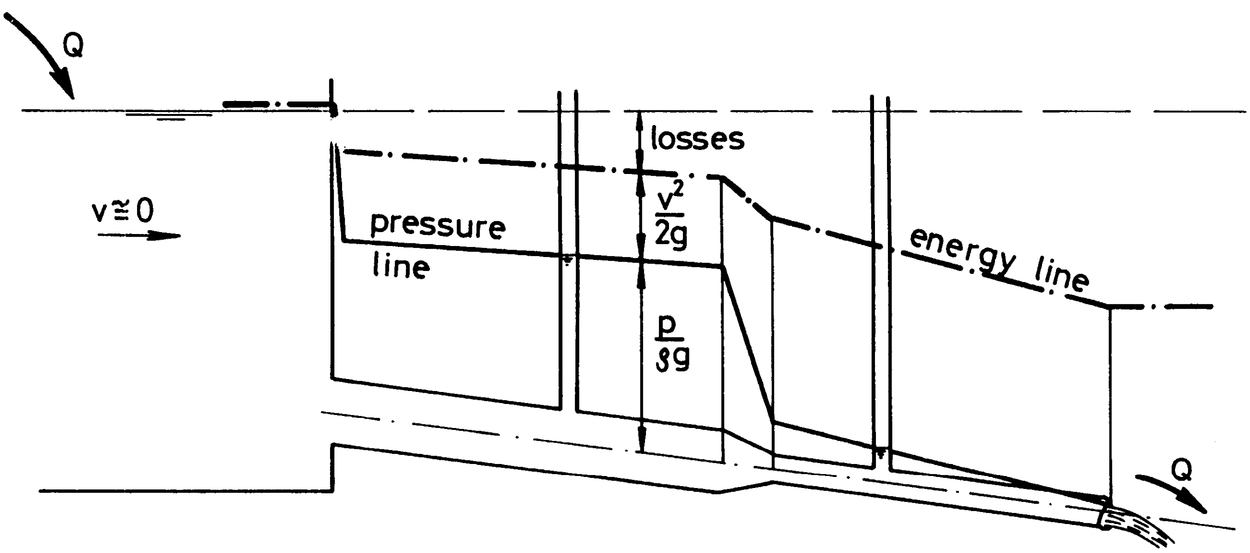


FIGURE A .3 :
Energy and pressure lines for a pipeline discharging from a reservoir

5. LOSSES IN PIPELINE SYSTEMS

5.1 General

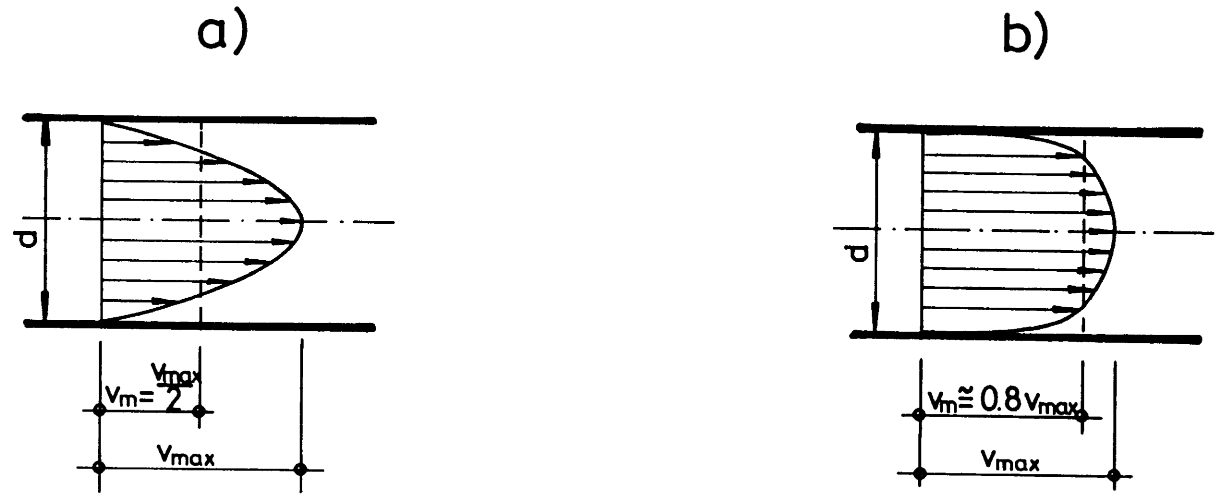
In real fluid flows, losses occur due to the resistance of the pipe walls and the fittings to this flow and lead to an irreversible transformation of the energy of the flowing fluid into heat. Two forms of losses can be distinguished:

- losses due to friction and

- local losses

5.2 Losses due to Friction

Losses due to friction originate in the shear stresses between adjacent layers of water gliding along each other at different speed. The very thin layer of water adhering the pipe wall does obviously not move while the velocity of every concentric layer increases to reach maximum velocity at the centre-line of the pipe. If the fluid particles move along smooth layers, the flow is called laminar or viscous and shear stresses between the layers dominate. In engineering practice however, the flow in a pipeline is usually turbulent, i.e. the particles move in irregular paths and changing velocities.



v_m = mean velocity

FIGURE A 4:
Velocity distribution in pipe flow for a) laminar flow and b) turbulent flow

To characterize the type of flow, the dimensionless Reynolds number Re is used:

$$\boxed{Re = \frac{v\ d}{\nu}} \qquad (A.5)$$

where v = mean velocity of flow (m/s)
d = inside diameter of the pipe (m)
ν = kinematic viscosity in m^2/s
for water at 10^oC : $1.31*10^{-6}\ m^2/s$
for water at 20^oC : $1.0*10^{-6}\ m^2/s$

If Re < 2000, flow will be laminar and above Re = 2500 to 4000, it will be turbulent; the range in between is a critical undefined zone where both forms of flow exist for the same Reynolds number.

For the calculation of friction losses for **turbulent flow**, the following formula (Darcy-Weisbach) is applied:

$$\boxed{H_{friction} = \frac{\lambda \; L}{d} \; \frac{v^2}{2\,g}} \qquad (A.6)$$

in m fluid column, where

- λ = friction factor according to the Moody diagram (see below)
- L = length of pipe section with constant diameter in m
- d = diameter of the pipe in m
- v = average velocity in m/s

Experimental work has been carried out to determine λ, the friction factor, for commercial pipes and has lead to an empirical formula attributed to Colebrook and White. For practical purposes the so-called Moody diagram is used (see below) which is a plot of λ values computed with the Colebrook formula for laminar and turbulent flow in smooth and rough straight pipes.
The friction factor depends, apart from the Reynolds number, on the absolute roughness of the pipe; some values for different pipe materials and conditions are given in Table 6.
As friction losses increase approximately with the square of the velocity of flow (v), pipes should be chosen with as large a diameter as possible. Since large pipes are expensive, a compromise between the hydraulic losses and the economic feasibility of the installation must be found. Diagram 7 indicates economic pipe diameters as a function of flow (Q).

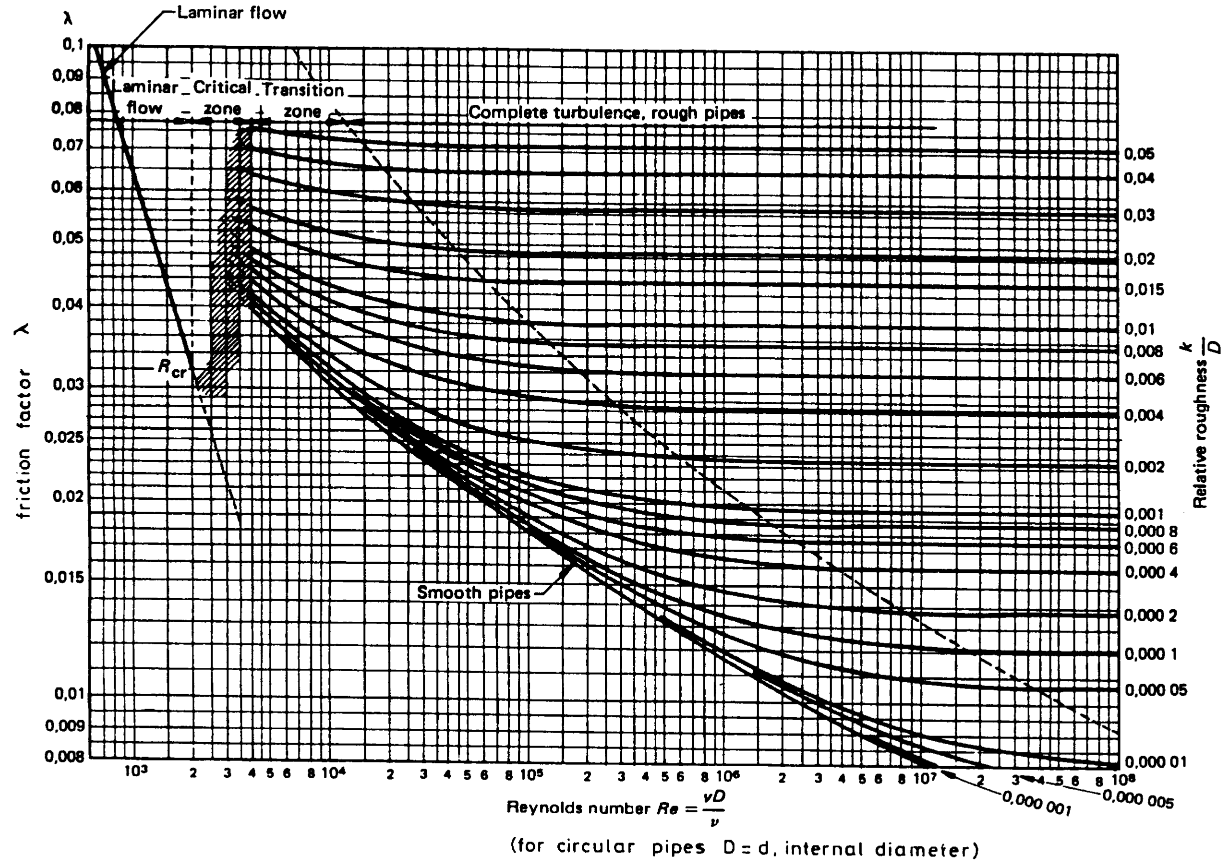


FIGURE A 5 :
Moody diagram

TABLE A6 :
Absolute roughness k of commercial pipes

Pipe material	Condition	k in mm
1. PVC, PE 2. Brass 3. Copper 4. Lead 5. Glass (pure)	technically smooth	0.0015 to 0.01
6. Galvanized steel	new	0.15
7. Cast iron	new with deposits with bituminous coating heavily corroded	0.25 to 1.0 1.0 to 4.0 0.12 to 0.3 up to 3.0
8. Riveted steel		1.0 to 10
9. Welded steel	in good condition with bituminous coating with deposits	0.04 to .010 0.05 to 0.1 1.5 to 4.0
10. Concrete		1.0 to 10

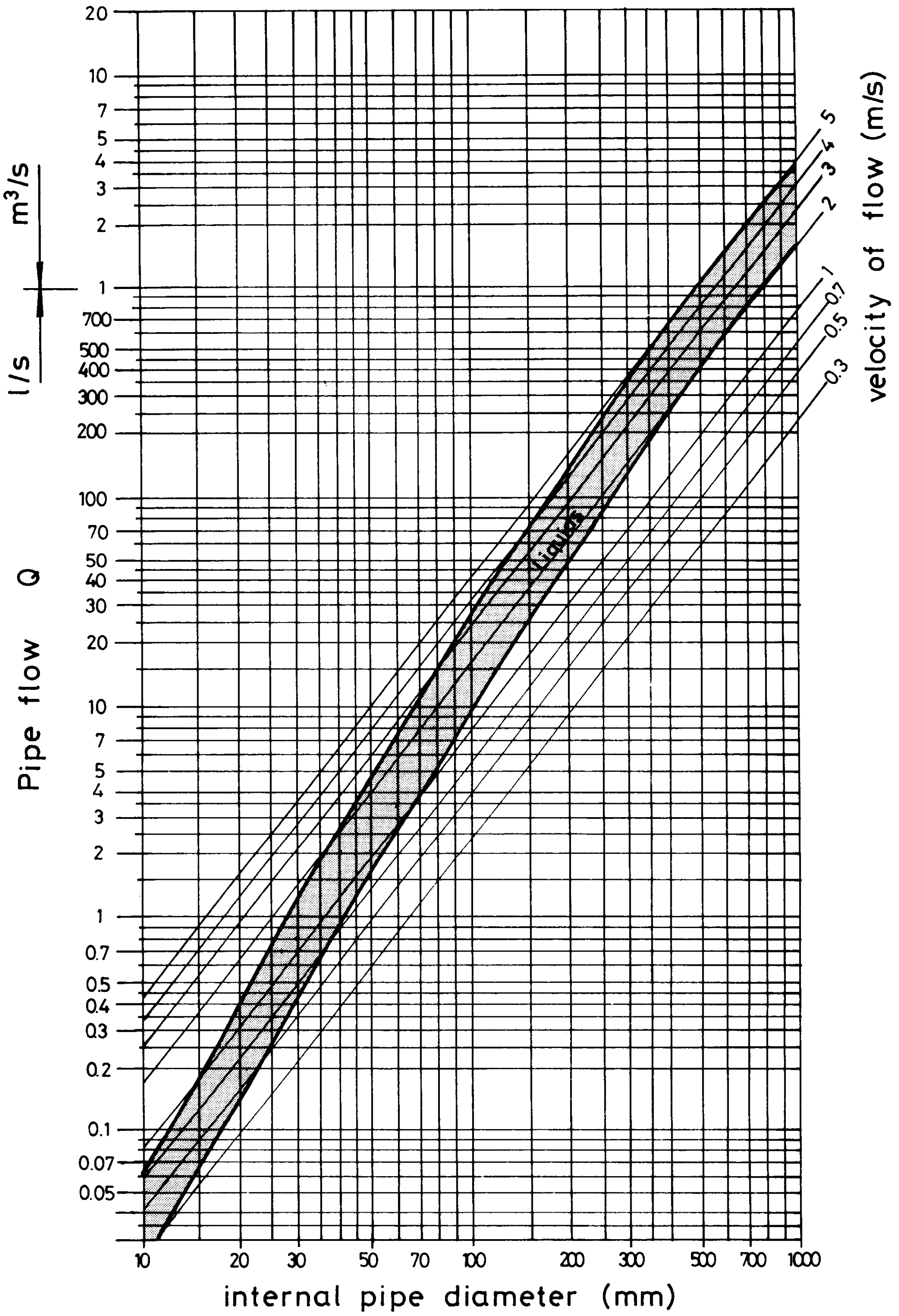


FIGURE A7 :
Economic pipe diameters as a function of flow (Q)

5.3 Local Losses

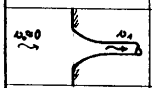
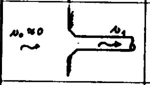
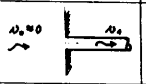
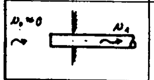
Local losses occur at changes of cross sections, at valves and at bends. These losses are sometimes referred to as minor losses since in long pipelines their effect may be small in relation to the friction loss. Local losses are expressed as multiples of the kinetic head (as are friction losses, see above):

$$\boxed{H_1 = \zeta_1 \frac{v^2}{2g}} \qquad (A.7)$$

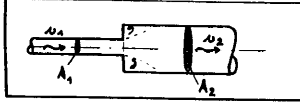
in m , where ζ is the loss coefficient;

for some frequently used cases, ζ values are given below.

Inlet loss $\Delta h = \zeta v_1^2/(2g)$

	remarks	ζ value
 $v_0 \approx 0$, v_1	conical	$\zeta = 0.05 - 0.15$
 $v_0 \approx 0$, v_1	chamfered edges	$\zeta = 0.25$
 $v_0 \approx 0$, v_1	sharp edges	$\zeta = 0.50$
 $v_0 \approx 0$, v_1	protruding	$\zeta = 3.0$

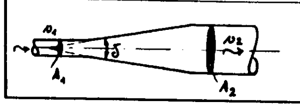
Sudden expansion $\Delta h = \zeta v_2^2/(2g)$



According to Borda – Carnot

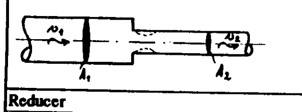
$\zeta = (A_2/A_1 - 1)^2$

Diffusor $\Delta h = \zeta v_2^2/(2g)$



$\zeta = \eta\,(A_2/A_1 - 1)^2$
with $\eta = 0$ for $\delta < 8°$
with $\eta = 1$ for $\delta \geq 30°$

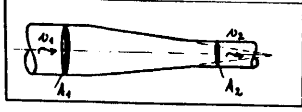
Sudden contraction $\Delta h = \zeta v_2^2/(2g)$



According to Franke

$\zeta = 0.45\,(1 - A_2/A_1)$

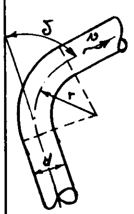
Reducer $\Delta h = \zeta v_2^2/(2g)$



$\delta \leq 8° \Longrightarrow \zeta = 0$

$\delta \leq 20° \Longrightarrow \zeta = 0.04$

Bends $\Delta h = \zeta v^2/(2g)$

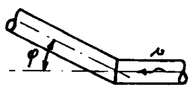


for smooth pipes ($k \leq 0.01$ mm)

r /d	1	2	4	6	10
$\delta = 15°$	0.03	0.03	0.03	0.03	0.03
$\delta = 22.5°$	0.045	0.045	0.045	0.045	0.045
$\delta = 45°$	0.14	0.09	0.08	0.075	0.07
$\delta = 60°$	0.19	0.12	0.10	0.09	0.07
$\delta = 90°$	0.21	0.14	0.11	0.09	0.08

for rough pipes ($k > 0.01$ mm):
double the values of the table

Elbows $\Delta h = \zeta v^2/(2g)$



angle φ	5°	10°	15°	22.5°
smooth	0.014	0.029	0.044	0.075
rough	0.021	0.045	0.064	0.105

angle φ	30°	45°	60°	90°
smooth	0.12	0.245	0.47	1.15
rough	0.165	0.325	0.6	1.3

Footvalve *self-acting closure at v = 0* $\Delta h = \zeta v^2/(2g)$

		NB (mm)	
		50 – 80	100 – 350
velocity in m/s	1.0	4.1	3.0
	2.0	3.0	2.25
	3.0	2.8	2.25

Non-return valve *self-acting closure at v = 0* $\Delta h = \zeta v^2/(2g)$

		NB (mm)		
		50	200	500
velocity in m/s	1.0	3.05	2.95	2.85
	2.0	1.35	1.3	1.15
	3.0	0.86	0.76	0.66

Regulated valves *fully opened* $\Delta h = \zeta v^2/(2g)$

	NB (mm)			
	100	200	300	500
Gate valve	0.25	0.25	0.22	0.15
Butterfly valve			0.25	0.25
Ball/Rotary valve	0.22	0.17	0.08	0.03

TABLE A 8 :
Loss coefficients ζ for local losses

6. PIPELINE SYSTEM ANALYSIS IN PUMPING PLANTS AND MHP

6.1Pump Station

6.1.1 Total Pump Head

Figure 9 shows a pump delivering water from a river to a higher irrigation channel, through a static head of H_{st} = 15 m at a discharge Q = 100 l/s. The pump must generate a total head equal to H_{st} **plus** the pipeline head losses. By plotting the energy and pressure lines on the longitudinal section of the system, the required pump head as well as the pressure head at each point of the system at the operating conditions can be read immediately.

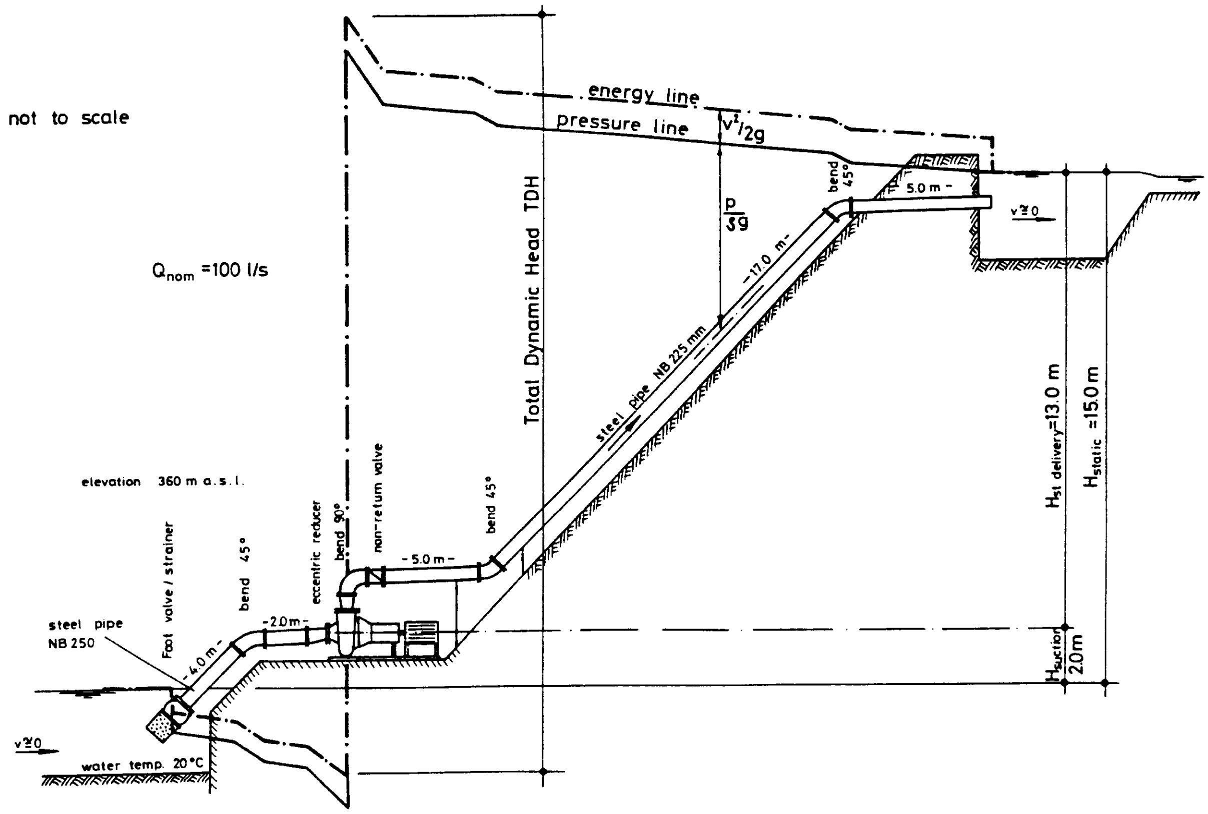


FIGURE A 9 :
Energy and pressure lines in a pipeline system with a pump

To select a suitable pump for this pump lift irrigation scheme, the total dynamic head must be determined; it consists of the static suction lift and the losses in the suction pipe and the static delivery head and the losses in the delivery pipe.
Total dynamic head:

$$TDH = H_{stat\ suction} + H_{l\ suction} + H_{stat\ deliv} + H_{l\ deliv} + \frac{v_{deliv}^2}{2g} \qquad (A.8)$$

a) <u>**losses suction side**</u> (loss coefficients according to sections 5.2 and 5.3 above)
A suction pipe of 250 mm nominal bore (NB) (pipe material: steel) is assumed; total pipe length incl. bends = 6.0 m
Friction losses, using the Moody diagram Figure 5 above:

$v = Q/A$ (continuity) $= 0.1/(0.25^2 * \pi /4) = 2.04$ m/s; $Re = v*d/\nu = 509'000$
absolute roughness $k = 1.0$ mm, relative roughness $= k/d = 1/250 = 0.004$
from Moody diagram: $\lambda = 0.0248$;

$H_f = \lambda * L/d * v^2/(2g) = 0.0248*6/0.25*2.04^2/(2g) =$	0.13 m

Local losses:

- strainer with foot valve: $\zeta = 3.0$	
$H = \zeta * v^2/(2g) =$	0.63 m
- bend 45°: $\zeta = 0.2$	
$H = \zeta * v^2/(2g) =$	0.04 m
- reducer NB 250 - NB 200 (suction branch pump)	
$v_s = Q/A_s = 3.18$ m/s, $\zeta = 0.04$ $H = \zeta * v_s^2/(2g) =$	0.02 m
Total suction head loss	0.82 m
Static suction lift	2.00 m
Total dynamic suction lift $H_{s\ lift}$ =	**2.82 m**

<u>**b) losses delivery side**</u>
A suction pipe of 225 mm nominal bore (NB) (pipe material: steel) is assumed; total pipe length incl. bends = 27.0 m
Friction losses $v = Q/A$ (continuity) $= 0.1/(0.225^2 * \pi /4) = 2.52$m/s;
$Re = v*d/\nu = 566'000$; absolute roughness $k = 1.0$ mm,
relative roughness $= k/d = 1/225 = 0.004$ from Moody diagram: $\lambda = 0.0248$; $H_{f\ deliv} =$

$\lambda * L/d * v^2/(2g) = 0.0248*27.0/0.225*2.52^2/(2g) =$	0.96 m

Local losses:

- diffusor (pump delivery branch NB 150 mm) $\zeta = (A_2/A_1 - 1)^2 = 1.56$	
$H = \zeta * v^2/(2g) =$	0.50 m
- bend 90°: $\zeta = 0.3$	
$H = \zeta * v^2/(2g) =$	0.10 m
- non-return valve: $\zeta = 1.3$	
$H = \zeta * v^2/(2g) =$	0.42 m
- bends 2 x 45°: $\zeta = 0.2$	
$H = 2*\zeta * v^2/(2g) =$	0.13 m
- outlet loss (= kinetic energy)	
$H = v^2/(2g) =$	0.32 m
Total delivery head loss	2.43 m
Static delivery head	13.00m
Total dynamic delivery head	**15.43 m**

c) total dynamic head

Total dynamic head (TDH) = suction + delivery head = 2.82 m +15.43 m = 18.25 m
Usually, a safety margin is added to allow for uncertainties of actual head loss and roughness coefficients, say 0.75m,

thus TDH = 19.0 m at Q = 100 l/s

(the selection of a suitable pump for these conditions will be shown in Appendix B)

6.1.2 Net Positive Suction Head (NPSH) - Cavitation

From Figure 9 above it can be seen that the pressure at the suction branch of the pump is well below atmospheric pressure. When water is forced into the delivery pipe, a vacuum is created in the pump and, due to atmospheric pressure, more water is pushed through the suction pipe into the pump. (Note that the term suction lift is actually misleading because a pump cannot suck in water, it must be pushed in by atmospheric pressure.)
Theoretically, a pump could therefore be located close to 10 m above water surface (normal atmospheric pressure at sea level = 9.81 m water column). In practice, this maximum suction lift cannot be attained since

a) the pressure at the suction branch of the pump is atmospheric pressure **reduced** by the losses in the suction pipe including the velocity head (at pump suction diameter which might be different from the bore of the suction pipe);

b) at low pressure, water starts to boil even at ambient temperature. If the absolute pressure in the pump falls below **vapour pressure** (pressure at which water vaporizes as a function of temperature), then vapour pockets occur which collapse after entering a region of higher pressure. This process, called **cavitation** can destroy a pump or cause it to deteriorate rapidly due to the violent hammering action on impeller and pump surfaces which the imploding bubbles come in contact with. A pump under cavitation may operate very inefficiently and noisily; beginning cavitation can be observed by a crackling noise as if gravel were passing through the pump. Fully developed cavitation sounds like gunfire and the pump may vibrate heavily.

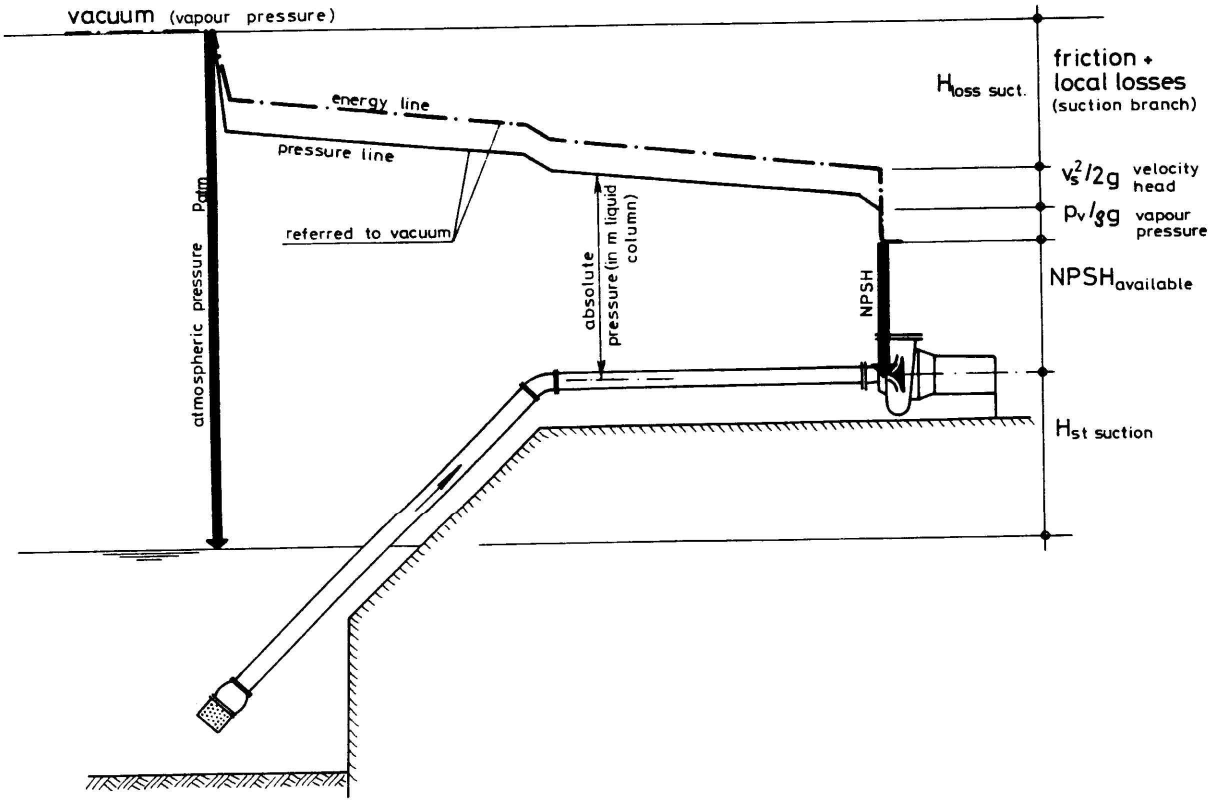


FIGURE A 10 :
Net positive suction head (NPSH) of a pump

NPSH available represents the remaining pressure head available to force the liquid into the pump impeller. According to Figure 10 above we can write (expressed in m liquid column):

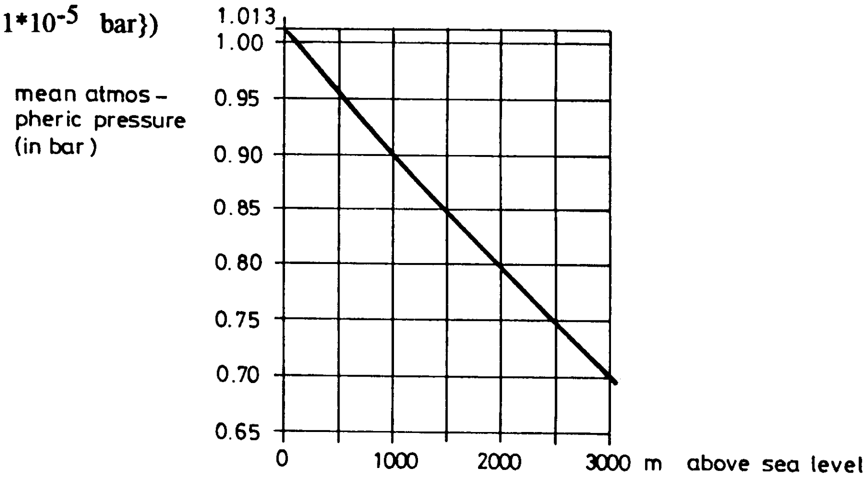
$$\boxed{NPSH_{available} = \frac{p_{atm}}{\rho\, g} - H_{stat\ suction} - H_{loss\ suction} - \frac{v_s^2}{2g} - \frac{p_v}{\rho\, g}} \quad \text{(A.9)}$$

(NPSH in m liquid column)

(note: $H_{stat\ suction}$ must be added if the pump is located below the water level = suction <u>head</u>)

where:

- p_{atm} = atmospheric pressure as a function of the elevation above sea level of the pump installation (in N/m2 {= $1*10^{-5}$ bar})



- ρ = density of water as a function of temperature (in kg/m3)
- p_V = vapour pressure of water as a function of temperature (in N/m2 {= $1*10^{-5}$ bar})

Temperature (°C)	Density (kg / m³)	Vapour pressure (N / m²)
0	999.9	611
5	1000	872
10	999.7	1228
20	998.2	2338
30	995.7	4243
40	992.2	7376

Any pump installation must have an NPSH available equal or greater than the required NPSH of the pump.

$$\mathbf{NPSH_{available} > NPSH_{required}}$$

NPSH required can be obtained from the pump performance curves (see Appendix B). It is a function of the pump design, its capacity and speed. Pumps with suction heads lower than NPSH required will invariably cavitate though the pressure at the pump suction branch may still be above vapour pressure. When water is picked up by the impeller, it is **<u>further accelerated</u>**; this causes (according to Bernoulli's Equation, see section 3) a drop in pressure and the formation of vapour pockets in the pump.

