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Contribution for Future Insights on Hydraulic Machines Lectures: A Portrayal Assessment of Actual Testing Methodologies at UMH and ISEC

Project Report to fulfill the Master's degree in Mechanical Engineering

Specialization in Project, Installation and Maintenance of Thermal Systems

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ABSTRACT

The modern world requires engineers to lead technological development, whether by creating new technologies or using existing ones more sustainably, with significant economic and social impacts. While technical knowledge traditionally comes from years of experience, engineers are now required to confront real-world challenges earlier, ideally during their academic training. Practical learning experiences equip future engineers with a strong understanding of both theoretical concepts and practical applications, while also developing the essential *hard* and *soft* skills necessary for their profession. However, providing practical learning experiences in the field of turbomachinery as closely as possible to reality is often challenging, due to the difficulty in replicating and maintaining the necessary operating conditions with the equipment typically available in educational institutions. The present work constitutes a contribution for future insights on hydraulic turbomachinery lectures, serving as a portrayal assessment of current testing methodologies at Instituto Superior de Engenharia de Coimbra (ISEC) and Universidad Miguel Hernández de Elche (UMH). The theoretical concepts related to each turbomachine are addressed, and the experimental setups and methodologies used to obtain performance curves and other relevant quantities that characterize the behavior of the turbomachines are described. The results are discussed in detail from both qualitative and quantitative perspectives, linking them to their theoretical foundations or other available literature whenever possible. Analogies are also drawn between the real operation of hydraulic turbomachines and their reproduction in laboratory and pedagogical contexts. The richness of the content presented, particularly from a pedagogical perspective, is reflected in a vast and diverse set of data collected through various testing methodologies and setups, with the primary goal of advancing engineering education in the field of turbomachinery.

Keywords: Turbomachinery performance, Pelton turbine, Centrifugal pump, Engineering education

Francisco José Ramos Morais Calhindo

*Aos meus queridos Pais e Irmã
Aos meus eternos Avós
À Adriana*

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NOMENCLATURE

B, B'	pump operation points
B_i	systematic uncertainty (interval of a normal distribution with $C\%$ confidence)
$C\%$	percentage of confidence level
C_H	head coefficient
C_P	power coefficient
C_Q	flow coefficient
D	characteristic diameter [m]
E	total energy per unit of mass [J/kg]
F	braking force [kgf]
g	acceleration of gravity [m/s ²]
H	total head [m]
h	enthalpy [J/kg]
h_a	pump actual head rise [m]
H_e	effective available head [m]
h_f	head loss [m]
H_g	total gross available head [m]
H_i	installation head curve [m]
h_i	pump ideal head rise or Euler Head [m]
H_n	net available head [m]
I	electric current [A]
i	generic variable
J	number of variables
K_N	nozzle velocity coefficient
$k\%$	expansion factor
k	factor accounting for friction loss in flow through the buckets of Pelton turbine
k_M	torque constant
l	characteristic length [m]

$\dot{m}$	mass flow rate [kg/s]
m	mass [kg]
N	sample size
n	rotational speed [rpm]
P	power [W]
p	pressure [Pa]
P_h	hydraulic power [W]
P_i	random uncertainty (interval of a normal distribution with $C\%$ confidence)
p_v	vapor pressure [Pa]
p_{atm}	atmospheric pressure [Pa]
P_{tank}	feed tank pressure [Pa]
$P_{turbine}$	pressure at turbine inlet [Pa]
Prob	probability (area under the Gaussian curve)
$\dot{Q}$	heat transferred per unit of time [W]
Q	volumetric flow rate [m ³ /s]
q	heat transferred per unit of mass [J/kg]
r	radial distance to the axis of rotation[m]
r_m	mean radial distance to the axis of rotation [m]
s	entropy [J/kgK]
S_B	standard deviation for systematic uncertainty
S_i	standard deviation for random uncertainty of a generic variable i
S_x	sample standard deviation
S_{B_i}	standard deviation for systematic uncertainty of a generic variable i
T	temperature [K]
t	t-value from the Student's t-distribution
t	time [s]
T_{shaft}	shaft torque [Nm]
U_x	uncertainty associated with the generic variable X , for a confidence level of $C\%$
$U_{\%}$	expanded uncertainty
U	blade linear velocity [m/s]
u	internal energy per unit of mass [J/kg]

u_c	combined standard uncertainty
u_r	combined standard uncertainty of the result
$\vec{V}$	absolute fluid velocity [m/s]
v	average fluid velocity [m/s]
v	degrees of freedom of the sample
V_θ	tangential component of absolute fluid velocity [m/s]
$\dot{W}$	work per unit of time [W]
$\dot{W}_{shaft}$	shaft power [W]
$\vec{W}$	fluid velocity relative to the blade [m/s]
w	work per unit of mass [J/kg]
w_{shaft}	shaft work per unit of mass [J/kg]
X_{best}	best estimate of a generic variable
$\bar{X}$	sample mean
X	generic variable
X_{true}	true value of a generic variable
z	coordinate in vertical direction [m]
β	blade angle (ch. 3) or jet deflection angle (ch. 5) [°]
ϵ	absolute surface roughness [m]
η	efficiency
η_e	electrical efficiency
η_h	hydraulic efficiency
η_m	mechanical efficiency
η_o	overall efficiency
η_P	efficiency of the pump
η_T	efficiency of the turbine
η_t	isentropic efficiency
μ	fluid dynamic viscosity [Pas] (ch. 3) or population mean (ch. 6)
ω	rotor angular velocity [rad/s]
Φ	speed ratio or peripheral speed
ρ	density [kg/m ³]
σ	population standard deviation

σ^2	population variance
τ	normalized deviation
θ	sensitivity coefficient

ACRONYMS

BEP Best Efficiency Point. 31

EES Engineering Equation Solver. 89

EGL Energy Grade Line. 59

EPFL