6.2 Micro-Hydropower Plants

6.2.1 Turbine Net Head

Figure 11 shows a micro-hydropower plant with a gross head of 15.0 m and a nominal discharge of 100 l/s. These are the same conditions as for the pump scheme presented in section 6.1 above. To calculate turbine output, the net head (= gross head minus losses) must be determined.

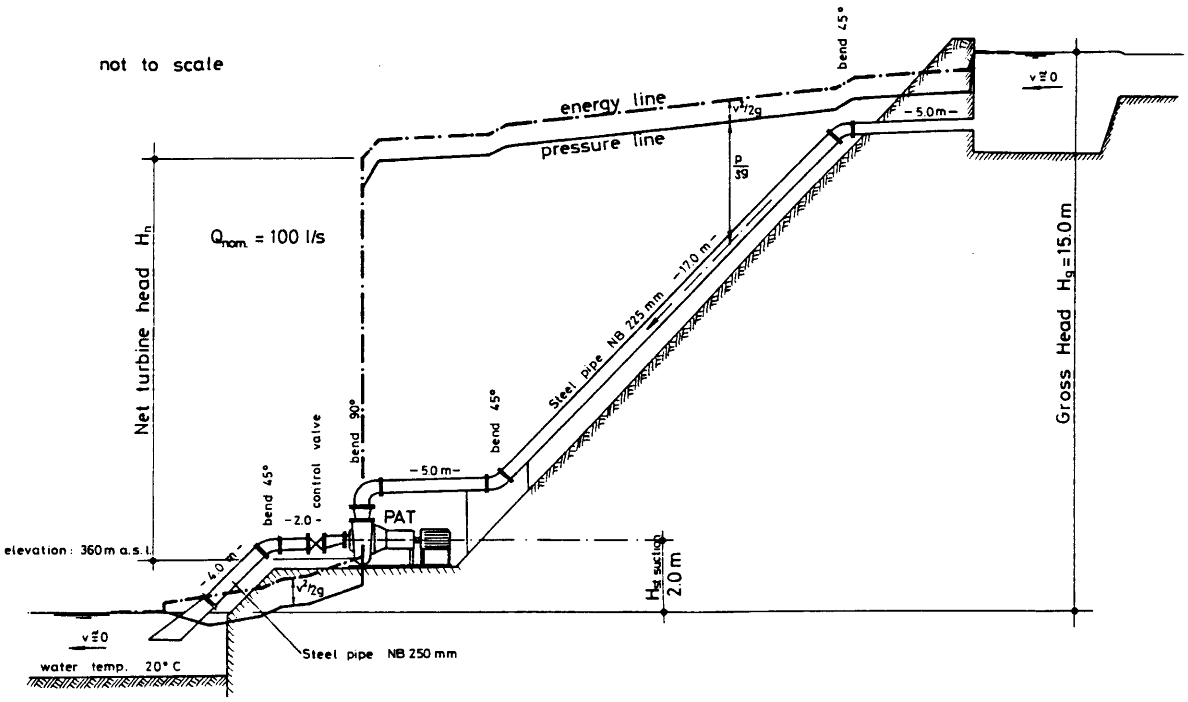


FIGURE A 11 :
Energy and pressure lines in the penstock of an MHP

Turbine Net Head (H_n) = Gross head - Penstock losses - losses draft tube (incl. outlet loss=kinetic energy)

a) penstock losses

The same tube (steel NB 225 mm, length 27 m) as for the pump scheme of section 6.1 is used.

Friction losses (see above) $v = 2.52$ m/s

H = 0.96 m

Local losses

- inlet loss: $\zeta = 0.5$

$H = \zeta * v^2/(2g) =$ 0.16 m

- bends 2 x 45^o: $\zeta = 0.2$

$H = 2 * \zeta * v^2/(2g) =$ 0.13 m

- bend 90^o : $\zeta = 0.3$

$H = \zeta * v^2/(2g) =$ 0.10 m

- reducer NB 225 - NB 150 ($v_2 = Q/A_2 = 5.65$ m/s) $\zeta = 0.04$

$H = \zeta * v_2^2/(2g) =$ 0.07 m

Total penstock losses = 1.42 m

b) losses in draft tube

The draft tube is gradually expanded over the last meter (from NB 250 to NB 500) to reduce outlet losses (steel pipe: total length incl. bends and diffusor = 6 m)

Friction draft tube (calculated with NB 250 mm over entire length of 6 m, see section 6.1.1 above)

$v = 2.04$ m/s H = 0.13 m

Local losses

- expansion turbine outlet to suction pipe NB 150 - NB 250:

$\zeta = (A_2/A_1 - 1)^2 = 3.16$ $v_2 = 2.04$ m/s $H = \zeta * v_2^2/(2g) =$ 0.67 m

- control valve: $\zeta = 0.25$ (fully opened gate valve)

$H = \zeta * v^2/(2g) =$ 0.10 m

- bend 45^o : $\zeta = 0.2$

$H = \zeta * v^2/(2g) =$ 0.04 m

- diffusor at draft tube outlet with angles smaller than 8^o

$\zeta = 0$ H = 0.00 m

- outlet loss: $v = Q/A = 0.1/(0.5^2 * \pi /4) = 0.5$ m/s,

$H = v^2/(2g) =$ 0.01 m

Total losses draft tube = 0.95 m

c) net turbine head

H_n = Gross Head - penstock losses - draft tube losses $\mathbf{H_n = 15.00\ m - 1.42\ m - 0.95\ m = 12.63\ m}$

(the selection of a suitable turbine or pump used as turbine will be presented in the main text, Chapter 7)

6.2.2 Cavitation of the turbine

As for pumps, the NPSH value available for a given installation is calculated and compared with the $NPSH_{required}$ (or sometimes called TREH = Total Required Exhaust Head, supplied by the turbine manufacturer or from graphs, see main text). For turbines TREH represents the minimum backpressure at the turbine outlet for which no danger of cavitation occurs. Note that NPSH for turbines with horizontal shaft is not referred to the centre-line of the shaft of the machine as for pumps but to the highest point of the runner blades.

If the NPSH is related to the net head H_n (and thus becomes independent of the turbine speed), it is called Thoma or Sigma (σ) number:

$$\sigma = NPSH / H_n \qquad \text{or} \qquad TREH / H_n$$

Experiments have shown that Sigma is strongly related to the specific speed of a machine (see Appendix B). Estimates on the cavitation characteristics of any machine (in the absence of detailed data) can therefore be drawn from graphs (see main text).

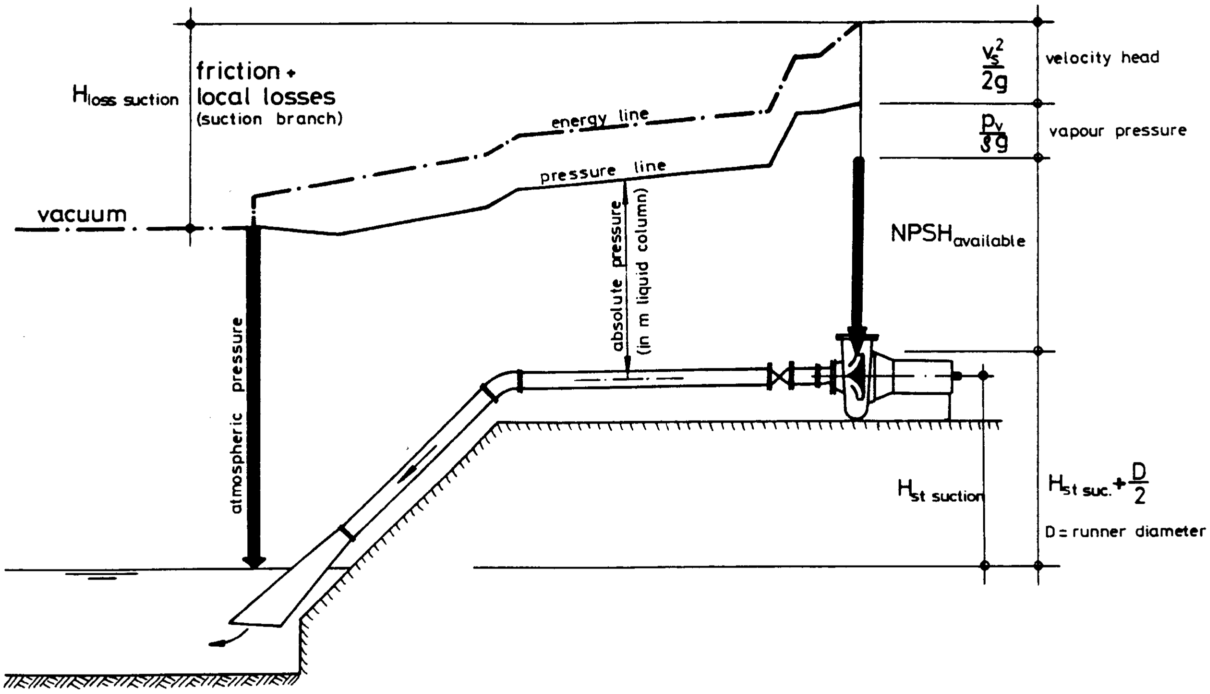


FIGURE A 12 :
NPSH or TREH of turbine

7. PRESSURE TRANSIENTS IN PIPELINES (WATERHAMMER)

7.1 General

Changes in the discharge in pipelines, caused by valve closure/opening or pump and turbine speed regulations, result in pressure surges which are propagated along the pipeline from the source. The simplest case of waterhammer, that due to the instantaneous closure of a valve at the end of a pipeline fed by a reservoir is used to illustrate the principle.

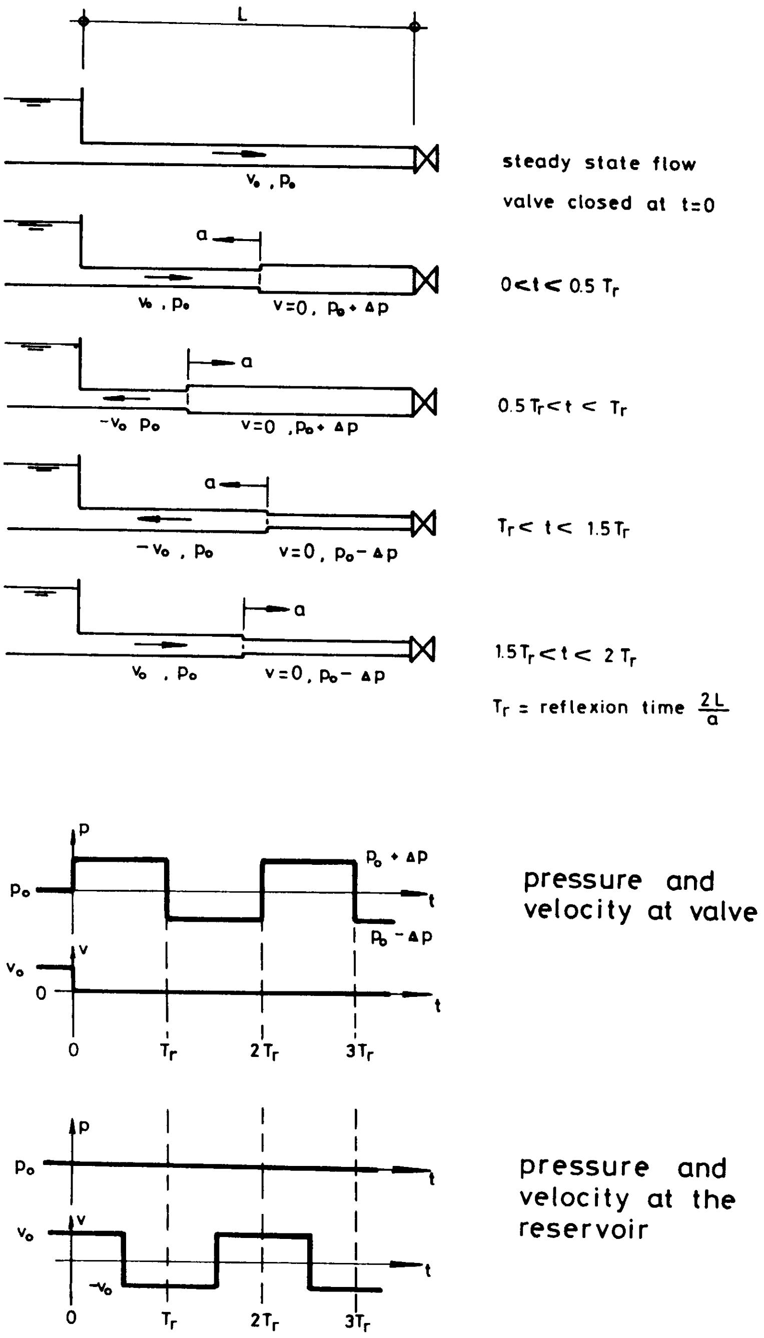


FIGURE A 13 :
Pressure transients due to sudden valve closure

On sudden closure of the valve, the water column close to the valve comes to a rest. Due to this change of momentum, high pressure is produced under which the water is compressed and the pipe slightly expanded; in other words, the kinetic energy of the flowing water is converted into strain energy. This elastic deformation of water and pipe can only be undone in the direction of the reservoir; however, the oncoming flow of water prevents such a relaxation. A shock wave is produced which separates the water flowing in the direction of the valve at speed v_0 (and pressure p_0) and the water at rest but under higher pressure ($p_0 + \Delta p$).

The shock wave travels against the direction of flow at a speed 'a', called its propagation velocity, towards the reservoir. Using the principle of conservation of energy, i.e. loss of kinetic energy = gain of strain energy, it can be shown that the propagation velocity of the shock wave is:

$$\boxed{a = \sqrt{\frac{\frac{E_{liq}}{\rho}}{1 + \frac{d\,E_{liq}}{e\,E_{pipe}}}}} \qquad (A.10)$$

- E_{liq} = modulus of elasticity of the liquid; water: $2*10^9$ N/m2

- ρ = density of the liquid; water: 1000 kg/m3

- E_{pipe} = modulus of elasticity of the pipe material;

  steel : $210*10^9$ N/m2

  PE : $0.1 - 1.1*10^9$ N/m2

  cast iron : $120 - 170*10^9$ N/m2

- d = internal diameter of the pipe
- e = wall thickness of the pipe

After the time t =L/a the shock wave has reached the reservoir where the high pressure $p_0 + \Delta p$ is released into the reservoir (which is under the pressure p_0). Due to this pressure gradient, the water starts to flow back into the reservoir at speed $-v_0$. A new wave front separating the backwards flowing water and the compressed liquid column at rest travels back to the valve where it arrives after the time

$$\boxed{T_r = \frac{2\,L}{a}} \qquad (A.11)$$

(T_r = period or reflexion time).

At this time, the entire water column of the pipe is moving backwards at speed $-v_0$ and creates a vacuum at the valve due to lack of water. Under this low pressure, the pipe is radially contracted and the water comes to rest. A shock wave leaving an area of low pressure behind it travels to the reservoir at speed 'a' where the low pressure is eliminated by water flowing at speed v_0 and pressure p_0 into the pipe; note that these are the initial conditions as before the closure of the valve. These conditions are restored over the entire pipe at time t = 4L/a and the cycle is completed. The whole sequence is repeated again; in practice, the oscillations damp out due to friction and the degradation of the strain energy in the pipe walls into heat energy.

The pressure rise due to the deceleration of the water column can be derived as follows:

According to Newton's second law of motion a force F is necessary to produce an acceleration ($\Delta v/\Delta t$) when acting on a mass m.

$$\boxed{F = m \frac{\Delta v}{\Delta t}} \qquad (A.12)$$

After the sudden closure of the valve in above example, the deceleration of water takes place at speed 'a' which will decelerate the mass m per unit time Δ t. (Note that the shock wave would actually travel at speed (a - v_o), but as v_o is in practice very small as compared to 'a', it can be neglected here.)

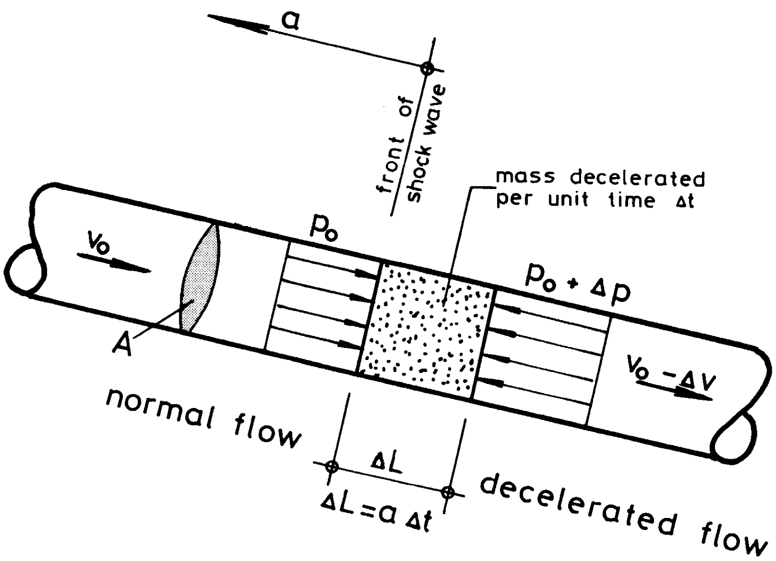


FIGURE A 14 :
Derivation of Joukowsky's Equation

$$m = \Delta L \quad A \quad \rho = a \quad \Delta t \quad A \quad \rho$$

$$F = m \frac{\Delta v}{\Delta t} = a \quad \Delta t \quad A \quad \rho \quad \frac{\Delta v}{\Delta t} \qquad \text{with} \quad F = p \quad A$$

$$\boxed{p = a \ \rho \ \Delta v} \qquad (A.13)$$

or p expressed in liquid column: ($p = \rho \ g \ \Delta h$)

$$\boxed{\Delta h = \pm a \frac{\Delta v}{g} = \pm a \frac{v - v_o}{g}} \qquad (A.14)$$

This is know as <u>Joukowsky's Equation</u>

a = propagation speed of the shock wave (see above)
v_o = initial velocity of the water
v = velocity after acceleration/deceleration
g = acceleration due to gravity (9.81 m/s^2)

7.2 Instantaneous change of flow

An instantaneous change of flow produces a pressure rise or drop (according to the above formula) which is of the same magnitude over the entire length of the penstock.

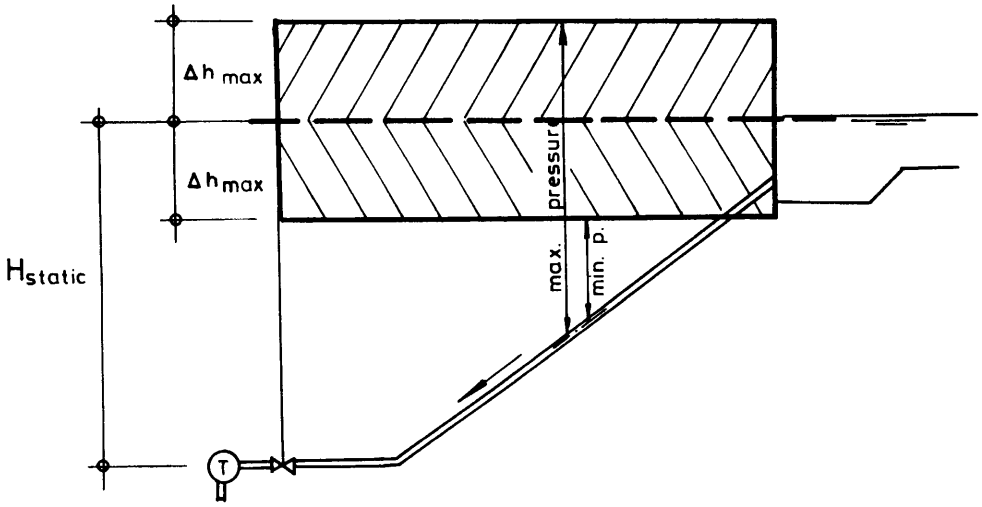


FIGURE A 15
Maximum and minimum pressure due to waterhammer after instantaneous closure of a valve

For the turbine installation shown in section 6.2, the instantaneous closure of the valve (which is very unlikely in practice) would create a maximum change of pressure as follows:

H_{static} on turbine and valve = 13 m, nominal discharge Q_n = 100 l/s
steel pipe NB = 225 mm, wall thickness 6 mm, v_0 = 2.52 m/s (see above)
wave propagation speed a = 1214 m/s and T_r = 2L/a = 2*27/1214 = 0.044s

complete closure within t ---> 0: Δ h max = ± a*Δ v/g =

Δh max = ± 1214*2.52/9.81 = **± 312 m**

decrease of flow of 10 % within t --- > 0: $Q_o = Q_n$ = 100 l/s, Q_t = 90 l/s

thus Δ v = $(Q_o - Q_t)$ / A= 0.25 m/s **Δhmax** = ± 1214*0.25/9.81 = **± 31.2 m**

These pressure head rises are several times the static head of the installation. Such high transient pressures can be avoided by increasing the time of valve closures or turbine shut-downs.

7.3 Gradual change of flow

If the closure/opening (T_f) of a valve takes place **within the first period T_r = 2L/a** of the waterhammer cycle, i.e. before the shock wave arrives at the valve or turbine, the maximum pressure rise/drop **at the valve is still of the same magnitude** as shown under 7.2 above.
However, the pressure rise **along the penstock is reduced** due to the effect of the shock wave reflected at the reservoir and travelling downstream before the pressure build-up is completed.

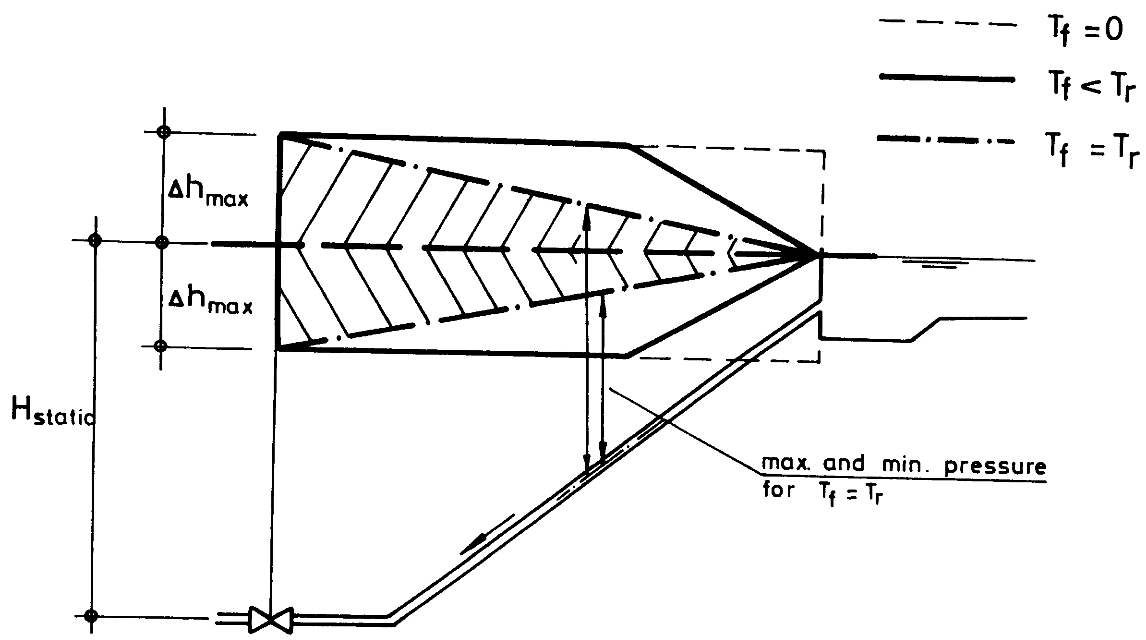


FIGURE A 16 :
Maximum pressure rise along the penstock due to different closure times (T_f) of a valve (T_f < = T_r)

If the valve closure is not completed within the first period T_r = 2L/a, then the pressure gradient in the shock wave is not as steep as with an instant shut-down and the maximum pressure rise is also smaller. In this case, it may be assumed that the maximum pressure rise is reached as soon as the reflected shock wave arrives at the valve (i.e. after the period T_r = 2L/a) and that the pressure cannot rise further under this influence.
In reality, this might not be true (see Figure 18 below). But assuming that the maximum pressure head is equal to the maximum pressure rise under instant shut-down corrected with the ratio T_r/T_f gives values with a good safety margin over the better but more complex methods for waterhammer analysis (e.g Allievi, Schnyder-Bergeron graphical method, characteristics method). The simple formula presented here might be used to check whether waterhammer is likely to be a problem for a given installation or not .

$$\boxed{\Delta h_{max} = \frac{2 \Delta v L}{g T_f}} \qquad (A.15)$$

(source : Vischer/Huber, Wasserbau, Springer-Verlag)
where

- L = penstock length in m
- T_f = closure/opening time of valve or turbine gate

Note that the maximum pressure rise/drop is independent of the propagation speed of the shock wave.

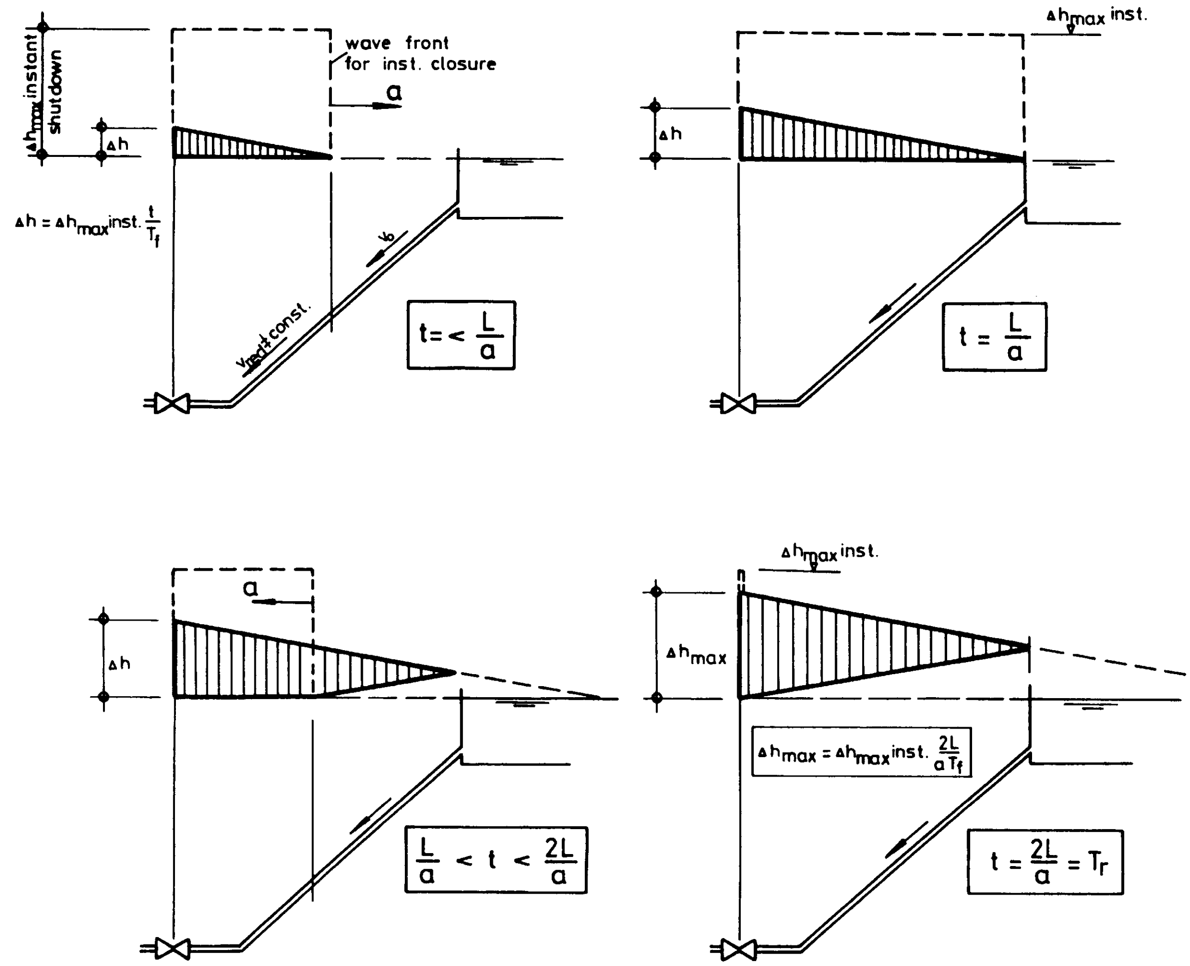


FIGURE A .17 :
Development of pressure transients due to gradual closure of the valve ($T_f > T_r$)

For the turbine installation shown in section 6.2 above, the maximum pressure rise for gradual change of flow is approximately as follows:
From instant shut-down (Section 7.2 above) $T_r = 0.044$s

Assumptions: **complete closure within t = 2 s:** $\Delta' h$ max = ± 2*Δ v*L/(g*T_f) =

<u>Δh max</u> = ± 2*2.52* 27/(9.81*2) = **<u>± 6.9 m</u>**

decrease of flow of 50 % within t = 0.5s: $Q_o = Q_n$ = 100 l/s, Q_t = 50 l/s

thus Δ v = (Q_o - Q_t) / A = 1.26 m/s

<u>Δh max</u> = ± 2*1.26*27/(9.81*0.5)= **<u>± 13.9 m</u>**

It can be seen that for short penstock lines with a reflexion time/period T_r much shorter than gate closure time T_f, waterhammer is not likely to cause serious problems.

For further information on waterhammer problems especially related to PATs under load rejection see main text.

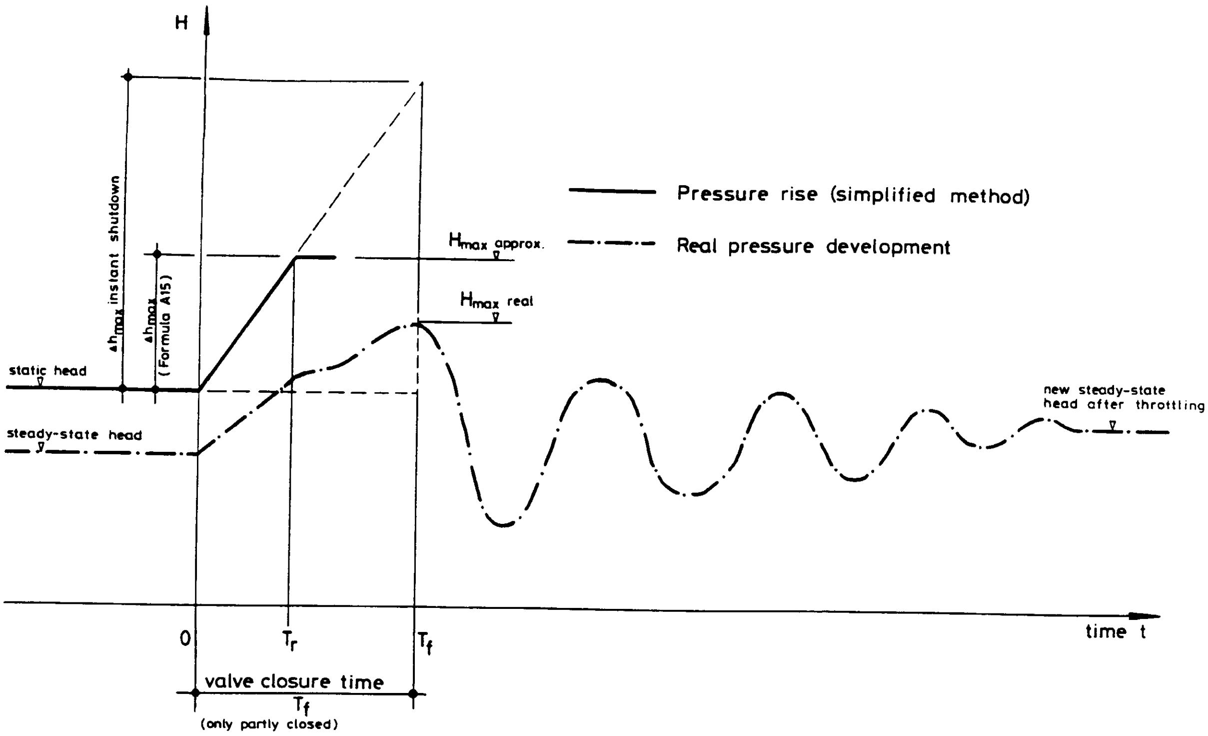


FIGURE A 18 :
Comparison between real waterhammer development and approximation

APPENDIX B:

BASIC THEORY OF HYDRAULIC MACHINES

1. INTRODUCTION

In a hydraulic machine, energy is transformed from one form into another, e.g mechanical energy into an output of fluid energy (in pumps and compressors) or an input of fluid energy into mechanical energy (turbines). Appendix B deals exclusively with rotodynamic machines, where the transformation of energy is effected by a rotating element, the impeller (in pumps) or the runner (in turbines).
In order to be able to estimate the rate of energy transfer to or from the fluid, the flow through an impeller or runner has to be analyzed; the basis of this analysis is given by the velocity vector diagrams which represent the flow as it enters and leaves the rotating element, and the Euler equation which describes the energy transfer from the fluid to the rotating element.

2. TRANSFER OF ENERGY IN HYDRAULIC MACHINES

2.1 Fluid Flow through a Centrifugal Pump

This type of pump is basically comprised of an impeller rotating in a volute or spiral casing (see Figure 1). The fluid, in our case water, enters the centre of the impeller (the eye) and is picked up by the vanes. The rotating motion of the impeller imparts energy to the water as it flows outwards through the impeller. From the outer periphery of the impeller, the water discharges at a high velocity into the volute casing whose cross sectional area increases gradually towards the delivery passage and much of the velocity energy is converted into pressure (see Bernoulli's equation, Appendix A). Thus, the function of the volute is to provide a smooth transition for the flow between the impeller periphery and the discharge pipeline because deceleration of flow is usually accompanied by losses (shock losses) due to the formation of eddies. If this deceleration takes place slowly over a long distance, shock losses are smaller but pump impellers and casings become much bigger. Some pumps incorporate a ring of fixed vanes (known as diffuser ring) around the impeller to guide and decelerate the flow with minimum losses incurred.

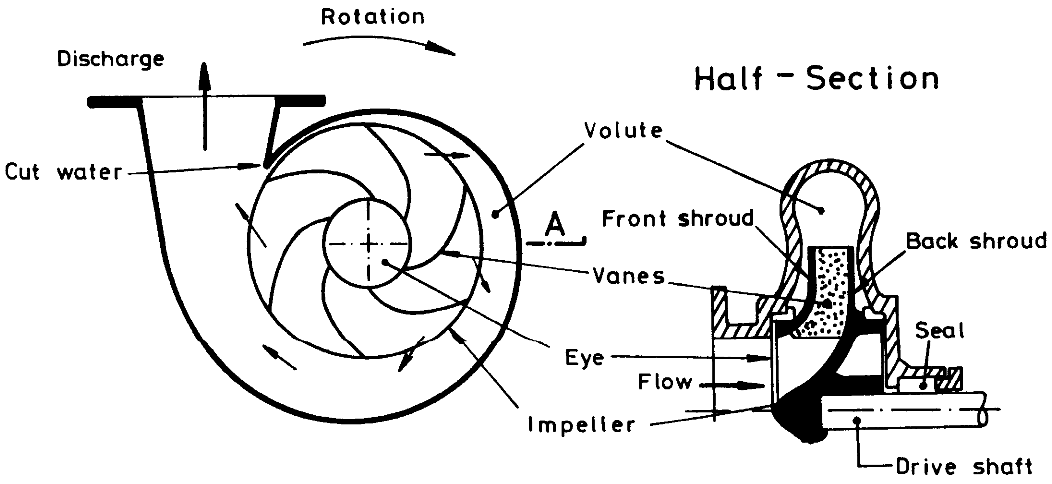
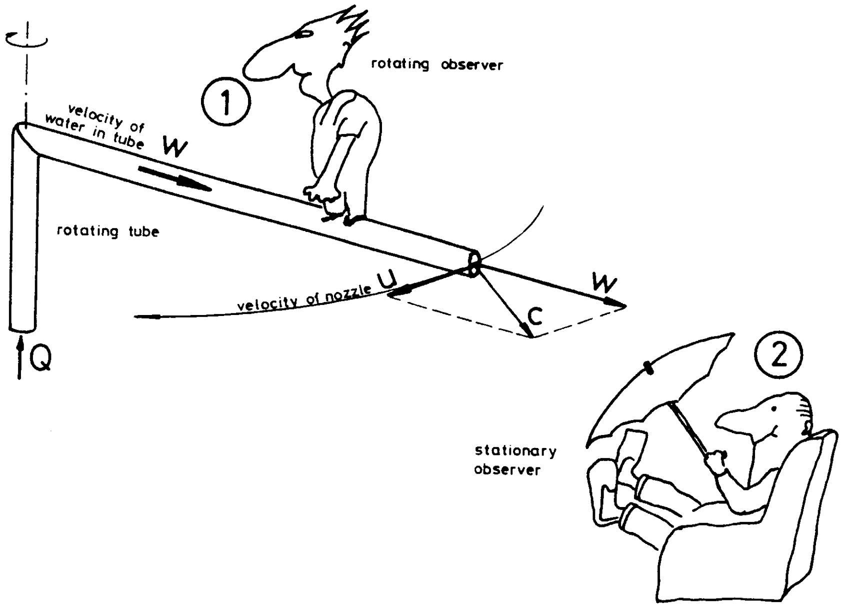


FIGURE B. 1 :
Sectional Views of a Centrifugal Pump

2.2 Velocity Vector Diagrams

The actual flow patterns through an impeller are very complex and it is necessary to make simplifying assumptions to describe this flow using velocity vector diagrams. Nevertheless, these diagrams still give a reasonable approximation of the actual flow.
When considering the flow through a passage between the blades of a rotating impeller, a distinction should be made between the **absolute flow** (velocities and paths with respect to the stationary walls of the casing) and the **relative flow**, considered with respect to the rotating impeller.



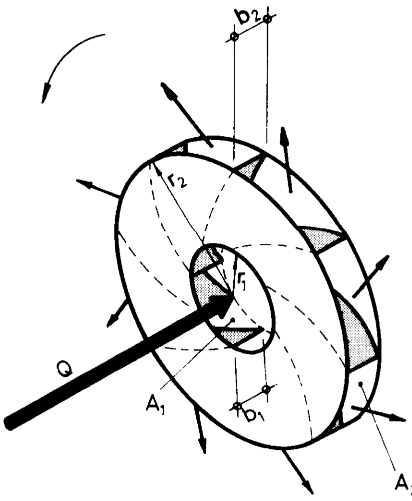
relative system (1):
flow passes the rotating observer at velocity w

absolute system (2):
flow approaches stationary observer at velocity c (vectorial addition of w + u)

Figure 2 shows the velocity vector diagram constructed for a centrifugal pump. The water enters the impeller in an axial direction and immediately turns so as to flow outwards in the plane of rotation. The water approaches the impeller vane with the absolute velocity c1, which is determined by the continuity equation

$$\boxed{c_1 = \frac{Q}{A_1}} \quad \text{with} \quad \boxed{A_1 = 2\,\pi\, r_1\, b} \qquad \text{(B .1)}$$

$c_1 = Q/A_1$, where A_1 = cross-sectional area of the impeller inlet passage with b_1 = impeller width at inlet and Q = nominal discharge of the pump.



If the impeller rotates at n revolutions/minute (rpm) the tangential velocity at the inner radius is

$$\boxed{u_1 = \frac{2\,\pi}{60}\,n\;r_1 = \omega\;r_1} \qquad \text{(B .2)}$$

The relative velocity w_1, i.e the speed of the water relative to the impeller (impeller considered as being at rest) is found by completing the vector triangle as shown. For the best flow pattern, the relative velocity w_1 should align with the vane angle β_1.

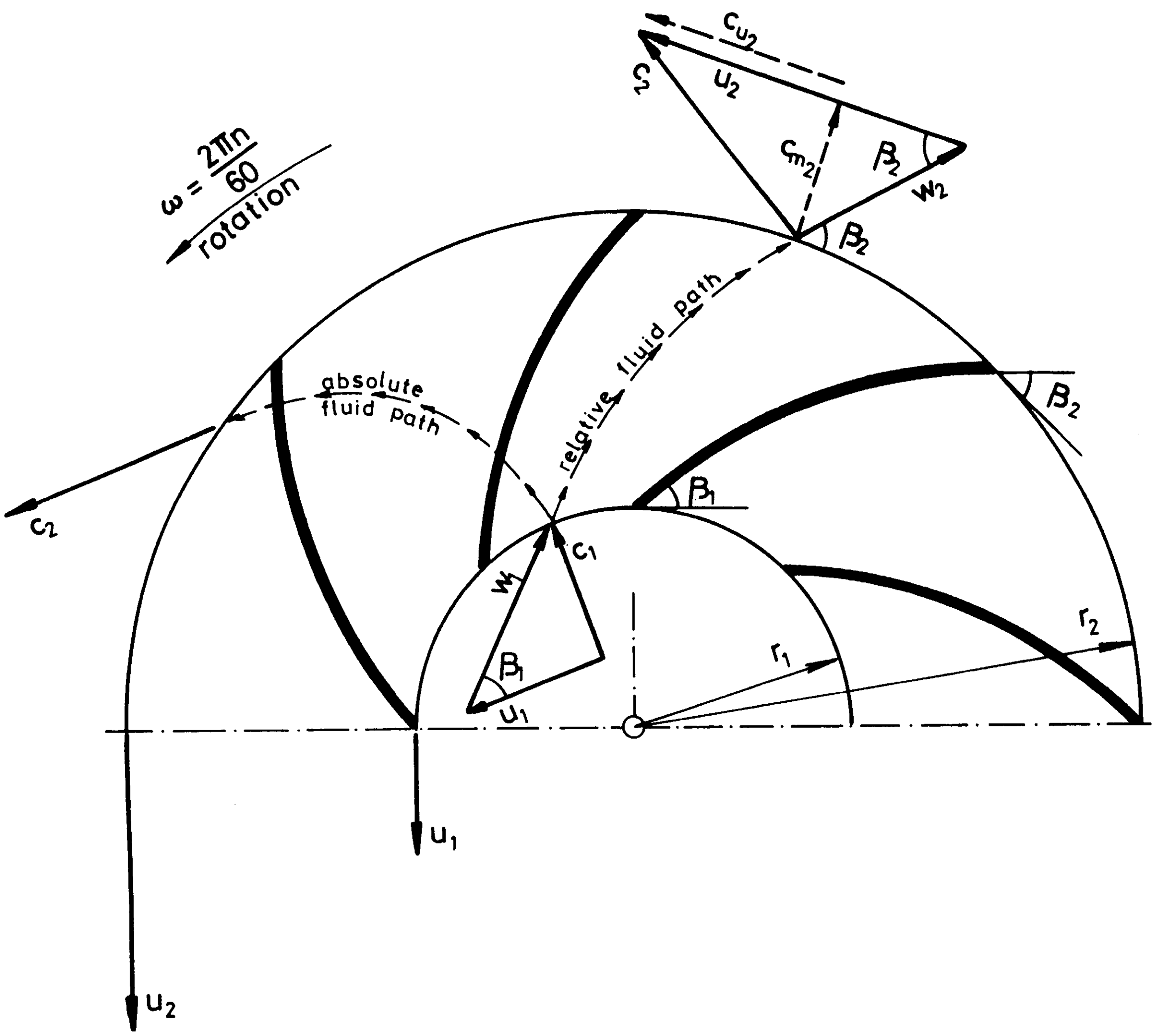


FIGURE B. 2 :
Velocity vector diagrams for a centrifugal pump

The water flows outwards in the passages between the vanes with its relative fluid path following the curvature of the vanes. It is discharged from the outer periphery of the impeller at the relative velocity w_2 and at the vane angle β_2. From this velocity, only its radial component, equal to c_{m2} can immediately be determined: applying the equation of continuity (see inlet triangle above) gives

$$\boxed{c_{2m} = \frac{Q}{A_2}} \quad \text{with} \quad \boxed{A_2 = 2\,\pi\,r_2\,b_2} \qquad \text{(B .3)}$$

The absolute tangential velocity of the impeller at the outer periphery (i.e. at r_2) is

$$\boxed{u_2 = \frac{2\,\pi}{60}\,n\,r_2 = \omega\,r_2} \qquad (B.4)$$

Using β_2, c_{m2} and u_2, the velocity triangle for the outlet periphery can be completed. The absolute velocity of the water c_2 is obtained by vectorially adding w_2 and u_2. Comparing c_2 with c_1 shows that the absolute velocity of the water has increased which is obviously the effect of the rotating impeller. An expression for the energy transferred from the impeller to the fluid will be developed in the next section.
One component in the vector diagram is to play an important role for further analysis: it is the tangential component c_u of the absolute velocity of the fluid, often referred to as the **whirl component**. Generally, absolute velocities at both impeller inlet and outlet periphery can have a whirl component; however, single stage centrifugal pumps are assumed to operate under whirlfree inlet velocities as shown in Figure 2 above (c_{u1} = 0 and $c_{m1} = c_1$). In practice, a slight pre-whirl in the suction pipe is usually produced by the rotating runner.

2.3 The Euler Equation for Energy Transfer

2.3.1 Basic Concept : Conservation of Momentum

Momentum of a body is the product of its mass and velocity. Newton's second law of motion states that the resultant external force acting on a body in any direction is equal to the rate of change of momentum of the body in that direction:

$$\boxed{F_x = = F_{1x} - F_{2x} = m\,\frac{d\,v_x}{d\,t}}$$ since for fluid flow

$$m = \rho\,Q\,d\,t$$

$$\boxed{F_x = \rho\,Q\,dv_x = \rho\,Q\,(\,dv_{1x} - dv_{2x}\,)} \qquad (B.5)$$

(for the component in y - direction, the same formula applies)

A bend of constant diameter (NB 225 mm) deflects the flow through an angle of 45°; this bend is installed in a pipeline system with a nominal discharge of 100 l/s (see Appendix A). Neglecting friction, the pressure at both ends of the bend is 1 m water column.
Figure 3 shows the bend cut out of the system and all external forces (here only pressure and momentum, gravity forces neglected) are acting on the bend. Vectorial addition of these forces gives the resultant force acting on the bend. To keep it in place, a concrete thrust block may be poured around the bend or, if the pipeline is rigid enough, the resultant force may be anchored at another place.

$F_{1,pressure,x} = h*\rho*g*A_1 = 1.0*1000*9.81*0.225^2*\pi/4$ = 390 N

$v_{1x} = I\,v_2\,I = Q/A = 0.1/(0.225^2*\pi/4) = 2.52$ m/s

$F_{1,momentum,x} = m*v/dt = \rho*Q*v = 1000*0.1*2.52$ = 252 N

$F_{1,x}$ = 642 N

The values of the forces at the outlet section are equal to those at the inlet, but their vectors are different:

$F_{2,y} = 642 * \sin 45^0 = 454$ N

$F_{res,y} = 454$ N

$F_{2,x} = -642 * \cos 45^0 = -454$ N

$F_{res,x} = 642 - 454 = 188$ N

$$F_{res} = \sqrt{F_{res,y}^{\ 2} + F_{res,x}^{\ 2}} = 491 \text{ N at } 67.5^0$$

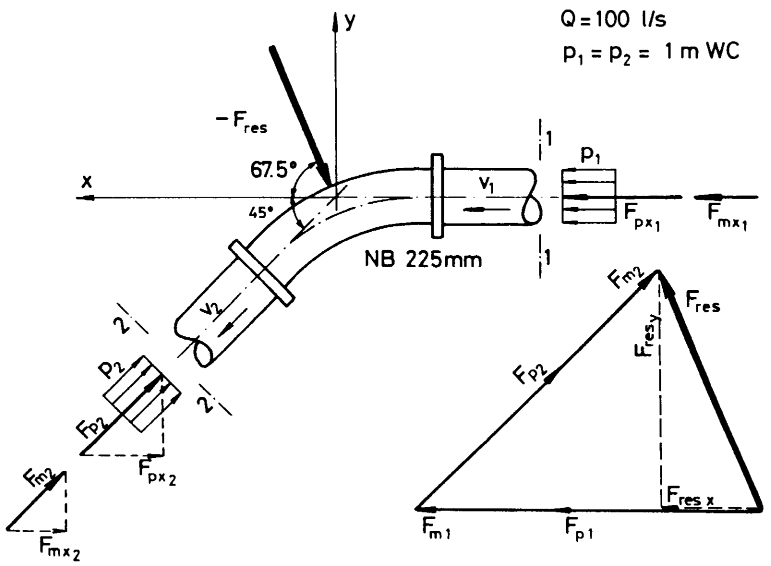


FIGURE B. 3 :
Application of the momentum equation on a pipe bend

2.3.2 Derivation of the Euler Equation

The same principle of conservation of momentum as shown in the previous section is applied to the impeller of a centrifugal pump. All forces acting on two sections of the flow path (here inner and outer periphery of the impeller) are considered. For the circular system, the forces due to pressure for both the outer and the inner periphery of the impeller as well as the radial components of the impulse forces (change of momentum due to velocities $c_{m,1}$ and $c_{m,2}$) cancel each other out because for each vector, an equal force but of opposite direction acts on the opposite half of the impeller.

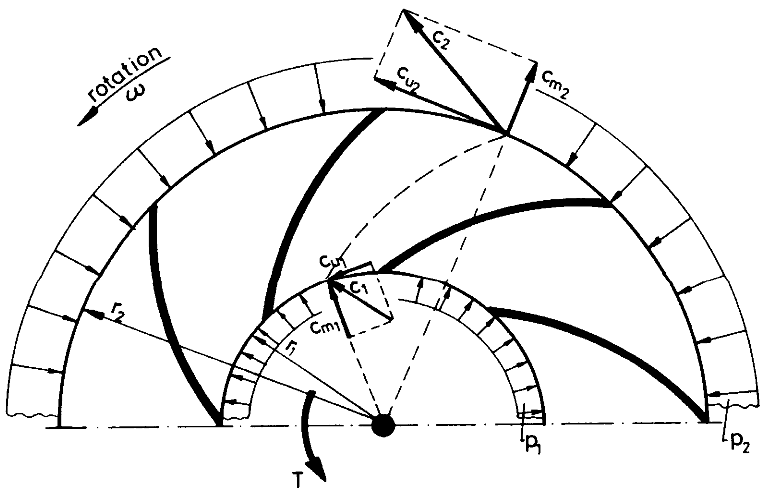


FIGURE B. 4 :
Derivation of the Euler Equation

The water entering the impeller per second at the inner periphery possesses the tangential momentum $\rho *Q*c_{u1}$; similarly the water leaving the impeller at the outer periphery possesses the momentum $\rho *Q*c_{u2}$. If tangential momentum is multiplied by radius, a moment of momentum is obtained: $\rho *Q*c_u*r$.
The rate of change of moment of momentum between inner and outer periphery is equal to the torque applied on the pump shaft: $T = \rho\ Q\ c_{u2}\ r_2\ -\ \rho\ Q\ c_{u1}\ r_1$

$$\boxed{T = \rho\, Q\, (c_{u2}\, r_2 - c_{u1}\, r_1)} \quad \text{(Nm)} \qquad \text{(B .6)}$$

Power = rate of energy transfer (for a linear system: force * distance moved in the direction of the force per unit time = force * velocity)
and for rotating systems: P = torque * rotational speed

$$\boxed{P = \omega\, T = \omega\, \rho\, Q\, (c_{u2}\, r_2 - c_{u1}\, r_1)} \quad \text{with} \quad \boxed{\omega = \frac{2\pi n}{60}}$$

$$\boxed{P = \rho\, Q\, (c_{u2}\, u_2 - c_{u1}\, u_1)} \quad \text{since} \quad \boxed{u = \omega\, r} \qquad \text{(B .7)}$$

which represents the energy transferred from the impeller to the liquid per unit time.

Another expression for power is work done per second (rate of doing work). Since the pump is designed to lift a mass of fluid through the pump head H, the power of the machine can also be expressed by: $P = g\, H\, m\, /dt$ and since $m/dt = Q\, \rho$ we obtain $P = \rho\ g\, Q\, H$. Inserted above leads to the Euler Equation:

$$P = \rho\, g\, Q\, H = \rho\, Q\, (c_{u2}\, u_2 - c_{u1}\, u_1)$$

$$\text{EULER EQUATION :} \quad \boxed{H_E = \frac{1}{g}\, (c_{u2}\, u_2 - c_{u1}\, u_1)} \qquad \text{(B .8)}$$

H_E represents the ideal increase in total head (in m fluid column) of the fluid due to the action of the impeller; i.e., the increase of head when all losses are ignored.
For real machines, a hydraulic efficiency is introduced to take account of the inevitable losses of the machine:

$$\boxed{\eta_{hydr} = \frac{H}{H_E}} \quad \text{Pump} \qquad \text{(B .9)}$$

(for further details on efficiencies see section 3.2)

2.3.3 Effect of Impeller Vane Angle

According to the Euler Equation, the increase of total head is only dependent on the whirl components c_u of the absolute velocities c at inlet and outlet of the impeller. The whirl component c_{u1} at the impeller inlet is determined by the layout of the system upstream of the pump (suction side) and is zero for a single stage pump with free suction inlet (no guide vanes). Thus, for whirl free inlet, the Euler Equation reduces to:

$$\boxed{H_E = \frac{1}{g}\, c_{u2}\, u_2} \qquad \text{(B.10)}$$

For a given discharge and impeller size ,it is theoretically possible to increase the transfer of energy by increasing the outlet vane angle β_2. This increase will be in the form of increase of kinetic energy since an increase in c_{u2} implies an increase in c.

Since the deceleration of the velocity c in the volute casing (transformation of kinetic energy into pressure) is always accompanied by losses, the velocity c and therefore the vane angle β_2 is kept within certain limits. Most centrifugal pumps (for pumping water) use a "backward" vane angle between 15 and 33^{o}.

2.4 Fluid Flow Through an Axial Flow Pump

The rotating element of an axial flow pump has the shape of a propeller. It is often mounted in a pipe or duct and the fluid passes in axial direction through the pump; it is not deflected away from the axis as with centrifugal pumps. The impeller consists of a number of blades attached to a central hub. The cross section of the blades are in **aerofoil form** (aeroplane wing) and the relative flow pattern is in many ways similar to flow around the wing of an aircraft. Due to the pressure increase on the concave side of the blade {see Bernoulli's equation: decreasing velocity along the blade (shorter path than on the convex side) invokes an increase of pressure}, an upthrust is produced by the flow which diminishes the torque to be applied on the pump shaft.
Figure 5 below shows the flow through an axial flow pump on the basis of velocity vector diagrams at the inlet and outlet sections. Energy transfer in axial flow pumps differs from that of centrifugal pumps (refer to Figure 2 above) in the following points:

- the fluid is not accelerated by centrifugal forces since it passes axially through the impeller and therefore tangential velocity of the impeller does not change for a given stream filament between inlet and outlet ($u_1 = u_2$);

- no real channel flow between blades is formed as with centrifugal pumps; hence, transformation of kinetic energy into pressure due to expansion of channel cross sectional area does not take place with axial flow pumps;

- similarly the blades are not much curved and deflection of flow is small.

For these reasons, the total head increase for axial flow pumps is small compared to centrifugal pumps. Energy transfer takes mainly place through the tangential acceleration of the fluid due to the impact of the impeller which can be seen by the fact that the fluid leaves the impeller in a vortex (c_2 is not in axial direction).

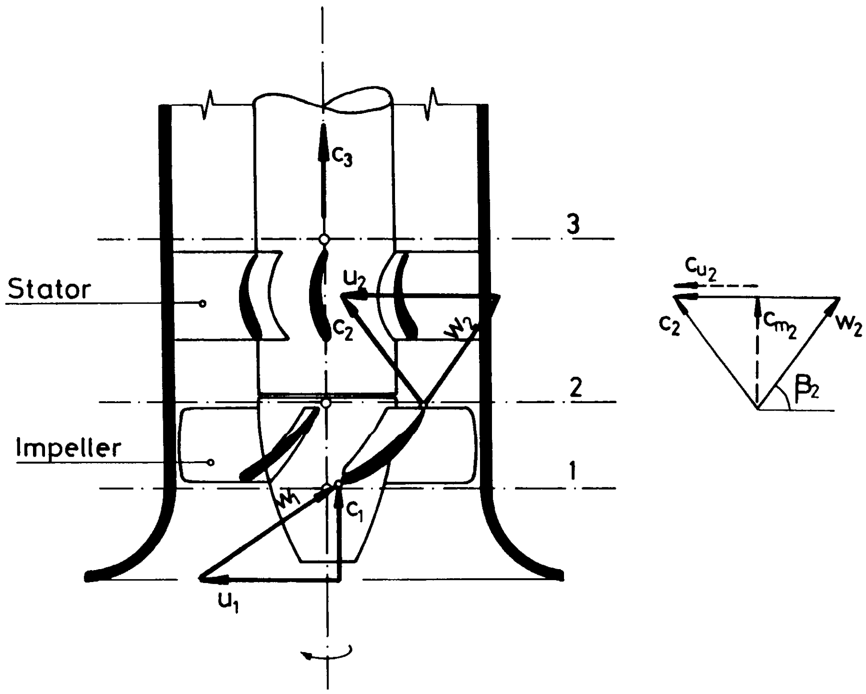


FIGURE B. 5 :
Velocity vector diagrams for an axial flow pump

Applying the momentum principle on the axial flow pump leads to the same Euler Equation as derived for a radial-type impeller: $H_E = 1/g*(c_{u2}*u_2 - c_{u1}*u_1)$. Since u is the same for inlet and outlet and c_{u1} is zero (provided the fluid approaches the impeller whirlfree), the **Euler Equation for axial flow pumps** reduces to:

$$\boxed{H_E = \frac{1}{g} c_{u2} u} \qquad (B.11)$$

Note that this equation only applies for one specific radius since u certainly varies with radius and c_{u2} may also change. In order to estimate the performance of the impeller, it is necessary to integrate between hub and tip (in practice a mean radius may be chosen as the representative flow pattern for the whole pump or a few sections are computed and then summed up). For each specific radius, shape and angle of the blade may be optimized which results in the **twisted blades** often installed in axial flow pumps. The blade angle β_2 is usually between 30 and 45^o for axial flow pumps pumping water.

Figure 5 above shows a set of stationary blades, known as **stator**, just downstream of the impeller. The stator blade angle is designed to straighten the flow so that the fluid leaves the stator in axial direction. For reasons of continuity, velocity c_3 must be equal to c_1 and therefore a reduction of kinetic energy between impeller outlet (velocity c_2) and stator (c_3) must have taken place. Applying Bernoulli's equation (see Appendix A) it can be shown that a reduction in kinetic energy must be compensated by a corresponding increase of pressure. Thus the effect of a stator is to improve overall pressure rise attainable by the pump.

Inferior to centrifugal pumps in total head, axial flow pumps are applied for **low head installations** where **large quantities** can be pumped at good efficiencies (large flows imply high velocities and consequently increase losses in the pump; due to the short and relatively straight flow pattern, losses in an axial flow pump are much smaller than in a comparable centrifugal pump of the same discharge).

Mixed flow pumps have a design somewhere in between axials and centrifugal pumps. They lift and at the same time accelerate the water. Mixed flow pumps are used for intermediate lifts over a wide range of flow rates. The design theory is similar to centrifugal pumps since no aerofoil effects are involved.

2.5 Fluid Flow through a Turbine

2.5.1 General

There are three basic types of water turbines: Francis, axial flow (Kaplan) and the Pelton. In the course of this manual on Pumps Used As Turbines, only the first two are dealt with since the Pelton is an impulse turbine which operates in air. There is unfortunately no pump design possible which could work on the same principle if run in reverse; the Pelton is therefore not relevant for this manual.

The design of a **Francis turbine** appears similar to that of a centrifugal pump. However, the water is fed into the outer casing section and flows inwards through the runner. Thus the flow and energy transfer in the turbine is the reverse of that for a pump. Another difference is that the turbine casing incorporates guide vanes around the runner which are adjustable to provide always an optimum angle between the approaching flow and the runner under varying conditions of flow and power output.

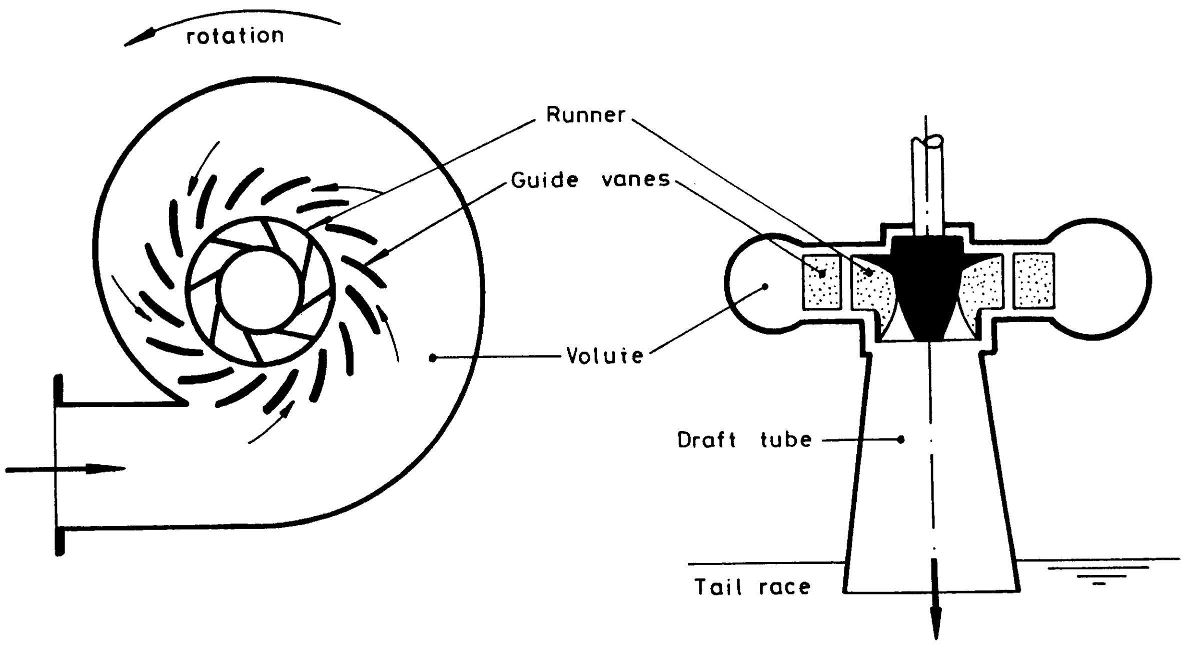


FIGURE B. 6 :
Sectional Views of a Francis Turbine

On entering the runner, the flow changes direction due to the curved vanes which corresponds to a change of tangential momentum. As explained in section 2.3.2 above this deflection imparts a torque on the runner, hence providing the power for a generator or mechanical machinery.
Note that the flow is accelerated on its way through the runner because circumferential cross section of the runner diminishes with the radius. Since accelerated flow is not subjected to shock losses as is decelerated flow (see centrifugal pumps above), change of velocity can be done quickly. The channels of the runner can therefore be kept relatively short and straight which results in a smaller runner as compared to a radial impeller of a pump.
The water is discharged into the draft tube and drained away into an open channel or tail race. The draft tube permits the head between turbine shaft and tail water level to be used for power generation. As the weight of the water column in the draft tube produces a negative pressure at the turbine outlet, the total pressure difference between turbine inlet and outlet flanges increases (see also Appendix A, pressure lines, section 6.2). The draft tube has the shape of a diffusor so as to minimize the loss of kinetic energy in the discharge to the tail race.

2.5.2 Euler Equation for Turbines

As for 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:

- flow approaching the runner (inlet triangle): the **main parameter is the guide vane angle α_2** it determines the shape of the velocity vector diagram at the inlet periphery. Note that the runner blade angle β_2 is not of primary importance for turbine performance! (see velocity vector diagrams in Figure 7)

- flow leaving the runner (outlet triangle): the shape of the velocity vector diagram at the inner periphery of the runner is mainly determined by the blade angle β_1.

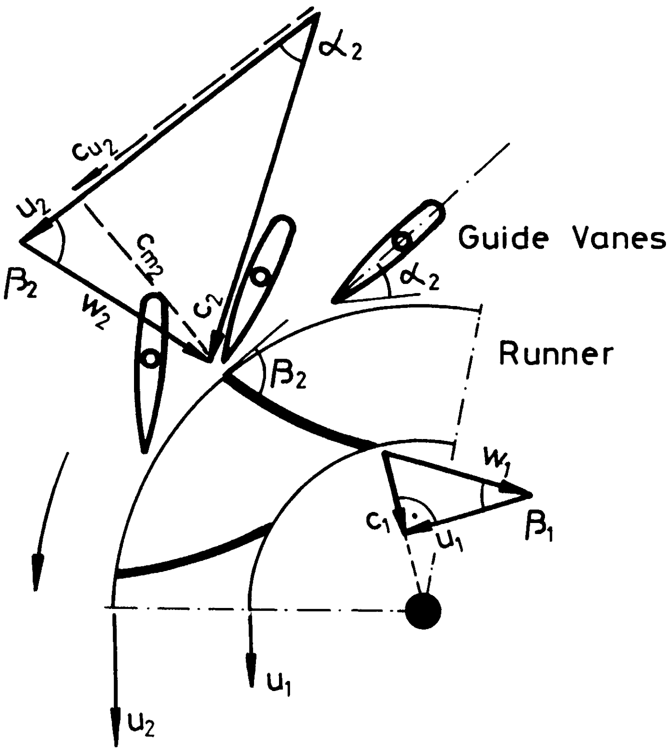


FIGURE B. 7 :
Velocity vector diagrams for a Francis turbine

Similar to pumps, the concept of conservation of the moment of momentum is applied on all torques acting on the turbine runner. Figure 7 above shows that only the tangential components of the inlet and outlet velocities cu_2 and cu_1 contribute to the rate of change of momentum which constitutes the torque T acting on the turbine shaft. All other forces and momentums (pressure, radial components of the velocities c) act radially on the runner and do not produce torque. We can write immediately:

$$\boxed{T = \rho\, Q\, (c_{u_2} r_2 - c_{u_1} r_1)} \qquad \text{(B .12)}$$

By comparing this equation with the derivation of the Euler Equation for centrifugal pumps above, we can show the validity of the Euler Equation for turbines which writes in its common form:

$$\boxed{H_E = \frac{1}{g}\,(c_{u_2} u_2 - c_{u_1} u_1)} \qquad \text{(B .13)}$$

H_E*g represents the massic energy ideally transferred from the fluid to the runner of the turbine; i.e. the energy generated by the turbine when all losses are ignored.
For real machines, a hydraulic efficiency is introduced to take account of the inevitable losses of the machine:

$$\boxed{\eta_{hydr} = \frac{H_E}{H}} \quad \text{(TURBINE)} \qquad \text{(B .14)}$$

Note that the hydraulic efficiency for turbines is the inverse of that of pumps; to generate the ideal massic energy in a turbine (H_E), the real (net) head (H) acting on the turbine must be bigger to account for the internal turbine losses, whereas for a pump, the real head (H) produced by the ideal energy transfer (H_E) will always be smaller due to the internal pump losses.

2.5.3 The effect of the guide vane and runner blade angles

According to the Euler Equation the ideal energy transfer in a turbine is only dependent on the whirl components c_u of the absolute velocities at the runner inlet and outlet.

Inlet triangle:

For a given flow and speed of the machine, the guide vane angle α_2 and the runner blade angle β_2 can only take distinct values dependent on each other. Deviating from these would cause a misalignment between the runner blades and the relative flow component w_2 at the runner inlet with subsequent losses (for a more detailed treatment of losses see section 3.2.2 below).
The same misalignment of velocity vectors and runner blades would occur if flow were increased or reduced by means of a throttling valve for example (see Figure 8b). If, however, flow is controlled by closing the guide vanes (angle α_2 to α_{2x}) the relative velocity w_2 remains in perfect alignment with the runner blade angle β_2. The great advantage of adjustable guide vanes becomes obvious: optimum direction of flow approaching the runner and therefore high efficiency can be maintained under varying conditions of flow and of power output.

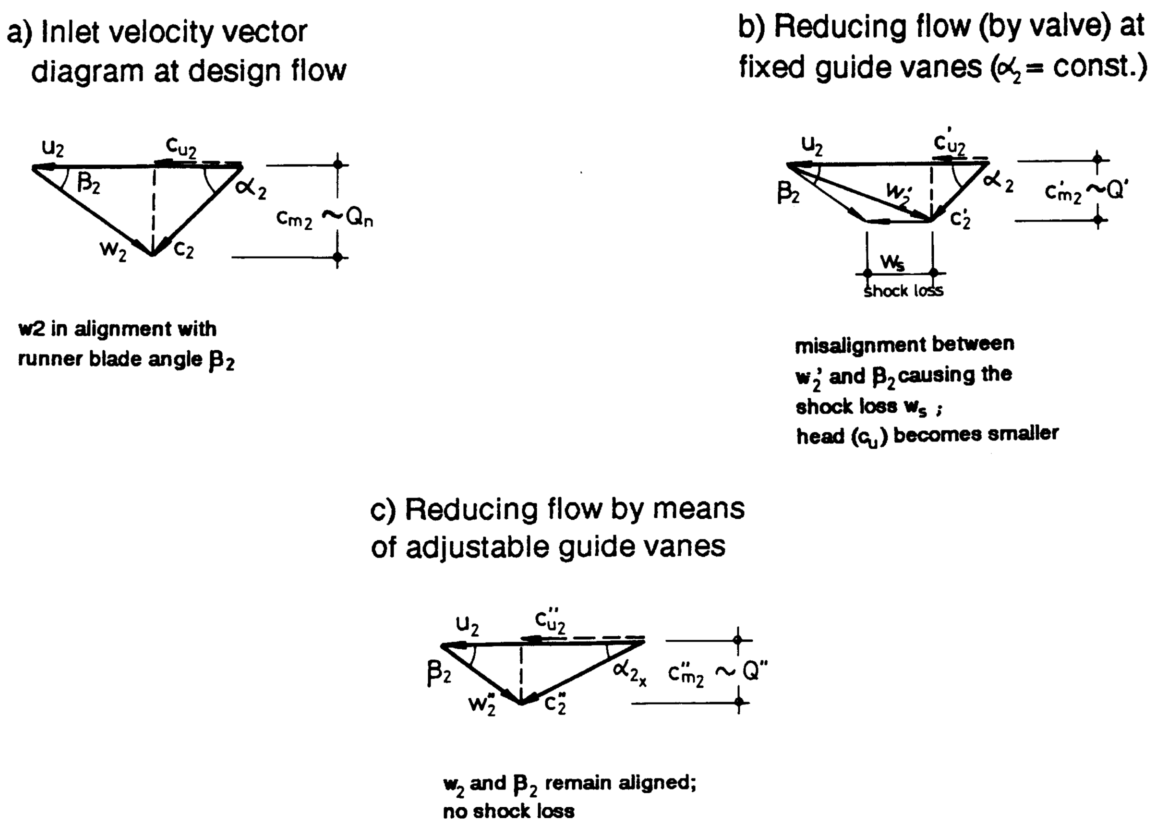


FIGURE B. 8 :
Flow control by means of adjustable guide vanes and its effect on the inlet velocity diagram

Outlet triangle

As can be seen from the Euler Equation, the tangential or whirl component c_{u1} of the outlet velocity c_1 reduces the ideal energy transfer in a turbine. Thus, whirl free outlet would theoretically produce optimum turbine performance. The runner blade outlet angle β_1 is therefore designed to provide a whirl free outlet velocity, i.e., $c_{u1} = 0$ and c_1 in radial direction at design flow and speed (see Figure 7 above).

3. PERFORMANCE OF PUMPS AND TURBINES

3.1 Parameters of pump and turbine performance

The principal parameters of performance of hydraulic machines are the following:

- discharge Q (or mass flow) in m3/s (kg/s)
- actual head H (total delivery head, TDH, for pumps; net head H_n for turbines) in m liquid column
- power P supplied to (pump) or delivered from (turbine) the shaft of the machine (mechanical power) in kW
- rotational speed n of the machine (revs/minute)
- efficiency η of the machine in % which is the ratio of power transferred to/from the fluid = ρ *g*Q*H and shaft power P
- guide vane opening α (for turbines only) in % to maximum

A machine is tested over its range of discharge and the performance data may then be plotted versus discharge Q. Two distinct features of turbo machines can be shown (see figure 9 below):

a) contrary to positive displacement pumps (e.g. piston pumps) discharge varies with head although the rotational speed of the machine is kept constant. If the discharge of a pump is throttled by a control valve in the discharge pipeline, the pump head increases and vice versa. (This can be explained using the velocity vector diagrams of Figure 2 above where the radial component c_m of the absolute velocity c represents flow Q and the tangential component c_u represents head H. A reduction in Q and therefore c_m implies an increase in c_u, hence head H.

b) the machine can be operated with the valve completely closed but no excessive pressure is produced as e.g. with piston pumps. The reason for this has already been given in Figure 4 above for centrifugal pumps: the torque transferred between fluid and impeller/runner is independent of the pressure since it acts radially on the circular sections of the impeller; thus pressure cannot rise beyond a certain limit corresponding to the energy of the accelerated fluid in the impeller (referring to the velocity vector diagrams: $H_{max} = u_2^2/g$).

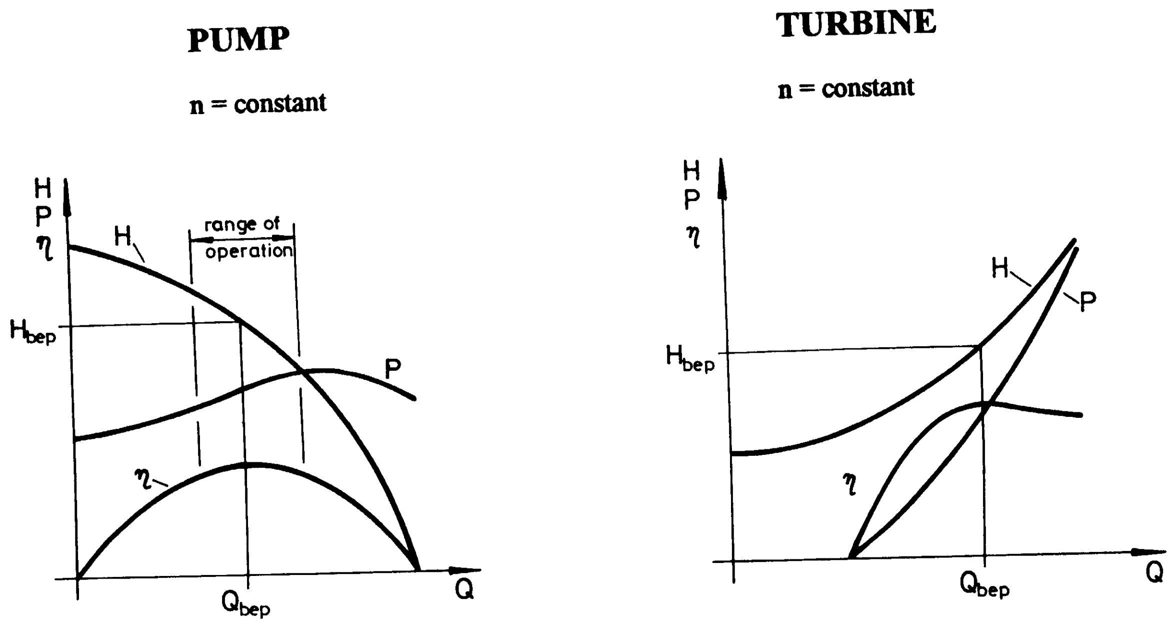


FIGURE B. 9 :
Typical performance curves of pumps and turbines

For the design of a hydraulic installation, it is not sufficient to know the performance data for the nominal (design) speed only. When studying transient behaviour of a machine in a given installation (starting, stopping, load changes), the performance at other than nominal speed is required. Performance curves at different speed are known as hill charts where points of constant efficiencies are extended into contour lines around the best efficiency point (bep = operating conditions H, Q and n with maximum efficiency) of the machine (see Figure 10).
The same presentation in hill charts is used for a series of performance curves of variable impeller diameters (trimming of pump impellers) or variable guide vane openings (turbine).

PUMP **TURBINE**

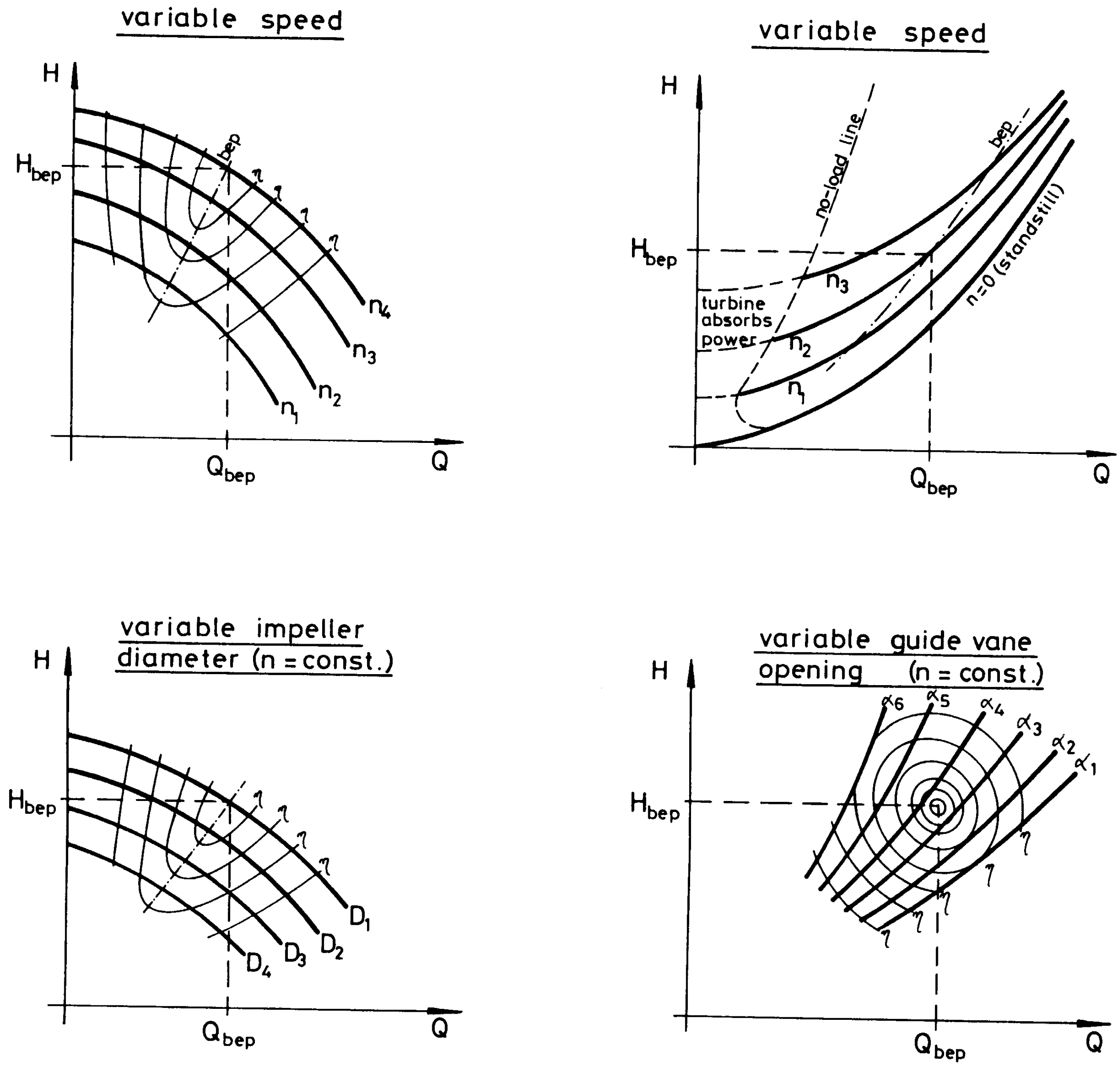


FIGURE B. 10 :
Hill Charts

3.2 Energy Losses in Pumps and Turbines

3.2.1 General

The energy transfer predicted by the Euler Equation is not attained in practice. The hydraulic efficiency introduced in section 2.3.2 above quantifies the losses between fluid power and mechanical power of the impeller/runner. However, these are not the only losses occurring in a hydraulic installation. Figure 11 summarizes these for both a pump scheme and an MHP.

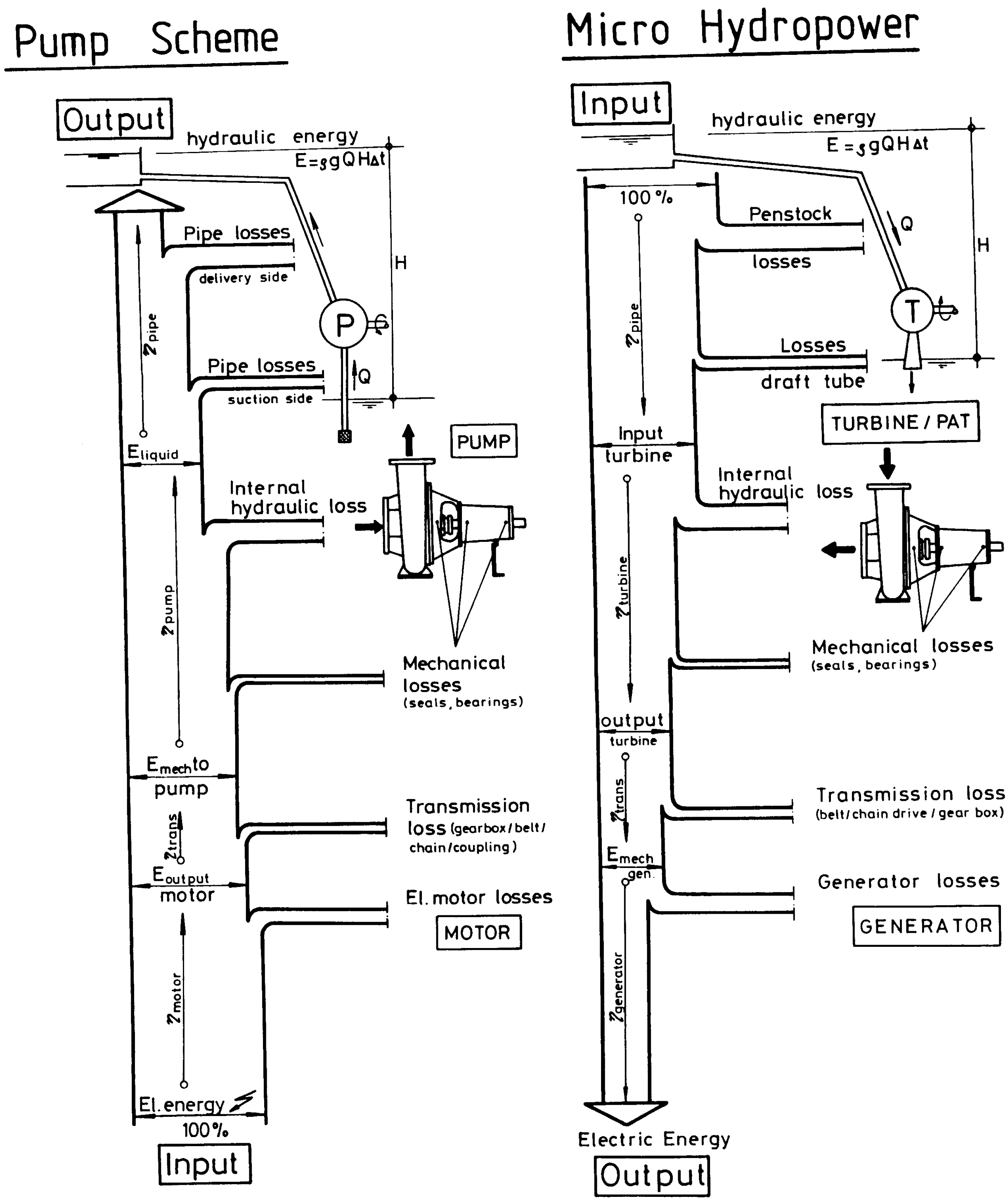


FIGURE B. 11 :
Energy losses in pump and micro hydropower schemes

3.2.2 Internal Hydraulic Losses

Apart from the pipeline losses which has been dealt with in detail in Appendix A, the internal hydraulic losses of the machine are the largest single losses; they account for some 20 % of machine input (shaft power for pumps, liquid power under net head for turbines). Internal hydraulic losses are comprised of the following five parts which each reduce the ideal energy transfer according to Euler's equation:

a) the velocity distribution in the channels of the impeller is not uniform; a **secondary flow** pattern is produced by the rotating impeller vanes which reduces the tangential velocity component c_{u2}. The impeller applies a tangential force on the fluid which implies that the pressure at the back of the vanes is lower and at the front, it is higher. The velocity vector w_2 therefore leaves the impeller at another less favourable angle than predicted by the Euler equation which assumed an infinite number of blades. This reduction of energy transfer to the fluid is called **circulation loss** and is due to the **finite number of blades** in a pump or turbine. The circulation loss of a turbine is negligible since it occurs at the inner periphery of the runner which has generally only a minor effect on turbine performance.

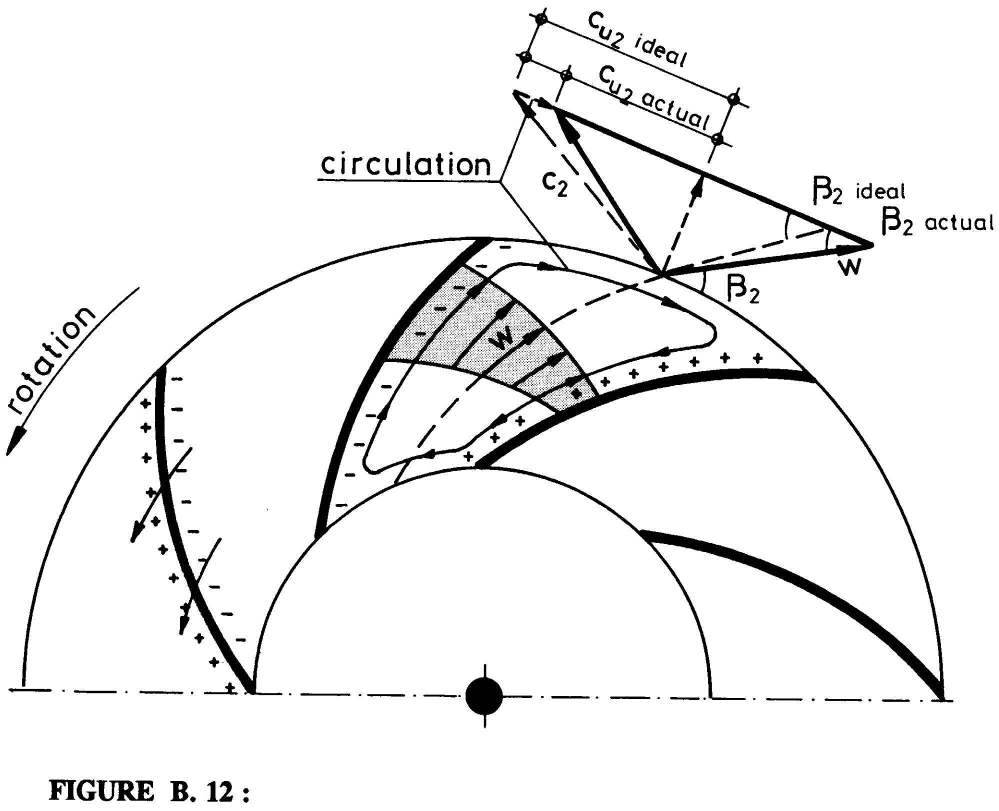


FIGURE B. 12 :
Distribution of relative velocity in impeller channels and its influence on the whirl component of the leaving fluid

b) as the flow passes through the impeller channels and the casing, friction and, in the case of pumps, losses due to deceleration or diffusion process occur; these losses increase with the square of the flow Q (see Figure 13 below);

c) the vane angle of the impeller and the angle of the guide vanes/discharge casing of a machine are determined for a certain flow, the design flow; for any other discharge Q, misalignment between the direction of flow and the angle of the volute casing and the runner
blade angles occurs. This misalignment invokes another velocity vector the so-called shock loss which is the reason for decreasing efficiencies of machines for flows other than design flow. At the design flow, shock losses are zero and increase with the square of the incremental flow for operation off the design flow. Figure 13 below shows the velocity vector diagrams for both a pump and a turbine.

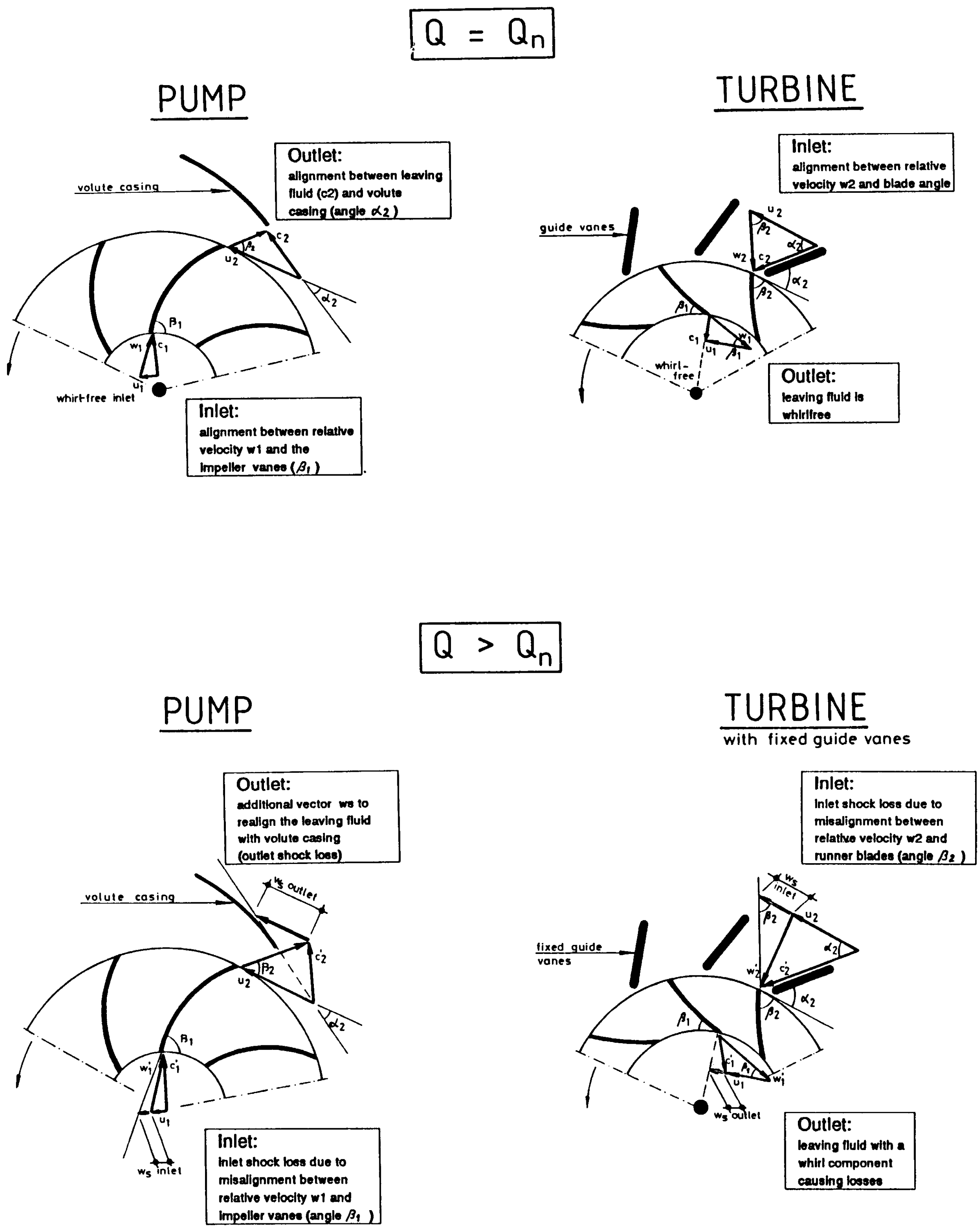


FIGURE B. 13 :
Shock losses due to pump/turbine operation off the design flow

d) further **friction** losses occur in the fluid film between the rotating impeller and the stationary casing;

e) there is a certain **leakage** of fluid from the high pressure side of the machine to the low pressure side through the small passages (clearance) between the impeller/runner and the casing; fluid leaking back is a waste of energy for both pumps and turbines;

Figure 14 shows the performance curves of a pump and a turbine which have been plotted not from test data but by reducing/adding the losses as explained above for every discharge from the ideal Euler line.

$H_E = 1/g*(u_2 * c_{u2} - u_1 * c_{u1})$

PUMP: The slope of the ideal Euler line for a pump only **depends on the outlet vane angle β_2** (hence on c_{u2}) if whirlfree inlet is assumed. At zero flow the maximum head generated by a pump is $H_{max} = u_2\ 2/g$. The losses (circulation, friction, shock) must be **subtracted** from the ideal Euler line.

TURBINE: The ideal Euler line for a turbine is a little bit more complicated since the outlet flow of the turbine is whirlfree only for the design flow, whereas for all other flows the fluid leaves in a whirl and the second term of the Euler equation ($u_1 * c_{u1}$) must be considered. The first term of the Euler equation gives a straight line (dotted line in Figure 14) through zero (at no flow, c_{u2} is zero too due to the constant guide vane angle α_2) and with a slope **depending on the guide vane angle α_2**. The losses (friction, shock) must be **added** to the ideal Euler line (note the difference to the pump).

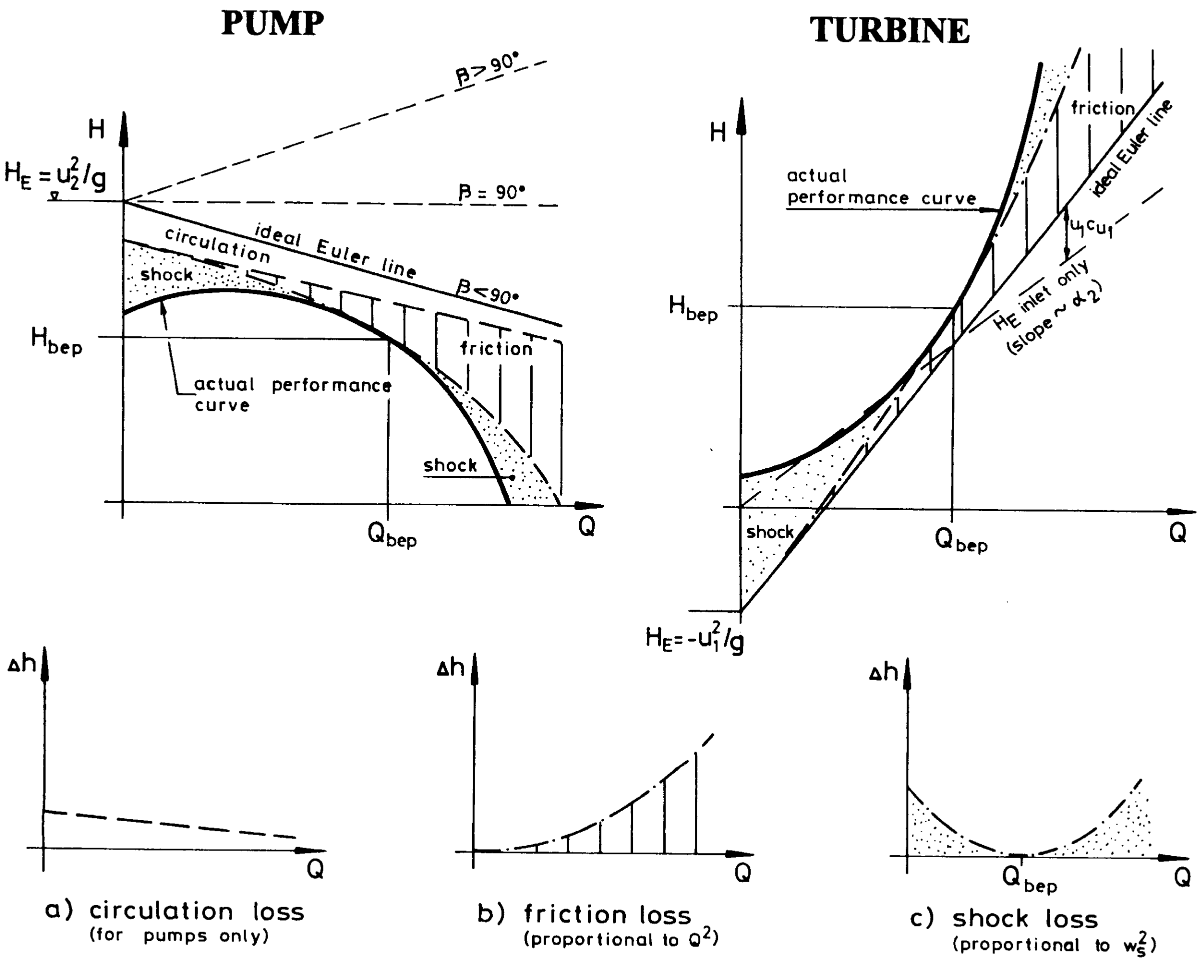


FIGURE B. 14 :
Development of the performance curves of pump and turbine: ideal Euler line and real performance

3.3 Affinity Laws

Looking at the characteristics of pumps and turbines in Figure 10 above it becomes clear that there must be a certain relationship between the different performance curves of a specific machine at **different speed**. This relationship is described by the affinity laws. These laws can be derived from the velocity vector diagrams (see Figure 2 above). By changing the rotational speed, the absolute tangential velocity u of both inlet and outlet sections of the impeller changes. It is clear that all other components of the velocity triangles (representing head and flow) must be adjusted in such a way that the triangles remain similar otherwise the machine would no longer operate with an optimum flow pattern, i.e., alignment of flow and machine geometry. Similarity considerations on the velocity triangles therefore lead to the affinity laws.

The laws are as follows: (Index 1: initial/known conditions; Index 2: new conditions of which only one parameter is known)

$$\boxed{\frac{Q_1}{Q_2} = \frac{n_1}{n_2}} \qquad \text{(B .15)}$$

$$\boxed{\frac{H_1}{H_2} = \left(\frac{n_1}{n_2}\right)^2} \quad \boxed{\frac{T_1}{T_2} = \left(\frac{n_1}{n_2}\right)^2} \qquad \text{(B .16)}$$

$$\boxed{\frac{P_1}{P_2} = \left(\frac{n_1}{n_2}\right)^3} \qquad \text{(B .17)}$$

(machine and impeller size and shape must be kept constant)

If performance curves are available for one condition, an approximation of the performance at another condition can be obtained by developing a new pump curve using the above laws. Note that this is only an approximation which yields accurate results for speeds deviating little from the design speed (this method assumes that efficiencies do not change from one condition to the other which is not always true as can be seen from the hill chart in Figure 10 above).

A second law based on the same principle of similarity is used to estimate the performance of a pump with **varying impeller diameter** while speed is held constant (note that this second set of laws applies to centrifugal pumps and Francis turbines only):

$$\boxed{\frac{Q_1}{Q_2} = \left(\frac{D_1}{D_2}\right)^3} \qquad \text{(B .18)}$$

$$\boxed{\frac{H_1}{H_2} = \left(\frac{D_1}{D_2}\right)^2} \qquad \text{(B .19)}$$

$$\boxed{\frac{P_1}{P_2} = \left(\frac{D_1}{D_2}\right)^5} \qquad \text{(B .20)}$$

where D = impeller diameter
(note, this is **not impeller trimming**; the shape of the impeller remains completely unchanged, but only increased or decreased in size)

3.4 Specific Speed

Hydraulic machines can be defined by the parameter which is called specific speed n_q and which reads:

$$n_q = n \frac{\sqrt{Q}}{H^{3/4}} \qquad n_{qi} = n \frac{\sqrt{Q_i}}{H_i^{3/4}} \qquad \text{(B .21)}$$

multistage pumps: number of stages i_{st}

$$H_i = \frac{H}{i_{st}}$$

multiflow pumps number of entries i_{fl}

$$Q_i = \frac{Q}{i_{fl}}$$

where n = nominal rotational speed in rpm

Q_i = nominal discharge in m3/s per single impeller at bep and

H_i = head (TDH for a single-stage pump or head produced <u>per</u> stage in a multistage pump; or net head for turbines) in m

<u>**all these parameters are measured at the best efficiency point (bep) of a given machine.**</u>

This expression is derived from the affinity laws above and may be interpreted as the speed at which a scaled model pump (or turbine) would deliver (or absorb) unit discharge (1 m3/s) at unit head (1m). It is used as a guide for determining the type of machine required for a given pumping or the turbine situation. The concept of specific speed assists the designer by indicating whether he should look at centrifugal, axial or mixed flow pumps. It is the result of years of experience and data analysis and represents the general consensus of this experience and data.

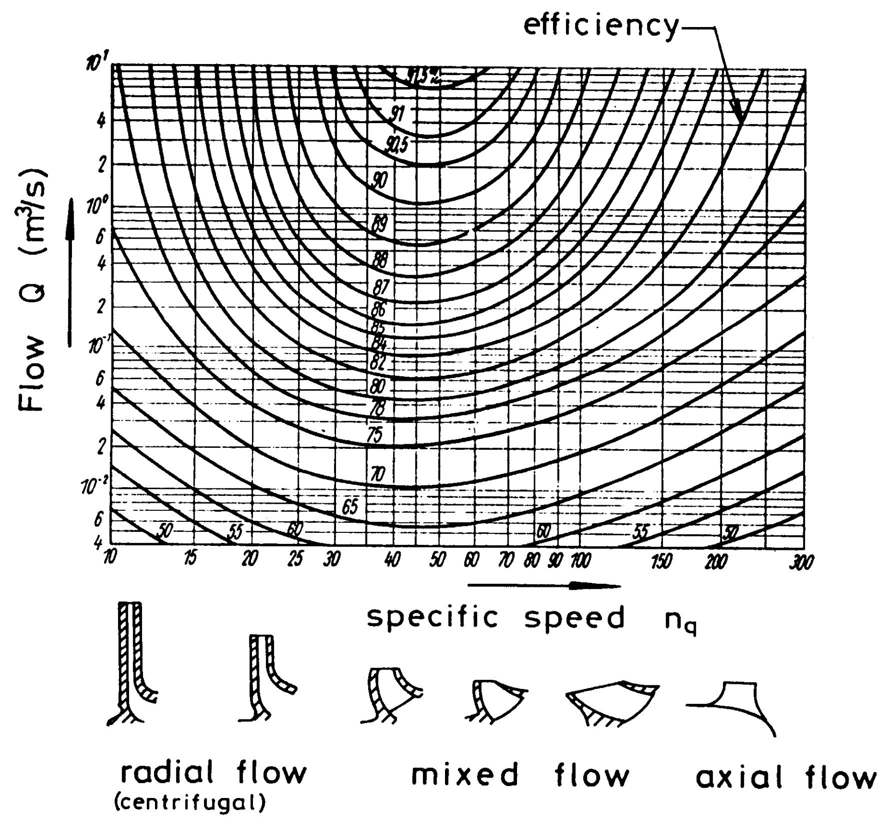


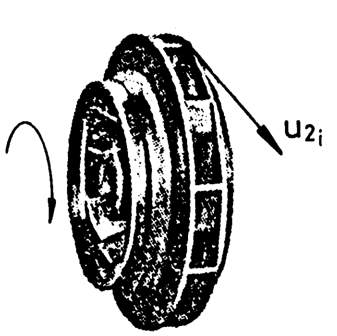
FIGURE B. 15 :
Pump efficiency versus specific speed and flow

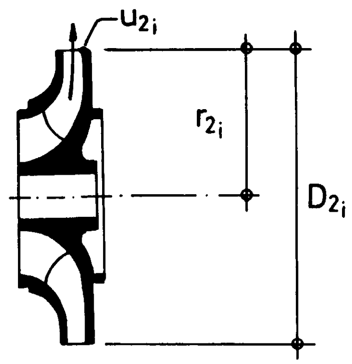
Based on the same practical experience mentioned above, it has been found that the **optimum tangential velocity coefficient** of a runner/impeller always takes about the same value (relative to the head) independent of specific speed:

$$\text{PUMPS:} \quad u_{2i} = 1.0 \ldots\ldots 1.1 \sqrt{2gH}$$

$$u_{2i} = \frac{2\pi n}{60} r_{2i}$$

u_{2i} in m/s and H = Total Delivery Head (TDH) in m

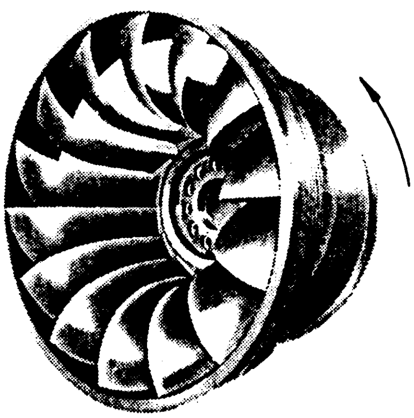


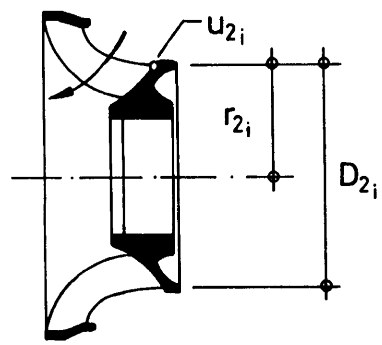


$$\text{TURBINES:} \quad u_{2i} = 0.75 \ldots\ldots 0.8 \sqrt{2gH}$$

$$u_{2i} = \frac{2\pi n}{60} r_{2i}$$

where H = net head H_n in m





Using these relationships, the size (diameter) of the impeller/runner required for a given installation can be determined without referring to manufacturers' offers since the impeller velocity is directly proportional to rotational speed (see formulae above on the right).

Or for a given impeller/runner diameter, the optimum rotational speed n of the machine can be determined using the same equations.

3.5 Installed Performance - Duty Point

So far, the hydraulic machine has been considered in isolation and the head was assumed constant for a given installation. This is not quite true since the performance of the machine and the characteristic of the hydraulic system (pipeline or penstock) are interdependent.
Similar to the performance curve of the machine, the characteristic of the hydraulic system can be found by plotting head versus discharge. Departing from a constant static head (or gross head for turbines) a system resistance curve can be drawn by adding/subtracting the losses for different flow rates. Since frictional as well as local losses of the pipeline / penstock are a function of $v^2/2g$ (see Appendix A) or $Q^2/(A^2*2g)$ the system curves take parabolic form.
Since the discharge through the machine is the same as the discharge through the pipe system, the head generated or absorbed by the machine must match the head required by the system. The point at which this occurs can be found by superimposing the system resistance curve on the performance curve of the machine. The intersection point of these two graphs is the <u>operating or duty point</u> of the installation.

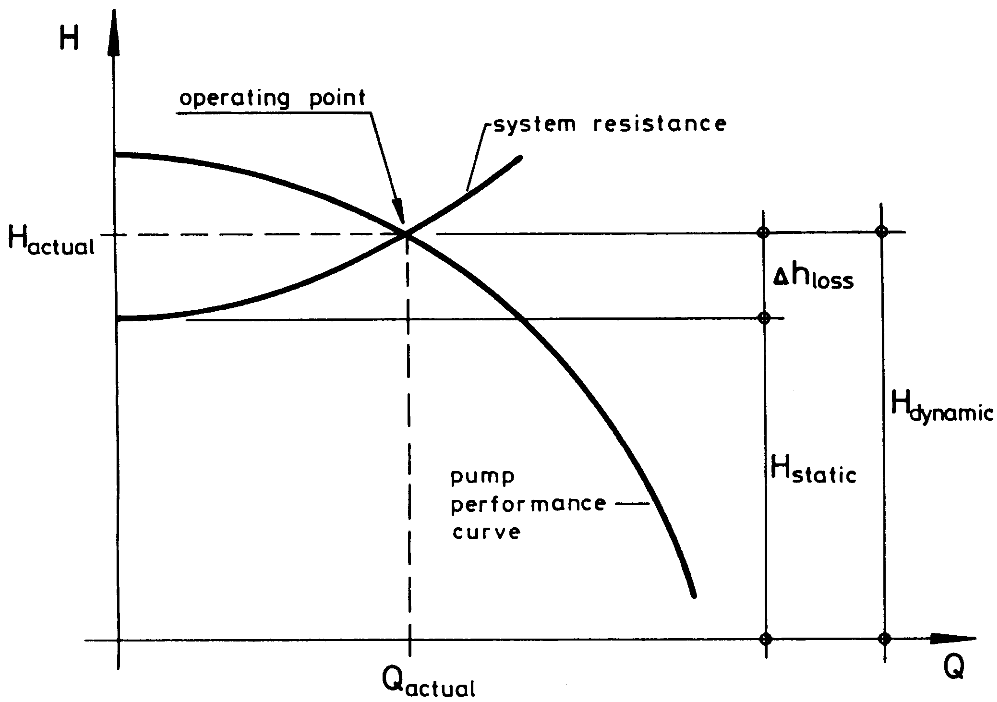


FIGURE B. 16 :
Operating point of a pump

4. WORKED EXAMPLES

4.1 Selection of a Pump

For the pump station defined in section 6.1 of Appendix A, a suitable pump is to be selected. The design conditions were as follows:

required duty point: Q_n = 100 l/s at TDH = 19 m

Using the concept of specific speed, the type of pump yielding optimum efficiency for the given installation can be determined:

$$n_q = n{*}Q^{1/2}/H^{3/4} \quad \text{with } n = \text{rotational speed in rpm}$$

It is assumed that the pump would be directly coupled to an electric motor (asynchronous motor) which limits the choice for the rotational speed n in above formula to the following values:

n = 2900 rpm (for a two pole motor at 50 Hz grid frequency)
n = 1450 rpm (for a four pole motor)

A motor with more poles (lower speed) or using a gear box is not considered an economic solution for this installation.

<u>with n = 2900:</u> $n_q = 2900 * (0.1)^{1/2} / (19)^{3/4} = 101$

According to Figure 15 above a diagonal flow pump should be chosen; efficiency can be expected to reach 80 %.
The size of the pump can be estimated with the relationship:

$$u_{2i} = 1.05\sqrt{2gH} = r_{2i}\, 2\pi \frac{n}{60}$$

$$r_{2i} = 1.05\frac{\sqrt{2gH}}{2\pi\ n/60} = 1.05\frac{\sqrt{2{*}9.81{*}19}}{2\pi{*}2900/60} = 0.07\text{m or outer diameter of the impeller } d = 14\text{ cm}$$

<u>with n = 1450:</u> $n_q = 1450 * (0.1)^{1/2} / (19)^{3/4} = 50$

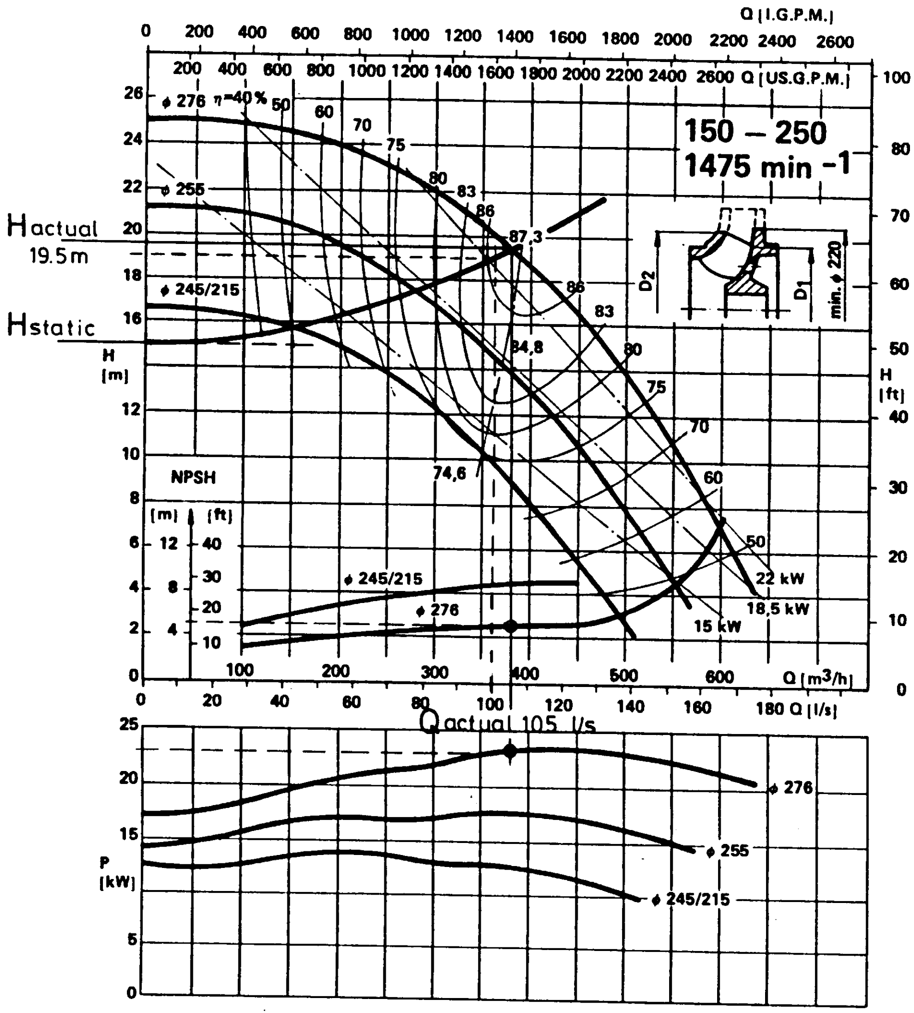
According to Figure 15 above a mixed flow pump should be chosen; efficiency can be expected to reach almost 85 %.
The size of the pump can be estimated with the relationship:

$$r_{2i} = 1.05\frac{\sqrt{2gH}}{2\pi\ {*}n/60} = 1.05\frac{\sqrt{2{*}9.81{*}19}}{2\pi{*}1450/60} = 0.14\text{ m or outer diameter of the impeller } d = 28\text{ cm}$$

The second possibility, n = 1450 rpm, would probably be chosen, since the efficiency of the mixed flow pump is slightly higher. This might compensate for the higher initial costs of the bigger pump and the 4-pole motor as compared to the high speed solution (n=2900 rpm).
A review of pump performance curves from several manufacturers reveals that the best pump for this situation is a single stage mixed flow pump with an impeller of 276 mm in diameter and a suction branch of NB 200 mm (pressure branch NB 150 mm). The performance curve of this pump is presented in Figure 17 below.
The system resistance curve is superimposed on the pump performance curve to obtain the operating point at the intersection of the two graphs. In Appendix A only one point of the system curve was calculated, namely $Q_{design} = Q_n$. Other points of the curve can be approximated with the following formula:

$$TDH_2 = H_{static} + H_{losses}{*}Q_2^{\,2}/Q_n^{\,2}$$

Note that this formula gives only accurate results for flows close to Q_n; to plot the <u>whole</u> system curve, the method described in Appendix A must be applied for different values of Q.



Laufradbreite
largeur de roue $b_2 = 45$ mm
impeller width

Saugstutzen-Tuyau d'aspiration-Suction branch **DN 200**

FIGURE B. 17 :
Operating point of the selected pump

The actual operating point is slightly higher than the design point and very close to the best efficiency point (bep) of this pump. The following data for operating conditions can be drawn from the graph:

actual TDH = 19.5 m; actual discharge Q = 105 l/s; efficiency 87 %;
power requirement P = 23 kW

Note that this operating point might shift due to:

- water level changes of both suction side (river, well) and delivery side (pond or storage basin);
- deteriorating conditions of the pipeline (deposits, corrosion) which increases roughness coefficients and hence losses.

It is therefore common practice not to indicate a single operating point but a <u>range of operating</u> conditions which can be obtained by plotting several resistance curves (new - corroded pipeline system, high - low water levels).

The manufacturer's performance curve also indicates the **NPSH required** for the pump. For the operating point with Q = 105 l/s and the impeller diameter of 276 mm, an NPSH value of 5.0 m is required. In Appendix A it was shown that

$$NPSH_{available} = \frac{p_{atm}}{\rho\ g} - H_{st,suction} - H_{loss,suction} - \frac{v_s^2}{2g} - \frac{p_v}{\rho\ g}$$

(required information and graphs see Appendix A)

p_{atm} = 0.97 bar = 97'000 N/m2 for an altitude of 360 m a.s.l. (see p. A12, Appendix A)

$H_{st,suction}$ = 2.0 m

$H_{loss,suction} = 0.82m*Q_2^2/Q_n^2 = 0.82m*105^2/100^2 = 0.90$ m

with the actual suction branch NB 200 mm of the selected pump $v_{s2} = Q_2/A = 0.105 / (0.2^2*pi/4) =$ 3.34 m/s

p_v = vapour pressure of water at 20° C = 2338 N/m2

ρ = density of water at 20°C = 998.2 kg/m3

$NPSH_{avail.}$ = 6.20 m; $NPSH_{req.}$ = 5.0 m; **safety margin = 1.2 m** (a minimum safety margin of 1 m should always be aimed at).

4.2 Selection of a Turbine/Pump Used as a Turbine

For the micro-hydropower installation presented in Appendix A, section 6.2, a turbine or PAT is to be selected. The design conditions are as follows:

Q_n = 100 l/s, H_n = 12.6 m

This example is presented in the main text of the manual on PATs, Chapter 7.

APPENDIX C:

PUMP DESIGNS AND CHARACTERISTICS

1. Typical Impeller Profiles and their Characteristics

Figure C1 presents the profiles of single- and double-entry impellers, approximate values of specific speed and the characteristic curves Head H, Power P and efficiency η in function of the flow Q. This generalized table enables the designer of hydraulic plants to immediately classify a pump simply by looking at the shape of its impeller or its performance curve and to estimate the suitability and behaviour of this pump for a specific application.

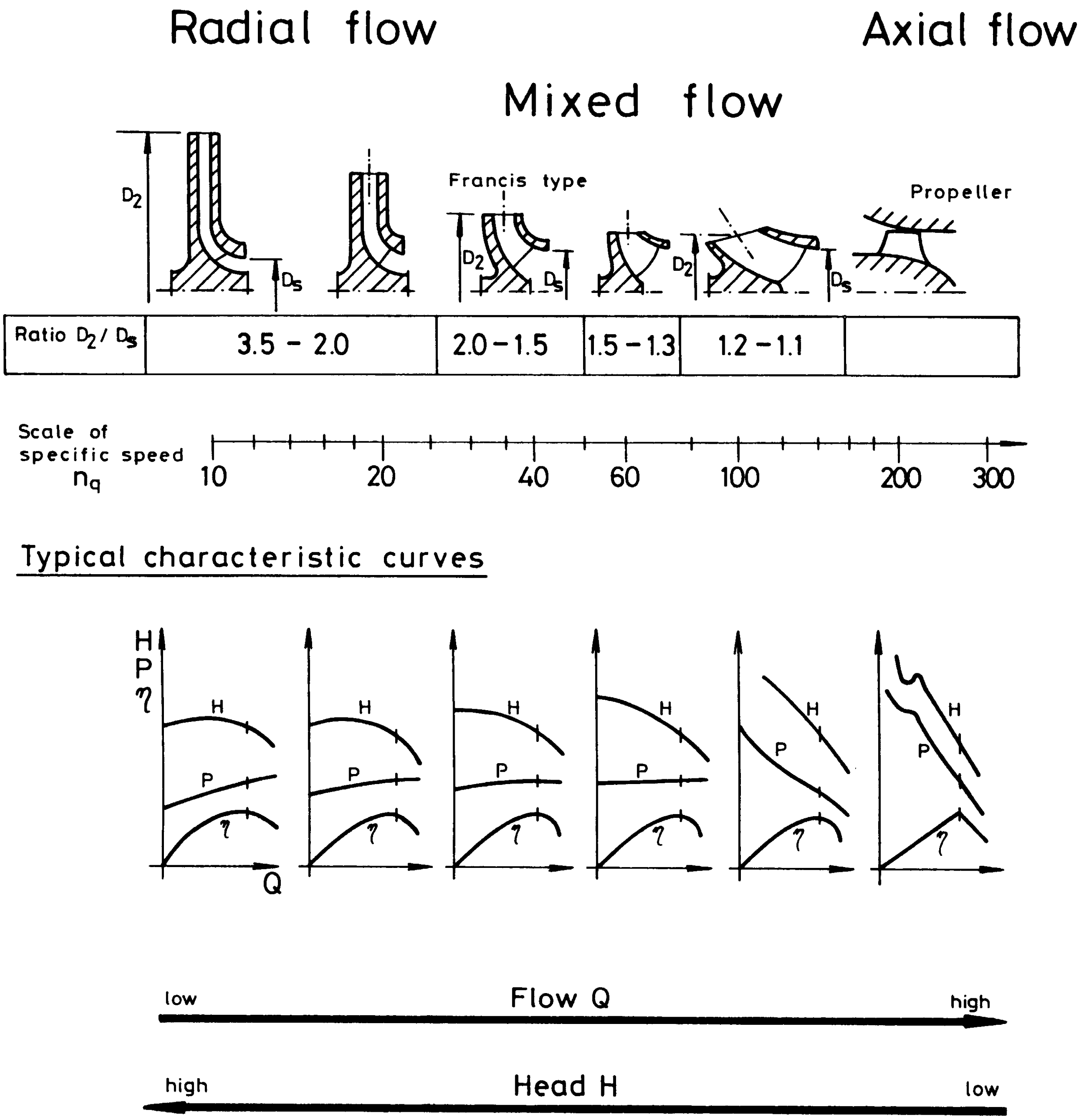


FIGURE C 1 :
Typical impeller profiles of pumps and their characteristics

2. General Application Range of Pumps

Figure C2 provides a general range of application of pumps in function of head, flow and hydraulic power. The indicated range of mixed and axial flow pumps covers single-stage units only; multistage mixed and axial flow pumps are available widening the range of these pump types. Figure C2 is therefore not suggested as an absolute criterion as there is a great deal of overlapping between the different designs.

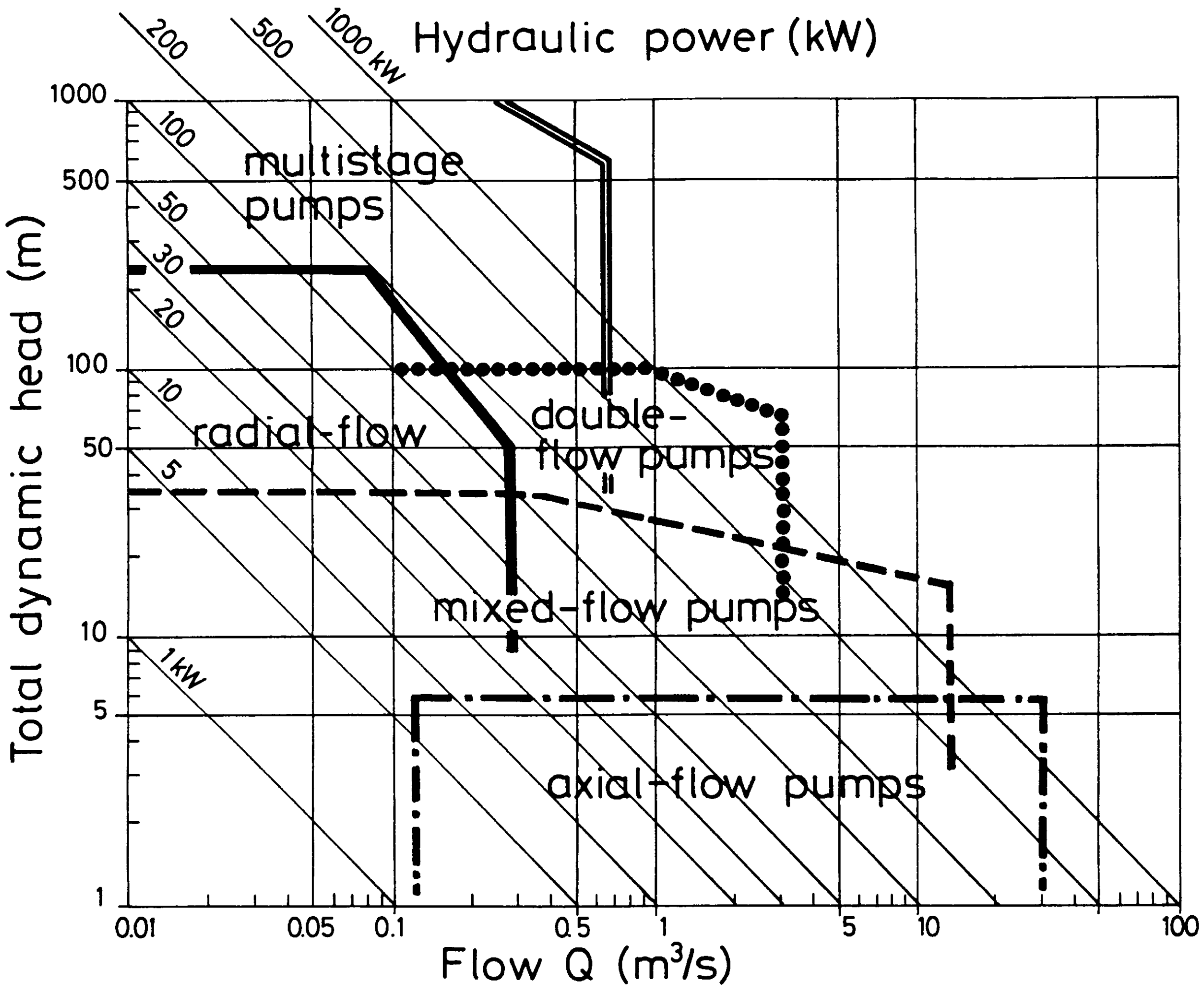


FIGURE C2 :
General range of application of various pump types

3. Typical Pump Designs

A large variety of different pump types are available for pumping water. Figure C3 shows three basic pump designs (rotodynamic pumps only): the single-stage radial-flow pump with horizontal shaft and the vertical shaft arrangements with mixed-flow and axial-flow impellers.

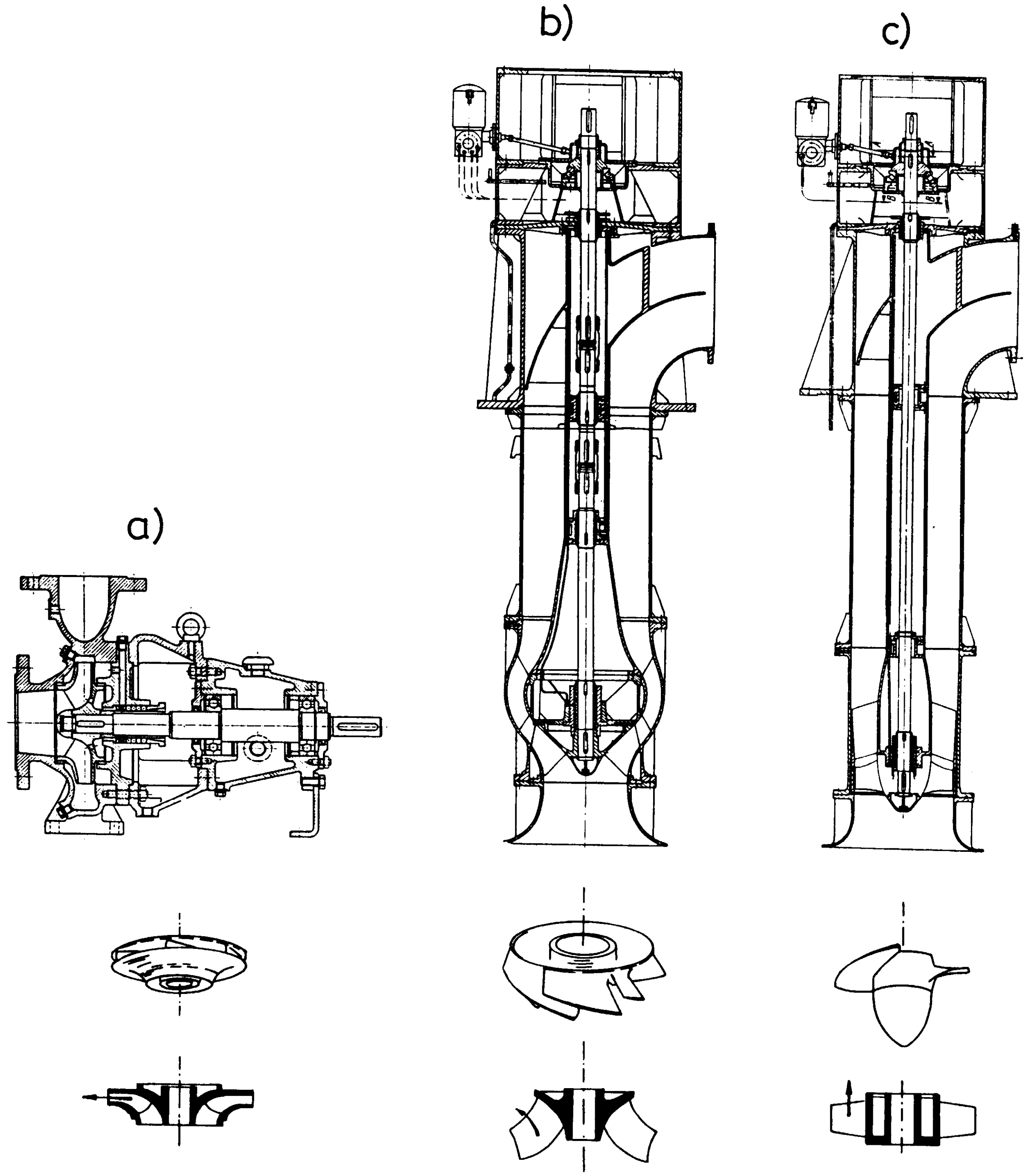


FIGURE C3 :
Examples of basic pump designs:
a) single-stage radial-flow pump
b) vertical mixed-flow pump
c) vertical axial-flow pump

4. Multistage and Multiflow Pumps

To achieve high heads, specific speed must decrease and impeller passages get progressively longer and narrow. As a result, the hydraulic resistance in the passages and the disc friction of the rotating impeller in the liquid absorb a considerable amount of input power and lower the efficiency of the pump. Below a specific speed of ≈ 10 , pumps cannot operate economically. To achieve higher heads, multistage pumps are therefore used, which distribute the total head uniformly between several stages (series connection of impellers). As characteristic value of the pump, the specific speed of a single impeller may be used:

$$\boxed{n_{q\ imp} = n \frac{\sqrt{Q}}{(H/i_{st})^{3/4}}} \qquad \text{(C1)}$$

where i_{st} = number of stages

Multistage pumps are usually equipped with a diffuser between each stage, consisting of a ring of fixed guide vanes which deflect the flow at the outlet of one stage to a virtually whirl-free inflow for the next stage. Figure C 4 shows a multistage pump with guide vanes and a qualitative performance curve.

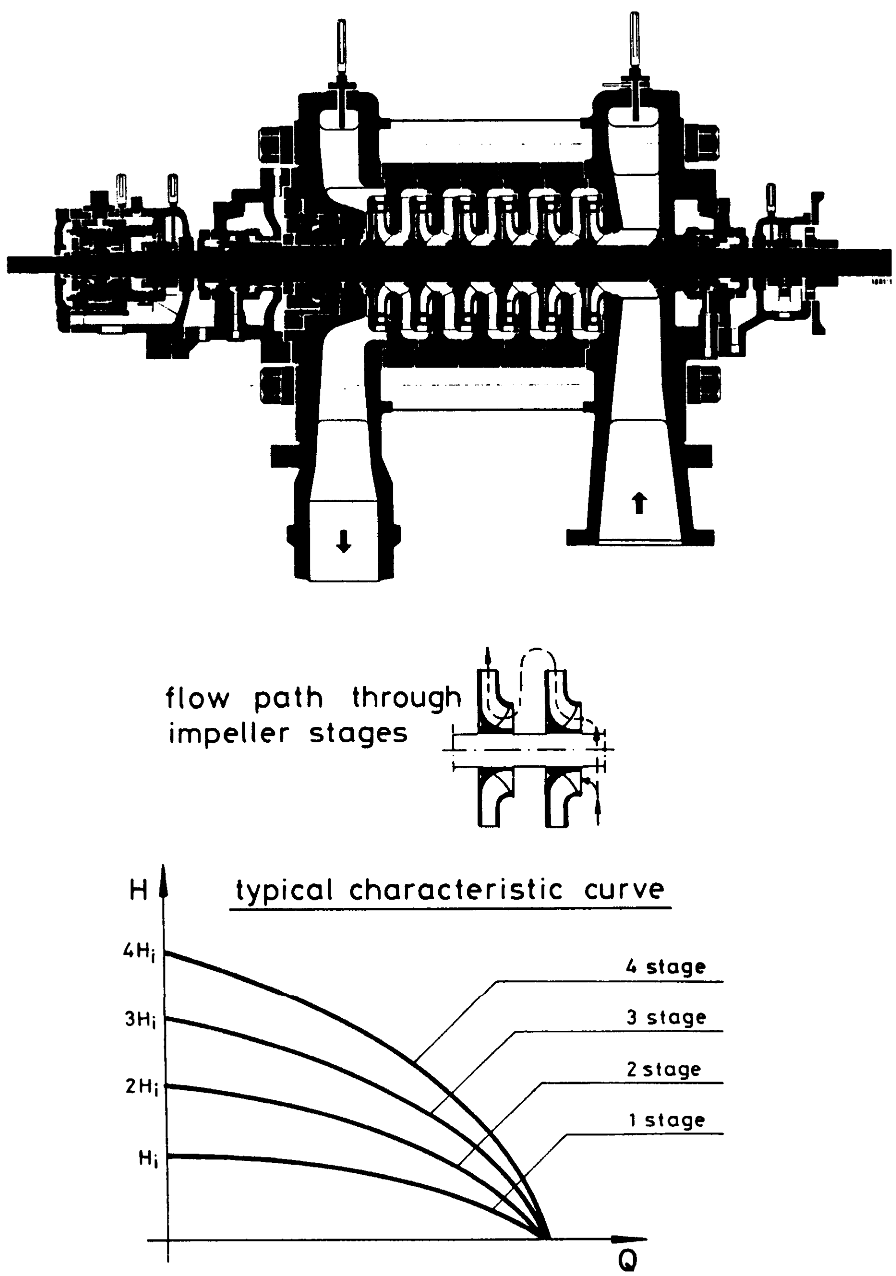


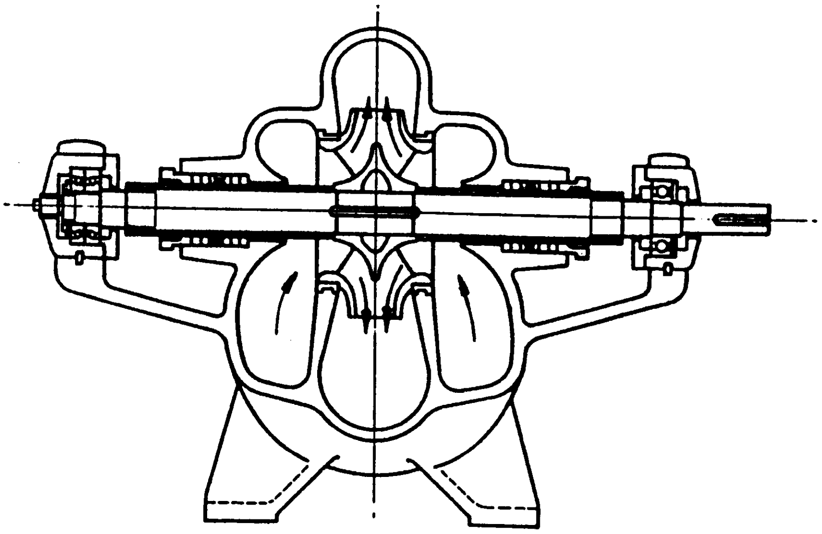
FIGURE C 4 :
Typical design and performance curve of a multistage pump

Multiflow pumps, usually double-flow pumps in practice, are designed to accommodate large flows at relatively high heads which would not be achieved by single flow or multistage pump designs (unfavourable shape of impeller with non-uniform velocity distribution). By splitting the flow into two entries (parallel arrangement of impellers), the inlet velocities at the impeller are considerably reduced which improves the suction behaviour of the pump (cavitation). The pair of impellers as shown in Figure C 5 below are arranged back-to-back on the pump shaft to compensate the axial thrust on the bearings.
Specific speed of a multiflow pump is usually expressed for a single impeller:

$$\boxed{n_{q\ imp} = n\frac{\sqrt{Q/i_{fl}}}{H^{3/4}}} \qquad \text{(C2)}$$

where i_{fl} = number of entries, usually 2

The double-flow pump (as shown in Figure C5) is often used as PAT due to its various advantages (favourable suction behaviour; split at shaft centre line which provides easy maintenance, i.e., all main parts can be replaced by simply removing the top cover)



typical performance curve

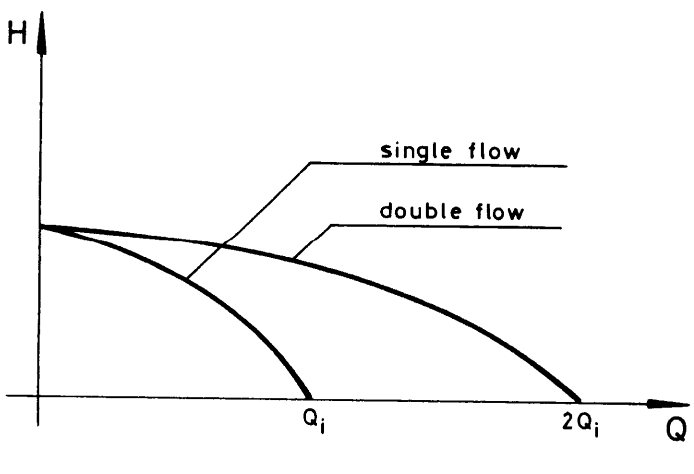


FIGURE C 5 :
Typical double-flow volute casing pump and qualitative performance curve

5. Open or Closed Impeller Types

Pumps are sometimes offered with open or closed impellers. Closed impellers have shrouds on both sides of the vanes, while open (or semi-open) impellers have only one shroud opposite the impeller entrance (see Figure C 6).
It has been found that closed impeller type pumps with specific speeds up to nq = 70 may theoretically yield higher efficiencies and most manufacturers offer these pumps with closed impellers (except for pumps designed for pumping liquids with high solid contents, non-clogging type). Closed or open impeller designs are usually available for mixed flow pumps.

Advantages of the open impeller design:
- possibility of machining the impeller surfaces which will increase efficiency of cast iron impellers
- sand locking does not occur as easily as with the closed type
- wear on the bowl and impeller vane edge may be compensated by adjustment to bring clearance and thus efficiency back to normal

Advantages of the closed impeller design:
- axial thrust is reduced which may increase bearing life or allow a more economical shafting
- wear on the impeller vanes is not as critical for the adjustment of impeller/bowl clearance as with open impellers .

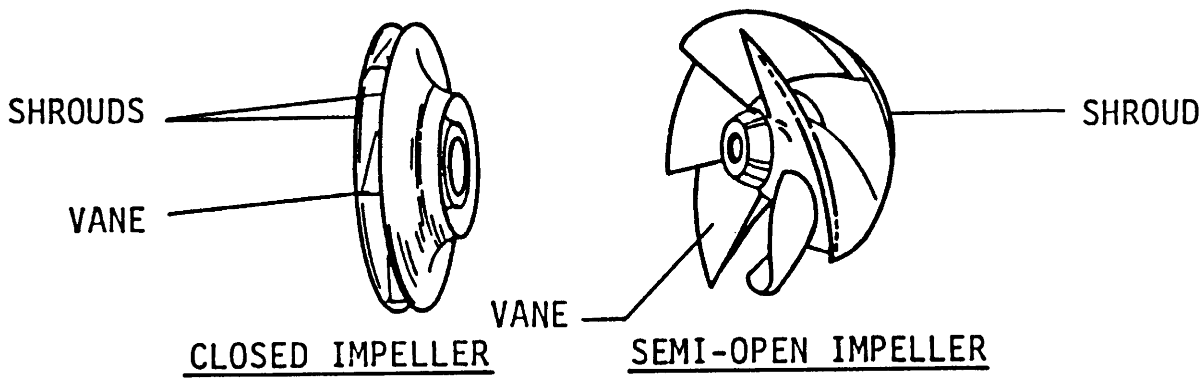


FIGURE C6 :
Examples of impeller types

APPENDIX D:

WORKING DIAGRAMS

The following diagrams are based on test results from over 80 PATs. The diagrams are intended to be used for the selection procedure of PATs as presented in the main text. A summary of the selection is shown in Section 3.8. A worked example on how to use the diagrams is presented in Chapter 7.

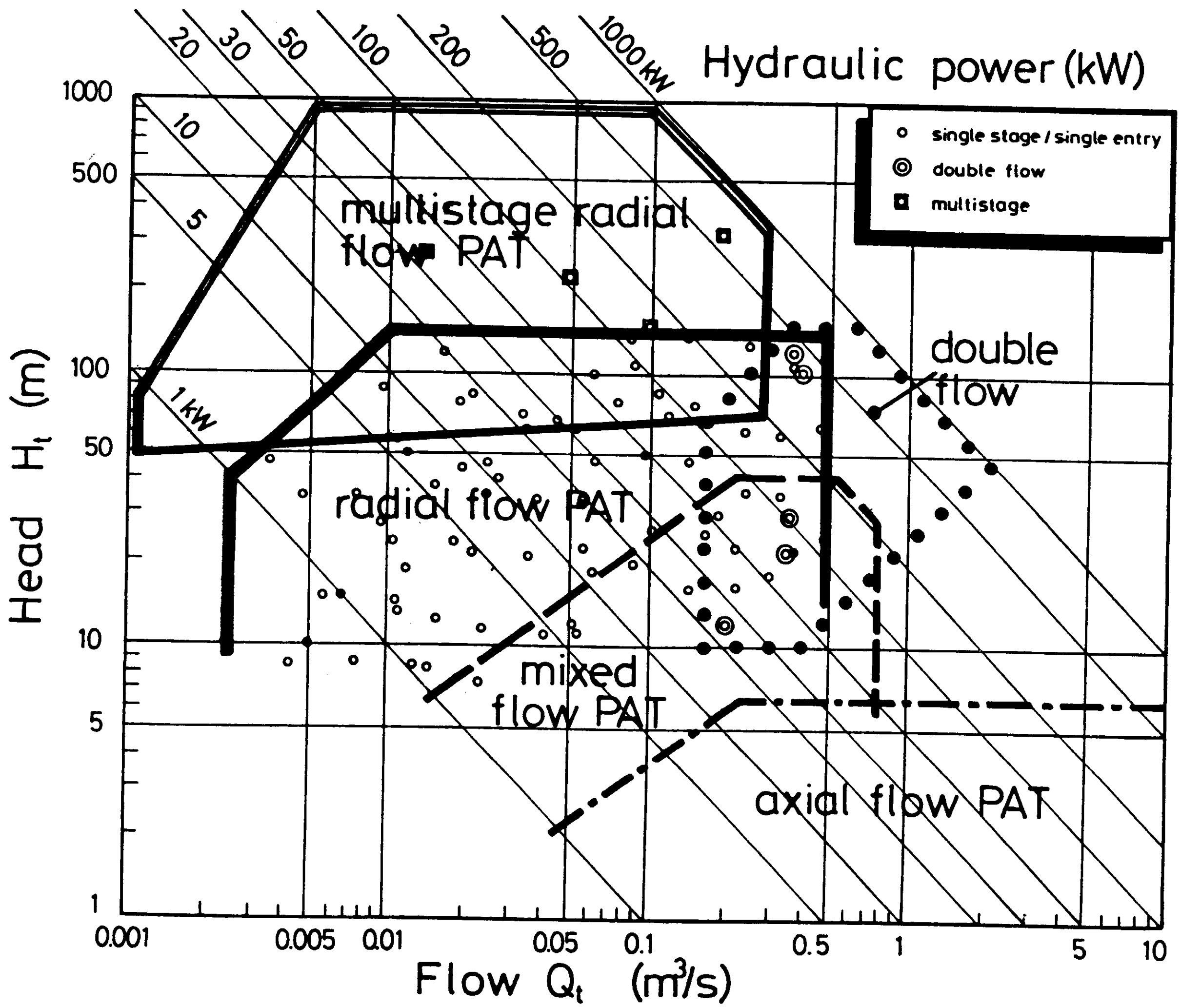
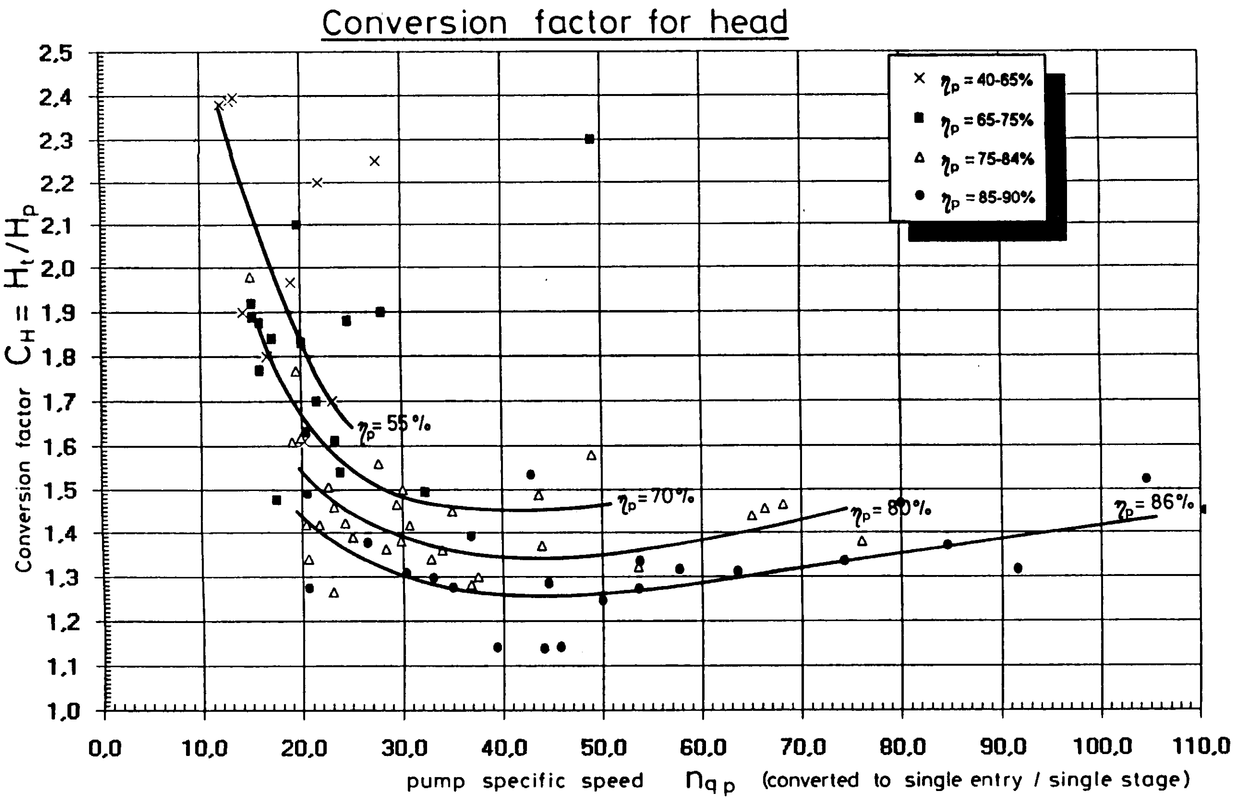


FIGURE D1 :
Range of Application of PATs



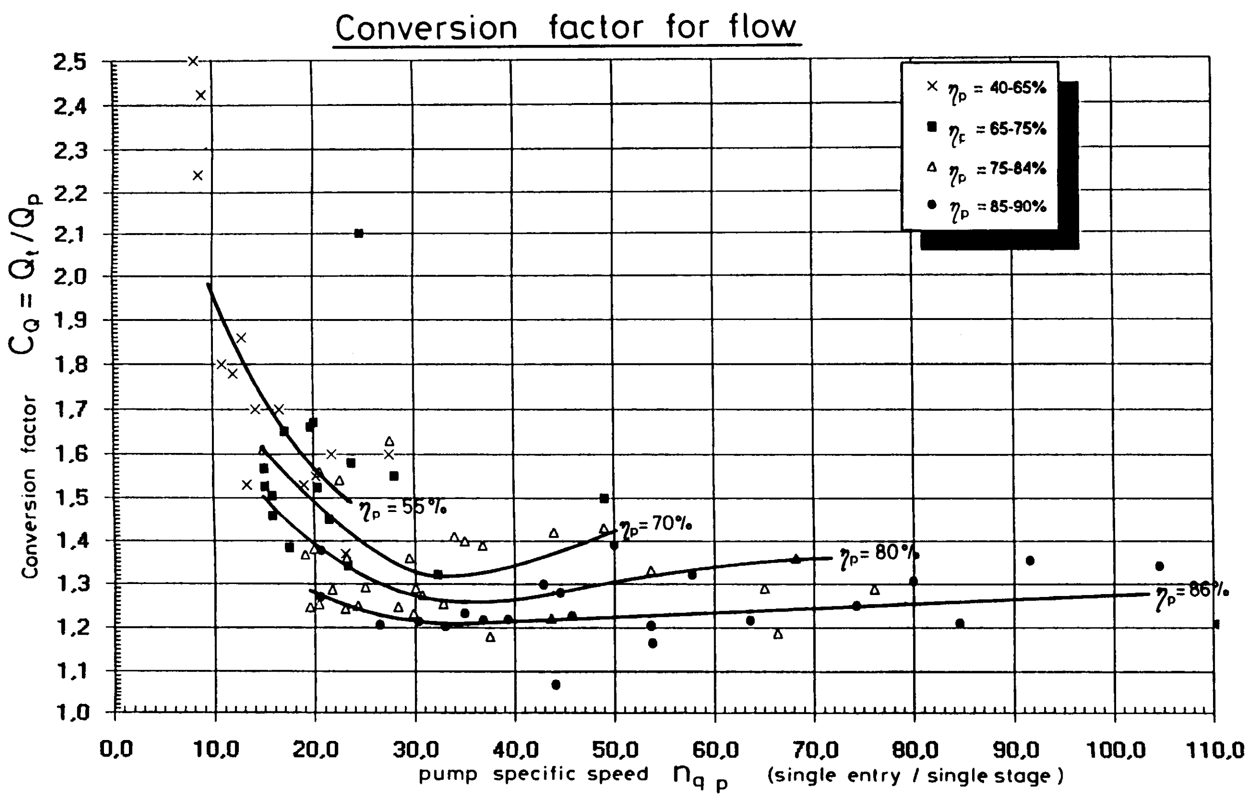


FIGURE D2 :
Factors for the conversion of pump design conditions into turbine design conditions

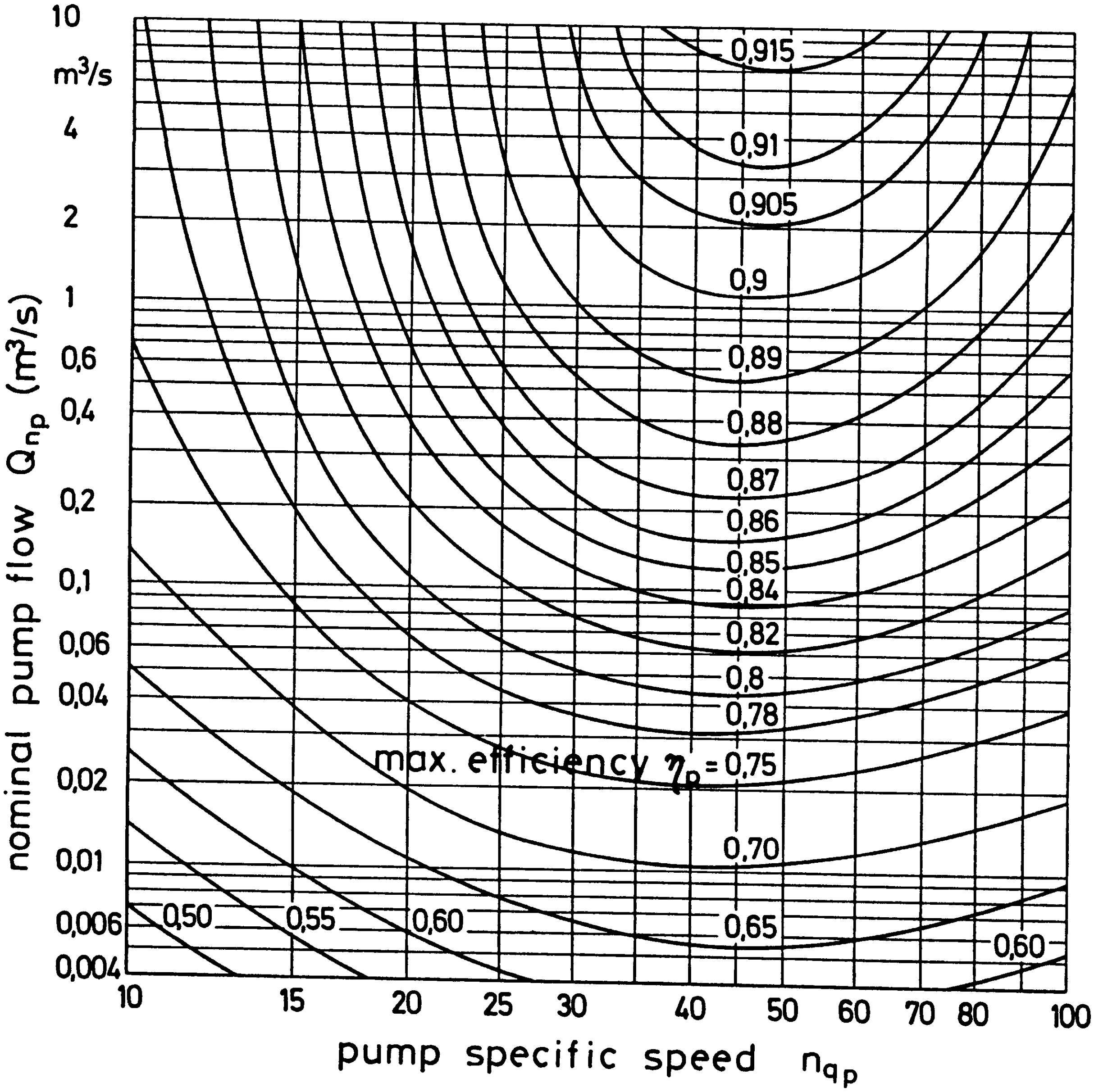


FIGURE D3 :
Maximum pump efficiency in function of specific speed and flow

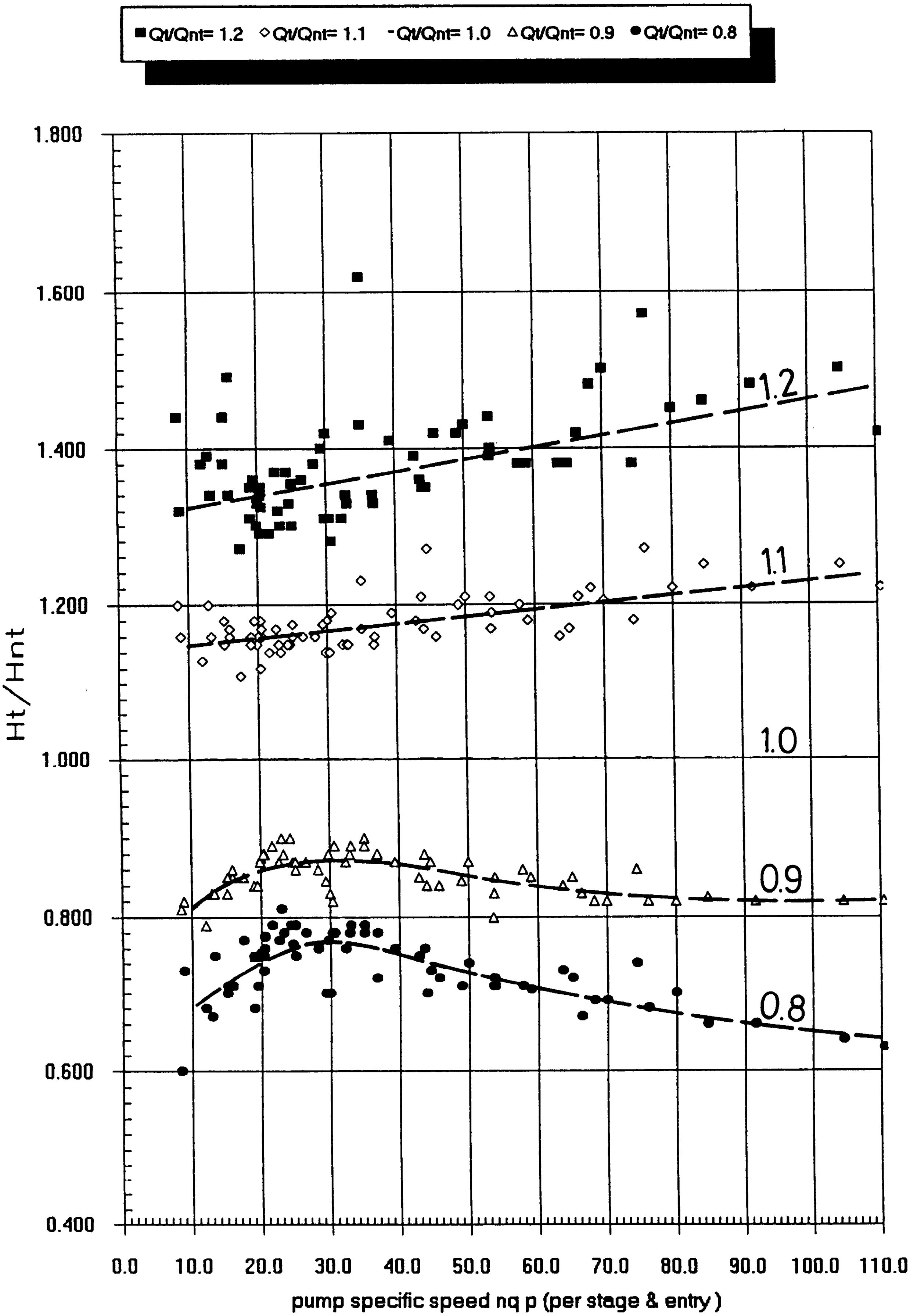


FIGURE D4 :
Turbine mode performance away from the best efficiency point: head versus flow

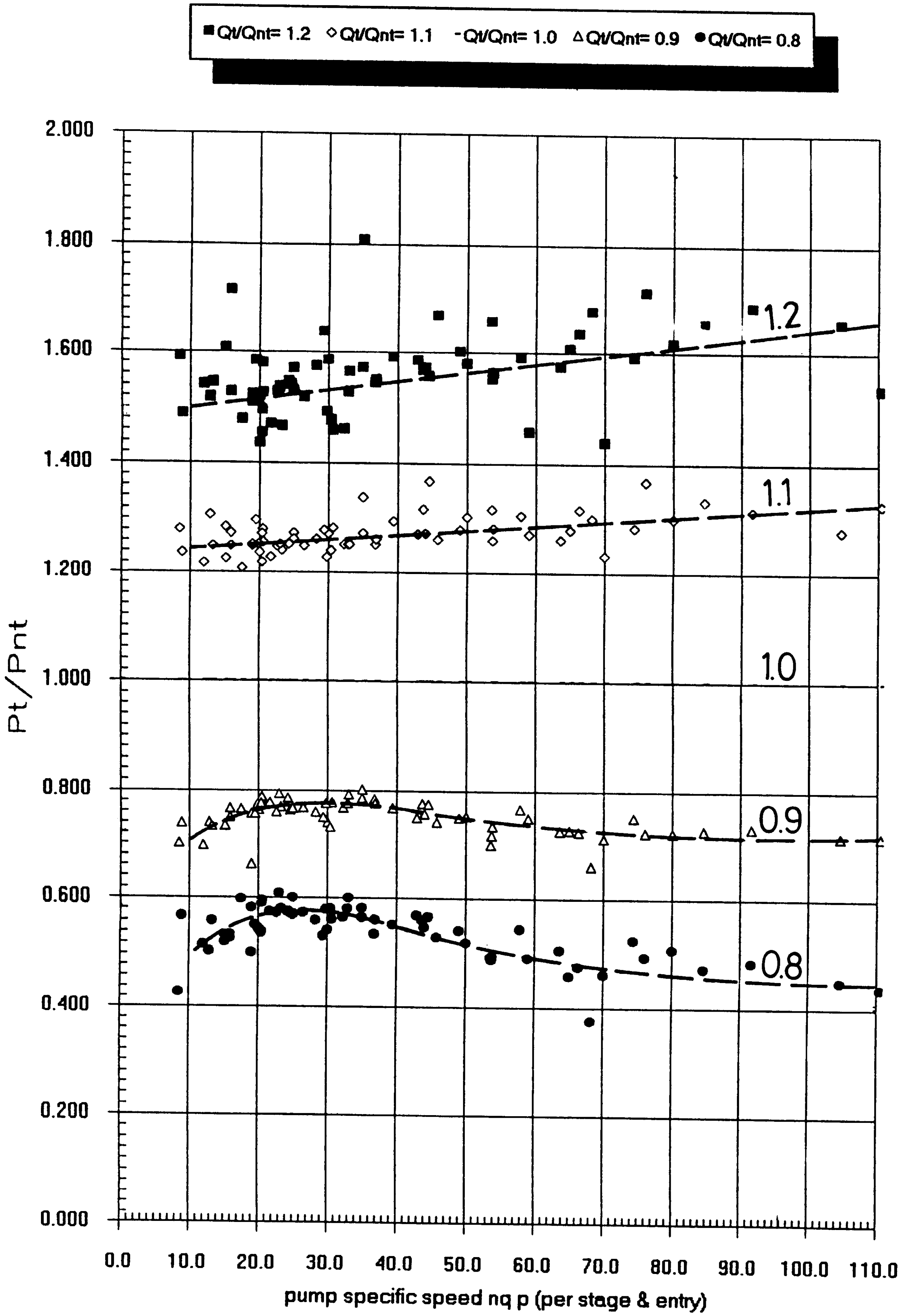


FIGURE D5 :
Turbine mode performance away from the best efficiency point: power versus flow

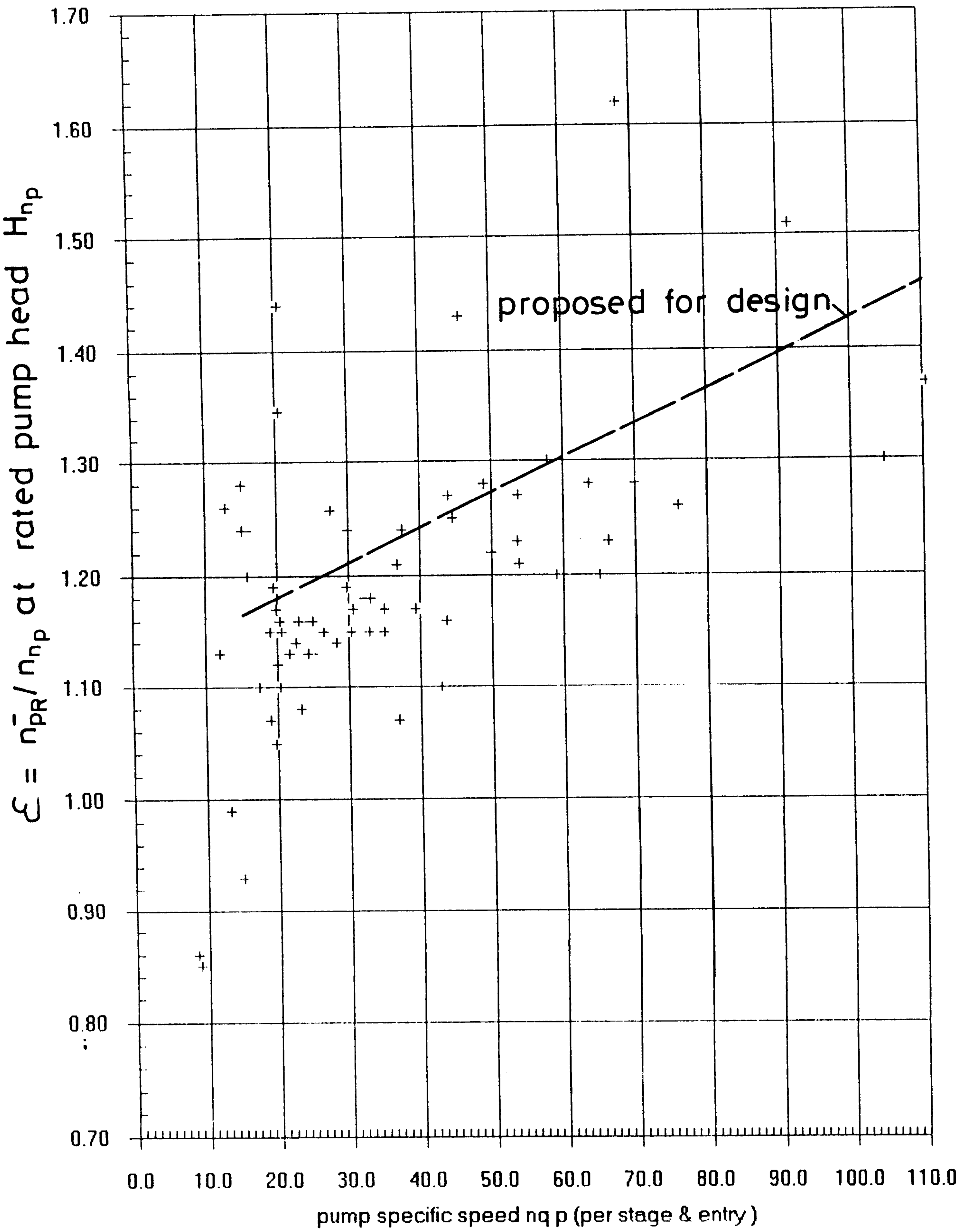


FIGURE D6 :
Runaway speed at rated pump head in function of specific speed

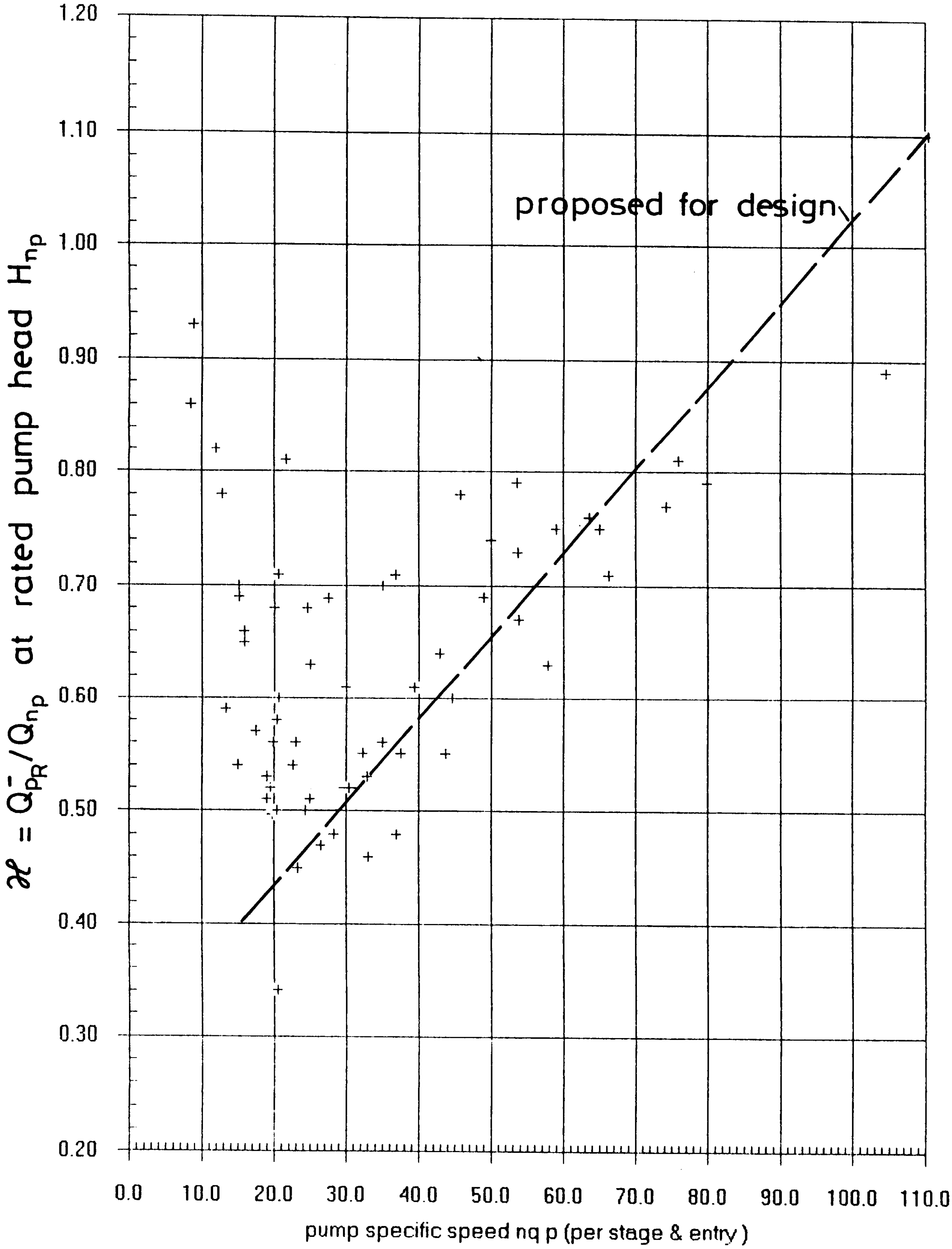


FIGURE D7 :
Flow at runaway conditions (for rated pump head) in function of specific speed

APPENDIX E:

VALVE CHARACTERISTICS

1. General

The control valve in an MHP may induce pressure surges into the penstock if closed or opened too rapidly. In order to check the permissible valve closure time, the characteristic curve flow versus valve opening must be known. The following tables provide the loss coefficients for a number of gates frequently used in MHPs. As mentioned in the main text, the system into which a particular valve is introduced must be taken into account when establishing the characteristic curve. This means for an MHP using a PAT that system resistance curves for various valve openings must be calculated in order to determine the corresponding flow. This procedure is shown schematically in Figure E1 for a PAT being shut down from steady state runaway conditions.

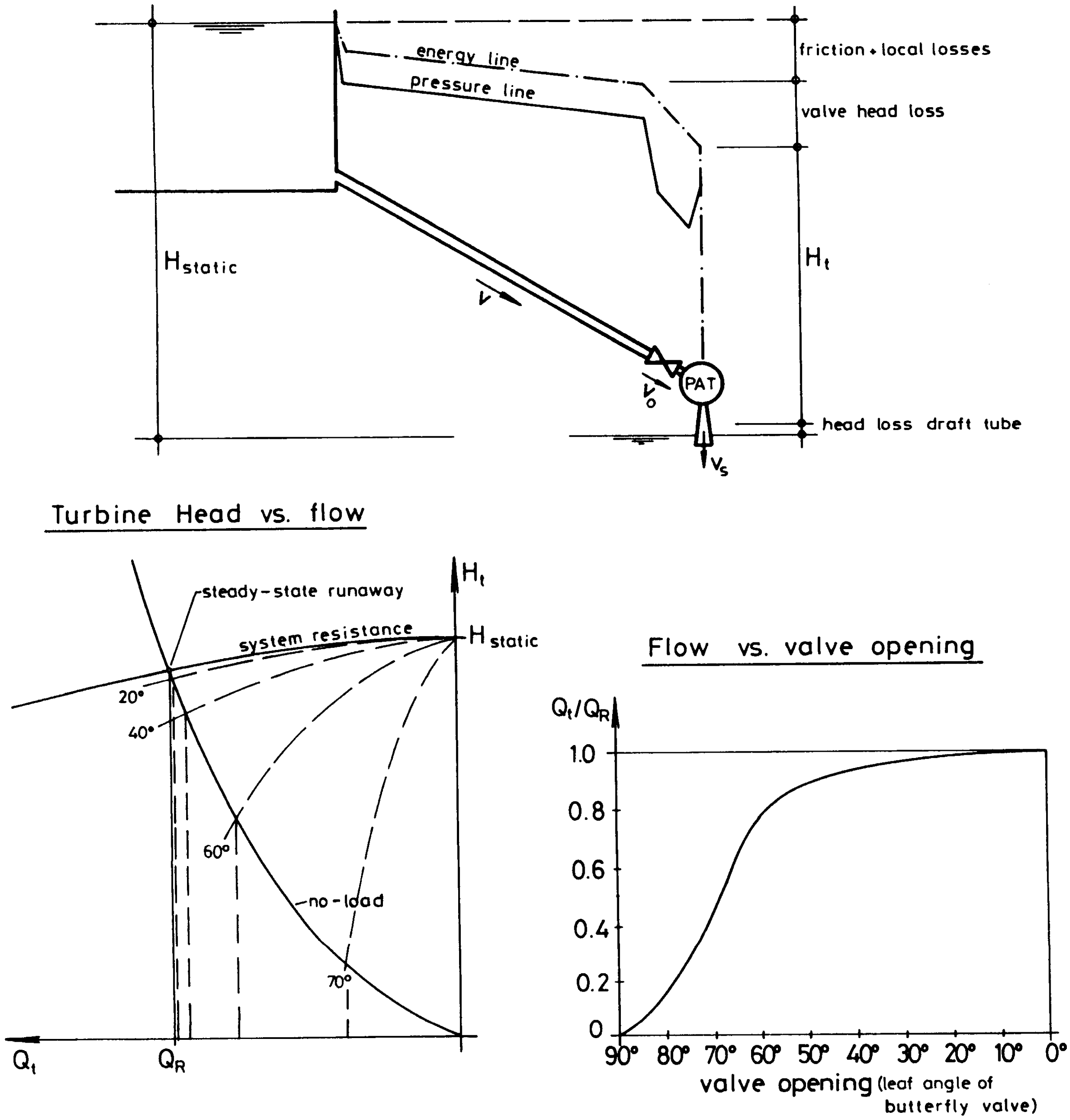
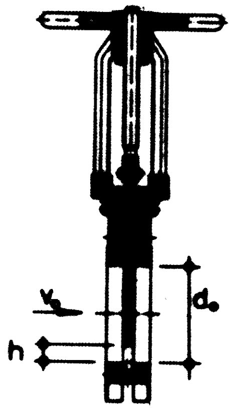


FIGURE E1 :
Establishing the valve characteristics in an MHP using a PAT (example shown for shut-down from steady-state runaway)

2. Simple Gate Valve (Single Disc)



Head loss ΔH across the valve (in m water column):

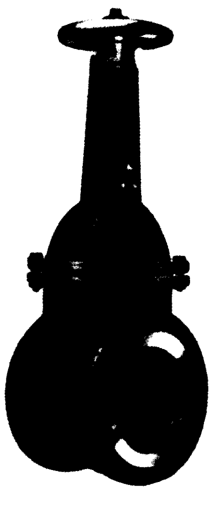
$$\boxed{\Delta H = \zeta \frac{v_o^2}{2g}} \qquad \text{(E1)}$$

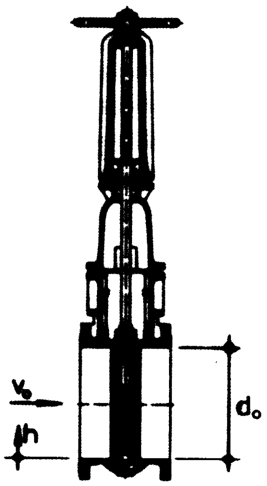
Loss coefficients ζ for varying gate opening h [1)]

For circular pipes:

h/d_o	0 closed	0.125	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0 open
ζ	∞	97.8	35.0	10.0	4.6	2.06	0.98	0.44	0.17	0.06	0

3. Wedge Valve (Double Disc)



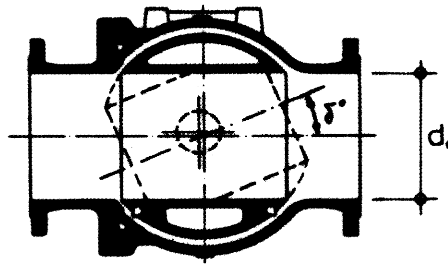


Head loss ΔH according to Formula E1

Loss coefficient ζ [1)]

h/d_o	0.25	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
ζ	30.0	22.0	12.0	5.3	2.8	1.5	0.8	0.3	0.15

4. Ball Valve



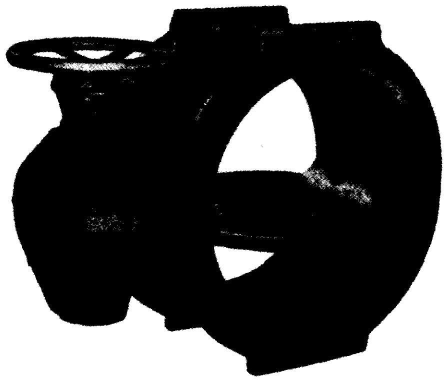
Head loss ΔH from Formula E1 above

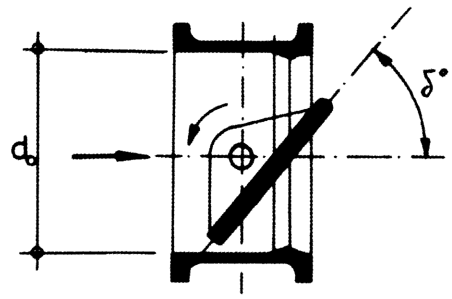
Loss coefficients ζ [1)]

for circular pipes

δ °	5	10	15	20	25	30	35	40	45	50	55	67
ζ	0.05	0.31	0.88	1.84	3.45	6.15	11.2	20.7	41.0	95.3	275	∞

5. Butterfly Valve





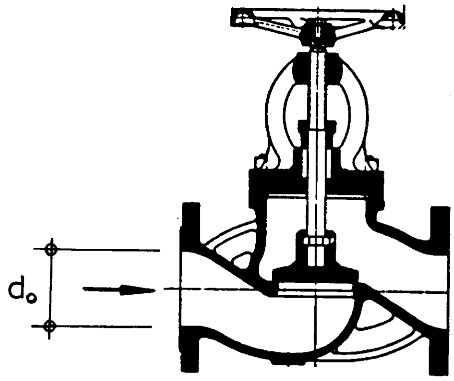
Head loss ΔH according to Formula E1

Loss coefficients [1)]

for circular pipes

δ °	5	10	15	20	25	30	40	50	60	65	70	90
ζ	0.24	0.52	0.90	1.54	2.51	3.91	10.8	32.6	118	256	751	∞

6. Globe Valve



Head loss ΔH according to Formula E1

Loss coefficients ζ (for nominal bore d_o = 250 mm) [2]

Opening %	100	80	60	40	20	0
ζ	5.2	6.9	13.7	82.6	2500	∞

1) Source :
I.E. Idel'cik, Memento des pertes de charge, Editions Eyrolles, Paris, 1986

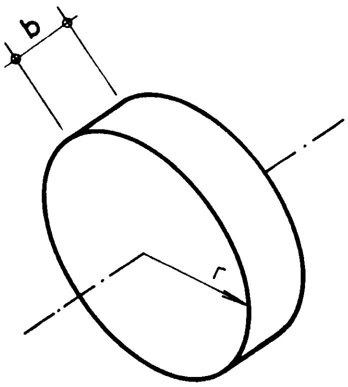
2) Source :
Adapted from *Wylie, Streeter*, Fluid Transients, FEB Press, USA, 1988

APPENDIX F:

INERTIA OF ROTATING ELEMENTS OF PATS AND MACHINERY

When analyzing transient conditions of an MHP (such as runaway speed), the inertia of the rotating elements against acceleration must be known. It is usually expressed by the mass moment of inertia J in kgm^2. This value may be obtained from manufacturers of turbines, generators, flywheels and machines. If it is not available, it may be calculated (approximately) using the following formulae:

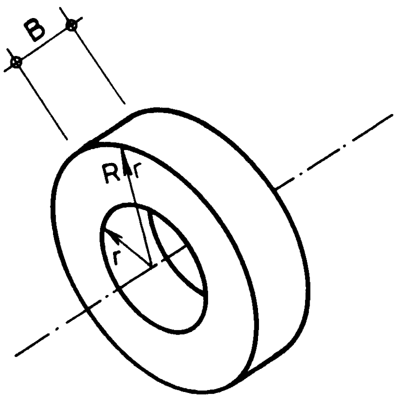
for disks and solid shafts:



$$\boxed{J_{disk} = \frac{1}{2} m_{disk} r^2 \quad (\text{in } kgm^2)} \tag{F1}$$

where m = mass of the disk $m_{disk} = \rho \pi r^2 b$

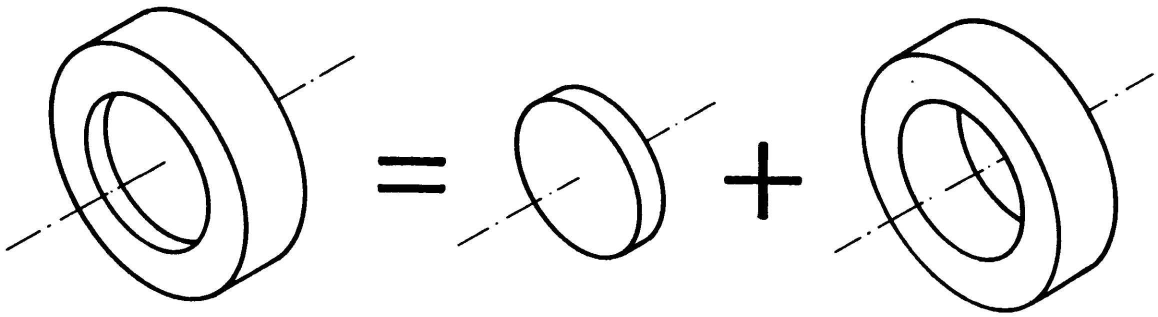
for a hollow cylinder:



$$\boxed{J_{cyl} = \frac{1}{2} m_{cyl} (R^2 - r^2)} \tag{F2}$$

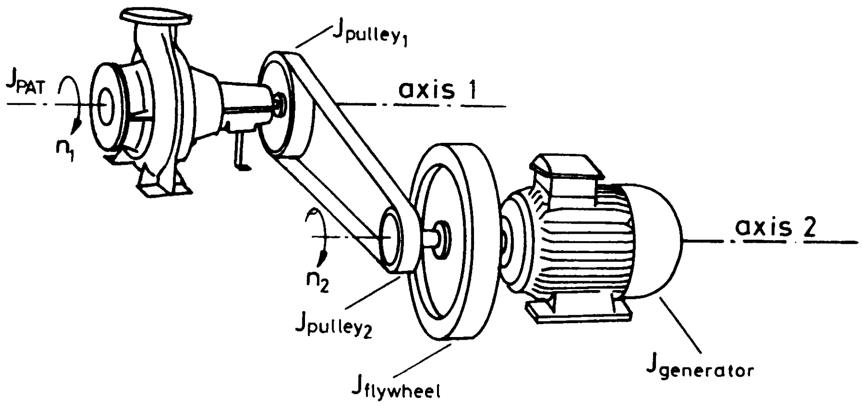
where m = mass of the hollow cylinder $m_{cyl} = \rho \pi B (R^2 - r^2)$

for combined elements consisting of both disks and hollow cylinders (e.g. flywheels), the mass moments of inertia of the single elements can added:



$$\boxed{J_{total} = J_{disk} + J_{cyl}} \qquad \text{(F3)}$$

If the rotating parts of a generating unit (turbine/PAT and generator or machinery) rotate on different shafts and at different speeds, the inertia of the individual elements must be transformed to a reference axis (usually the generator axis):



$$J_{axis_1} = J_{PAT} + J_{pulley_1}$$

$$J_{axis_2} = J_{generator} + J_{flywheel} + J_{pulley_2}$$

$$\boxed{J_{total} = J_{axis_2} + J_{axis_1} \left(\frac{n_1}{n_2}\right)^2}$$ referred to generator shaft (axis 2) (E4)

Note that for maximum acceleration of the PAT and the corresponding (worst case) pressure surge induced into the penstock, the inertias of PAT and $pulley_1$ alone must be used (belt may fail or slip off).

The inertia of generators or machines is sometimes expressed by the **moment of gyration GD^2** (kgm^2) which is the product of the mass (in kg) of the rotating elements and the square of the diameter of gyration D^2. D is the diameter where the mass may theoretically be concentrated to have the same inertia effect as the real rotating element. The relation to the mass moment of inertia J is as follows:

$$\boxed{GD^2 = 4\ J} \qquad \text{(F5)}$$

Inertias of PATs are usually very small; for PATs direct-coupled to generators, J_{PAT} is seldom more than 20 % of $J_{generator}$ and may sometimes even be neglected. In the absence of values of inertia of different types of generators, the following rough indications may be used for pre-feasibility studies:

Inertia J (kgm^2) for different types of generators

Four-pole machines (synchronous speed 1500 rpm)

Power output	Synchronous Generator	Induction Generator	Induction motor as generator (IMAG)
4 kW	0.05		0.01
10 kW	0.09	0.05	0.035
20 kW	0.28	0.17	0.10
30 kW	0.40	0.23	0.19

APPENDIX G:

GENERATOR EFFICIENCIES AND SLIP

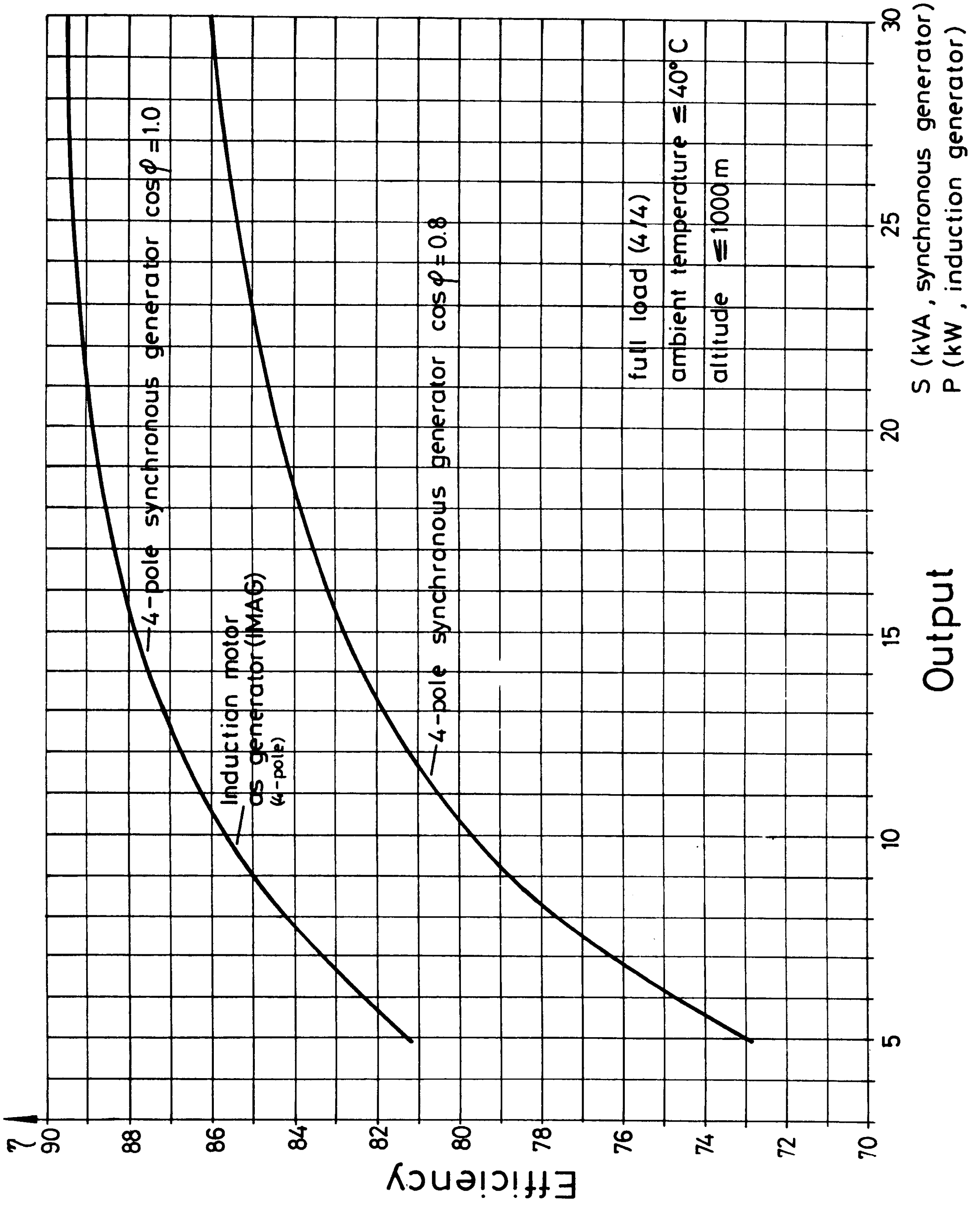


FIGURE G1 :
General range of efficiencies for generators

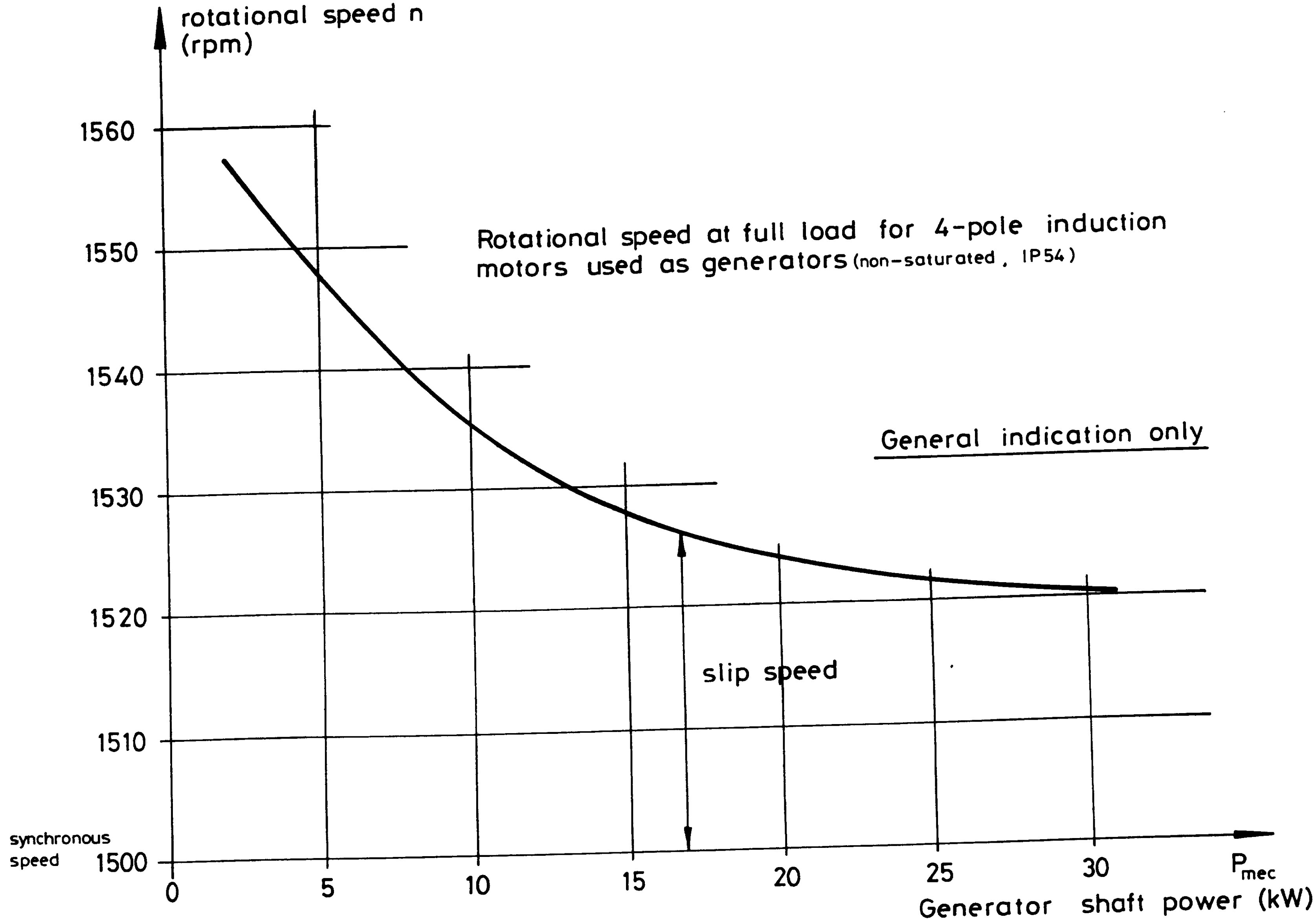


FIGURE G2 :
Rotational speed including slip for induction motors used as generators

APPENDIX H:

DISCHARGE MEASUREMENT STRUCTURES

1. General

The most common discharge measuring structures for micro-hydropower installations are sharp-crested weirs. They are among the most accurate discharge measuring devices for field application. A disadvantage is the relatively large head loss incurred.

2. V-Notch Sharp-Crested Weir

The V-notch weir is suitable for measurements of a wide range of flows with high accuracies (1 - 120 l/s). It is often referred to as the Thomson Weir.
It should be placed perpendicular to the sides and the bottom of a straight channel section (e.g. the tailrace channel). Best results are obtained if the weir is placed in a rectangular approach channel whose bed and sides are sufficiently remote from the edges of the V-notch so that a fully contracted flow over the weir may develop. The limits of application are the following (symbols and notations see Figure H1):

Fully contracted Thomson Weir
$h_1 / p <= 0.4$
$h_1 / B <= 0.2$
$0.05\ m < h_1 <= 0.38\ m$
$p >= 0.45\ m$
$B_1 >= 0.90\ m$

Note that the tailwater level should remain at least 0.05 m below crest level.
Table H2 gives the discharge in function of head for a Thomson Weir with a 90° V-notch.

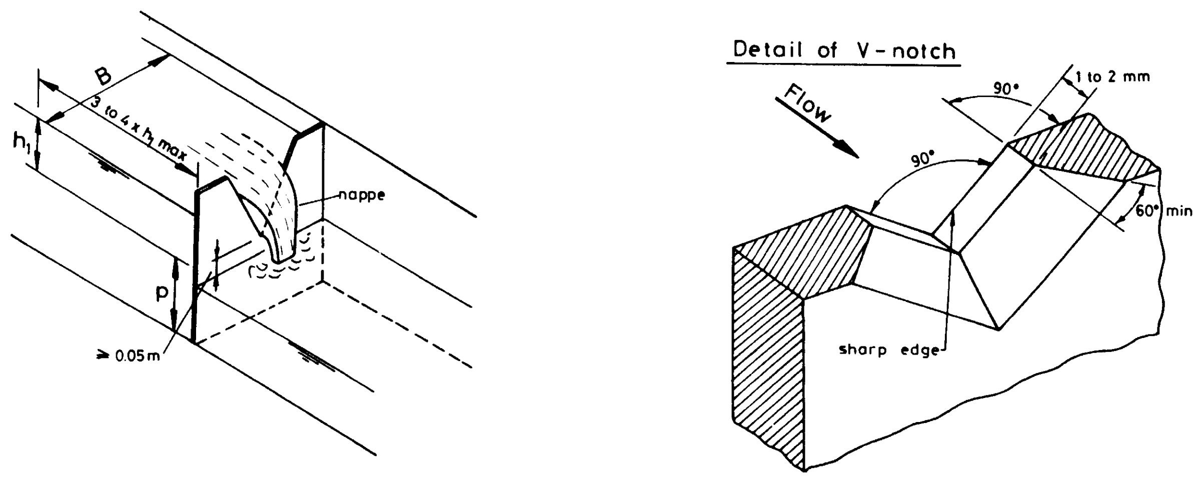


FIGURE H1 :
V-notch sharp-crested weir (Thomson Weir)

TABLE H2 :

Rating table for a 90° V-notch weir (fully contracted)
(adapted from: ILRI, Wageningen, 1978)

Head (m)	Discharge (l/s)	Head	Discharge (l/s)	Head (m)	Discharge (l/s)	Head (m)	Discharge (l/s)	Head (m)	Discharge (l/s)
0.050	0.803	0.118	6.653	0.186	20.621	0.254	44.907	0.322	81.314
0.052	0.884	0.120	6.935	0.188	21.180	0.256	45.796	0.324	82.583
0.054	0.970	0.122	7.224	0.190	21.748	0.258	46.696	0.326	83.863
0.056	1.061	0.124	7.522	0.192	22.322	0.260	47.606	0.328	85.155
0.058	1.156	0.126	7.827	0.194	22.906	0.262	48.527	0.330	86.459
0.060	1.257	0.128	8.139	0.196	23.501	0.264	49.458	0.332	87.775
0.062	1.362	0.130	8.458	0.198	24.106	0.266	50.400	0.334	89.103
0.064	1.473	0.132	8.785	0.200	24.719	0.268	51.353	0.336	90.448
0.066	1.588	0.134	9.119	0.202	25.339	0.270	52.317	0.338	91.811
0.068	1.710	0.136	9.461	0.204	25.969	0.272	53.291	0.340	93.175
0.070	1.836	0.138	9.810	0.206	26.610	0.274	54.276	0.342	94.551
0.072	1.967	0.140	10.167	0.208	27.261	0.276	55.272	0.344	95.940
0.074	2.105	0.142	10.532	0.210	27.921	0.278	56.282	0.346	97.340
0.076	2.248	0.144	10.904	0.212	28.588	0.280	57.306	0.348	98.753
0.078	2.397	0.146	11.284	0.214	29.264	0.282	58.335	0.350	100.19
0.080	2.551	0.148	11.671	0.216	29.953	0.284	59.375	0.352	101.63
0.082	2.710	0.150	12.066	0.218	30.651	0.286	60.425	0.354	103.08
0.084	2.876	0.152	12.471	0.220	31.359	0.288	61.487	0.356	104.54
0.086	3.048	0.154	12.883	0.222	32.077	0.290	62.560	0.358	106.02
0.088	3.225	0.156	13.304	0.224	32.803	0.292	63.645	0.360	107.52
0.090	3.409	0.158	13.732	0.226	33.535	0.294	64.748	0.362	109.02
0.092	3.598	0.160	14.169	0.228	34.282	0.296	65.858	0.364	110.54
0.094	3.795	0.162	14.614	0.230	35.039	0.298	66.976	0.366	112.06
0.096	3.997	0.164	15.067	0.232	35.806	0.300	68.106	0.368	113.62
0.098	4.206	0.166	15.529	0.234	36.582	0.302	69.246	0.370	115.17
0.100	4.420	0.168	15.999	0.236	37.369	0.304	70.398	0.372	116.73
0.102	4.641	0.170	16.477	0.238	38.166	0.306	71.568	0.374	118.31
0.104	4.869	0.172	16.964	0.240	38.973	0.308	72.750	0.376	119.91
0.106	5.103	0.174	17.459	0.242	39.790	0.310	73.936	0.378	121.52
0.108	5.344	0.176	17.963	0.244	40.617	0.312	75.135	0.380	123.13
0.110	5.592	0.178	18.478	0.246	41.454	0.314	76.344		
0.112	5.847	0.180	19.001	0.248	42.302	0.316	77.566		
0.114	6.108	0.182	19.531	0.250	43.160	0.318	78.802		
0.116	6.377	0.184	20.071	0.252	44.028	0.320	80.057		

3. Rectangular Sharp-Crested Weir

The rectangular sharp-crested weir should be located preferably in a rectangular approach channel. It can be used for measuring larger flows which cannot be accommodated by the V-notch weir. The width of the weir is chosen in such a way that the head h_1 remains within the limits given below for all flows to be measured. The conditions and limitations of the fully-contracted rectangular weir are as follows:

Rectangular sharp-crested weir
$B - b >= 4\,h_1$
$h_1 / p <= 0.5$
$h_1 / b <= 0.5$
$0.07\ m <= h_1 < 0.60\ m$
$b >= 0.30\ m$
$p >= 0.30\ m$

Note that the tailwater level should remain at least 0.05 m below the weir crest (aeration of the nappe).

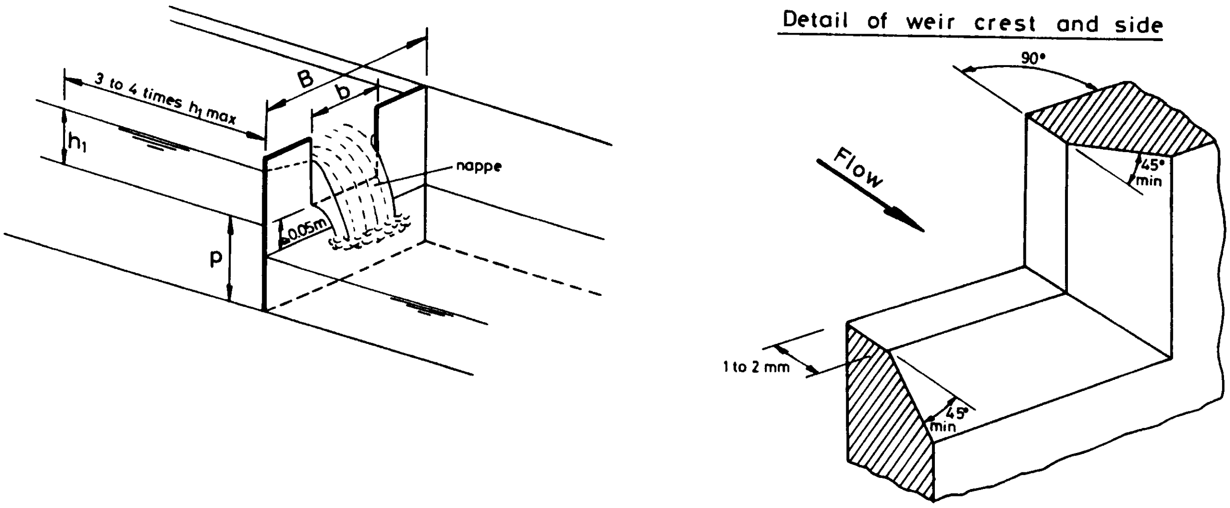


FIGURE H3 :
Rectangular sharp-crested weir

The discharge through the sharp-crested rectangular weir is determined by the following formula (according to Kindsvater and Carter, 1957):

$$\boxed{Q = C_e \frac{2}{3} \sqrt{2g}\ b_e\ h_e^{\ 1.5}} \qquad \text{(H1)}$$

where Q = discharge in m^3/s
C_e = effective discharge coefficient according to Table H4 below
b_e = effective crest width $b_e = b + k_b$ (with b = weir crest in m and k_b = correction factor according to Figure H5 below)

h_e = effective head; $h_e = h + 0.001$ m (with h = head in m)

TABLE H4 :
Discharge coefficient Ce in function of the ratio b/B

b / B	Discharge coefficient C_e
1.0	$0.602 + 0.075\ h_1 / p$
0.9	$0.599 + 0.064\ h_1 / p$
0.8	$0.597 + 0.045\ h_1 / p$
0.7	$0.595 + 0.030\ h_1 / p$
0.6	$0.593 + 0.018\ h_1 / p$
0.5	$0.592 + 0.011\ h_1 / p$
0.4	$0.591 + 0.0058\ h_1 / p$
0.3	$0.590 + 0.0020\ h_1 / p$
0.2	$0.589 + 0.0018\ h_1 / p$
0.1	$0.588 + 0.0021\ h_1 / p$
0.0	$0.587 + 0.0023\ h_1 / p$

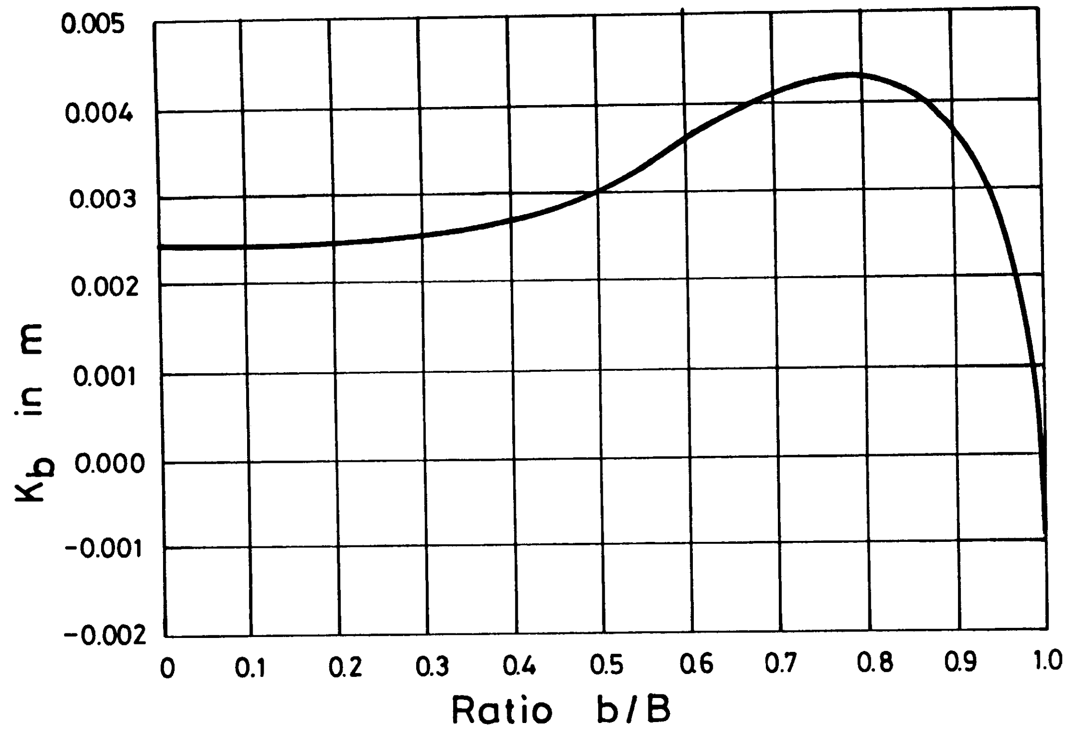


FIGURE H5 :
Correction factor for crest width K_b in function of b / B

APPENDIX I :

CAPITAL RECOVERY FACTORS

$$RF = \frac{i\,(\,i+1\,)^{n}}{(\,i+1\,)^{n}-1} \qquad \text{(I 1)}$$

where i = interest rate in % / 100
n = period in years

number of years n	interest rate i (%) 1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
1	1.010	1.020	1.030	1.040	1.050	1.060	1.070	1.080	1.090	1.100	1.110	1.120	1.130	1.140	1.150
2	0.508	0.515	0.523	0.530	0.538	0.545	0.553	0.561	0.568	0.576	0.584	0.592	0.599	0.607	0.615
3	0.340	0.347	0.354	0.360	0.367	0.374	0.381	0.388	0.395	0.402	0.409	0.416	0.424	0.431	0.438
4	0.256	0.263	0.269	0.275	0.282	0.289	0.295	0.302	0.309	0.315	0.322	0.329	0.336	0.343	0.350
5	0.206	0.212	0.218	0.225	0.231	0.237	0.244	0.250	0.257	0.264	0.271	0.277	0.284	0.291	0.298
6	0.173	0.179	0.185	0.191	0.197	0.203	0.210	0.216	0.223	0.230	0.236	0.243	0.250	0.257	0.264
7	0.149	0.155	0.161	0.167	0.173	0.179	0.186	0.192	0.199	0.205	0.212	0.219	0.226	0.233	0.240
8	0.131	0.137	0.142	0.149	0.155	0.161	0.167	0.174	0.181	0.187	0.194	0.201	0.208	0.216	0.223
9	0.117	0.123	0.128	0.134	0.141	0.147	0.153	0.160	0.167	0.174	0.181	0.188	0.195	0.202	0.210
10	0.106	0.111	0.117	0.123	0.130	0.136	0.142	0.149	0.156	0.163	0.170	0.177	0.184	0.192	0.199
11	0.096	0.102	0.108	0.114	0.120	0.127	0.133	0.140	0.147	0.154	0.161	0.168	0.176	0.183	0.191
12	0.089	0.095	0.100	0.107	0.113	0.119	0.126	0.133	0.140	0.147	0.154	0.161	0.169	0.177	0.184
13	0.082	0.088	0.094	0.100	0.106	0.113	0.120	0.127	0.134	0.141	0.148	0.156	0.163	0.171	0.179
14	0.077	0.083	0.089	0.095	0.101	0.108	0.114	0.121	0.128	0.136	0.143	0.151	0.159	0.167	0.175
15	0.072	0.078	0.084	0.090	0.096	0.103	0.110	0.117	0.124	0.131	0.139	0.147	0.155	0.163	0.171
16	0.068	0.074	0.080	0.086	0.092	0.099	0.106	0.113	0.120	0.128	0.136	0.143	0.151	0.160	0.168
17	0.064	0.070	0.076	0.082	0.089	0.095	0.102	0.110	0.117	0.125	0.132	0.140	0.149	0.157	0.165
18	0.061	0.067	0.073	0.079	0.086	0.092	0.099	0.107	0.114	0.122	0.130	0.138	0.146	0.155	0.163
19	0.058	0.064	0.070	0.076	0.083	0.090	0.097	0.104	0.112	0.120	0.128	0.136	0.144	0.153	0.161
20	0.055	0.061	0.067	0.074	0.080	0.087	0.094	0.102	0.110	0.117	0.126	0.134	0.142	0.151	0.160
21	0.053	0.059	0.065	0.071	0.078	0.085	0.092	0.100	0.108	0.116	0.124	0.132	0.141	0.150	0.158
22	0.051	0.057	0.063	0.069	0.076	0.083	0.090	0.098	0.106	0.114	0.122	0.131	0.139	0.148	0.157
23	0.049	0.055	0.061	0.067	0.074	0.081	0.089	0.096	0.104	0.113	0.121	0.130	0.138	0.147	0.156
24	0.047	0.053	0.059	0.066	0.072	0.080	0.087	0.095	0.103	0.111	0.120	0.128	0.137	0.146	0.155
25	0.045	0.051	0.057	0.064	0.071	0.078	0.086	0.094	0.102	0.110	0.119	0.127	0.136	0.145	0.155
26	0.044	0.050	0.056	0.063	0.070	0.077	0.085	0.093	0.101	0.109	0.118	0.127	0.136	0.145	0.154
27	0.042	0.048	0.055	0.061	0.068	0.076	0.083	0.091	0.100	0.108	0.117	0.126	0.135	0.144	0.154
28	0.041	0.047	0.053	0.060	0.067	0.075	0.082	0.090	0.099	0.107	0.116	0.125	0.134	0.144	0.153
29	0.040	0.046	0.052	0.059	0.066	0.074	0.081	0.090	0.098	0.107	0.116	0.125	0.134	0.143	0.153
30	0.039	0.045	0.051	0.058	0.065	0.073	0.081	0.089	0.097	0.106	0.115	0.124	0.133	0.143	0.152

number of years n	interest rate i (%) 16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
1	1.160	1.170	1.180	1.190	1.200	1.210	1.220	1.230	1.240	1.250	1.260	1.270	1.280	1.290	1.300
2	0.623	0.631	0.639	0.647	0.655	0.662	0.670	0.678	0.686	0.694	0.702	0.711	0.719	0.727	0.735
3	0.445	0.453	0.460	0.467	0.475	0.482	0.490	0.497	0.505	0.512	0.520	0.528	0.535	0.543	0.551
4	0.357	0.365	0.372	0.379	0.386	0.394	0.401	0.408	0.416	0.423	0.431	0.439	0.446	0.454	0.462
5	0.305	0.313	0.320	0.327	0.334	0.342	0.349	0.357	0.364	0.372	0.379	0.387	0.395	0.403	0.411
6	0.271	0.279	0.286	0.293	0.301	0.308	0.316	0.323	0.331	0.339	0.347	0.354	0.362	0.370	0.378
7	0.248	0.255	0.262	0.270	0.277	0.285	0.293	0.301	0.308	0.316	0.324	0.332	0.340	0.349	0.357
8	0.230	0.238	0.245	0.253	0.261	0.268	0.276	0.284	0.292	0.300	0.309	0.317	0.325	0.333	0.342
9	0.217	0.225	0.232	0.240	0.248	0.256	0.264	0.272	0.280	0.289	0.297	0.306	0.314	0.323	0.331
10	0.207	0.215	0.223	0.230	0.239	0.247	0.255	0.263	0.272	0.280	0.289	0.297	0.306	0.315	0.323
11	0.199	0.207	0.215	0.223	0.231	0.239	0.248	0.256	0.265	0.273	0.282	0.291	0.300	0.309	0.318
12	0.192	0.200	0.209	0.217	0.225	0.234	0.242	0.251	0.260	0.268	0.277	0.286	0.295	0.304	0.313
13	0.187	0.195	0.204	0.212	0.221	0.229	0.238	0.247	0.256	0.265	0.274	0.283	0.292	0.301	0.310
14	0.183	0.191	0.200	0.208	0.217	0.226	0.234	0.243	0.252	0.262	0.271	0.280	0.289	0.298	0.308
15	0.179	0.188	0.196	0.205	0.214	0.223	0.232	0.241	0.250	0.259	0.268	0.278	0.287	0.297	0.306
16	0.176	0.185	0.194	0.203	0.211	0.220	0.230	0.239	0.248	0.257	0.267	0.276	0.285	0.295	0.305
17	0.174	0.183	0.191	0.200	0.209	0.219	0.228	0.237	0.246	0.256	0.265	0.275	0.284	0.294	0.304
18	0.172	0.181	0.190	0.199	0.208	0.217	0.226	0.236	0.245	0.255	0.264	0.274	0.283	0.293	0.303
19	0.170	0.179	0.188	0.197	0.206	0.216	0.225	0.235	0.244	0.254	0.263	0.273	0.283	0.292	0.302
20	0.169	0.178	0.187	0.196	0.205	0.215	0.224	0.234	0.243	0.253	0.263	0.272	0.282	0.292	0.302
21	0.167	0.177	0.186	0.195	0.204	0.214	0.223	0.233	0.243	0.252	0.262	0.272	0.282	0.291	0.301
22	0.166	0.176	0.185	0.194	0.204	0.213	0.223	0.232	0.242	0.252	0.262	0.271	0.281	0.291	0.301
23	0.165	0.175	0.184	0.194	0.203	0.213	0.222	0.232	0.242	0.251	0.261	0.271	0.281	0.291	0.301
24	0.165	0.174	0.183	0.193	0.203	0.212	0.222	0.232	0.241	0.251	0.261	0.271	0.281	0.291	0.301
25	0.164	0.173	0.183	0.192	0.202	0.212	0.222	0.231	0.241	0.251	0.261	0.271	0.281	0.290	0.300
26	0.163	0.173	0.182	0.192	0.202	0.211	0.221	0.231	0.241	0.251	0.261	0.271	0.280	0.290	0.300
27	0.163	0.172	0.182	0.192	0.201	0.211	0.221	0.231	0.241	0.251	0.261	0.270	0.280	0.290	0.300
28	0.163	0.172	0.182	0.191	0.201	0.211	0.221	0.231	0.241	0.250	0.260	0.270	0.280	0.290	0.300
29	0.162	0.172	0.181	0.191	0.201	0.211	0.221	0.231	0.240	0.250	0.260	0.270	0.280	0.290	0.300
30	0.162	0.172	0.181	0.191	0.201	0.211	0.221	0.230	0.240	0.250	0.260	0.270	0.280	0.290	0.300

APPENDIX J:

MANUFACTURERS OF PATs AND LOAD CONTROLLERS

(list not exhaustive)

1. Pump Manufacturers with experience in the field of PATs

C.I.M.H.
Constructeur de turbines
42 Avenue Jacques Desplats
81100 Castres France Phone 63 35 63 15 Fax 63 59 57 21

Klein Schanzlin Becker AG (KSB)
Johann Klein Str. 9
D 6710 Frankenthal (Pfalz) Germany Phone 06233/86-0 Telex 465 211-0 ks Fax 06233/863311

Ritz Pumpenfabrik
GmbH
Becherlehenstr.
D 7070 Schwäbisch Gmünd Germany Phone 07171 / 6090 Telex 7 248 806 Fax 07171/609287

Rütschi AG
Pumpenbau Brugg
CH 5200 Brugg Switzerland Phone 056/410455 Telex 825130 Fax 056/411331

Sulzer AG Pump Division
CH 8401 Winterthur Switzerland Telex 89606040 SZCH Fax 052/225401

Worthington Group
PUMP Division
Mc.Graw-EDISON Company
Taneytown, MD 21787

2. Suppliers of electronic load controllers

Mr.Gerry Pope
G.P. Electronics
Bovey Tracey
Devon TQ13 9DS, U.K. Phone No. U.K. 0626 832670

Thomson and Howe Energy Systems Inc.
Site 17, Box 2 , S.S. 1
Kimberley, British Columbia / CANADA Phone 604/427-4326 Fax 604/427-7723 or -4326

IREM SPA-SEDE /Head Quarters
Mr. Alberto Bonini
Via Vaie 42
I- 10050 S. Antonino /ITALY Phone 011/ 649133 Telex 212134 Fax 011/ 96 49 933

Mr. N. Smith
Dept.. of Electrical Engineering
Trent Polytechnic
Burton Street
Nottingham NG1 4BU / U.K. Phone 0602 / 418 418 FAX 0602 / 484 266

V.Schnitzer (he has also experience with PATs)
Hydro Power
Industriestr. 68
D-6901 Bammental
Germany

Gugler KG
A-4085 Niederanna 41 Austria Phone 07285/514536 TELEX 116540 Fax 07285/6242

H.Kobel
Elektroapparatebau
CH-3416 Affoltern i.E. SWITZERLAND Phone 034 / 751413
(contact adress international : J. M. Chapallaz Dryade 2 CH 1450 Ste. Croix
SWITZERLAND Phone/FAX 024 / 611042)

Würtemberger & Haas Mr. H.P. Roth
Bannwaldallee 44-46
D-7500 Karlsruhe /Germany Phone 0721 / 551022 Telex 7826823 FAX 0721 / 557154

Ets. FERRERO
Electronique
B.P. 16
F-32600 L'Isle Jourdain / France Phone 62 07 01 51 FAX 62 07 02 44

ITDG Myson House
Railway Terrace
Rugby CV 21 3 HT / U.K. Phone 0788 / 60631 FAX 0788 / 540270

Mr. Jacques Dos Ghali
Ecole Polytechnique Fédérale, Lausanne (EPFL)
ELG - Ecublens
CH 1015 Lausannen/ Switzerland Phone 021 / 693 26 36

APPENDIX K:

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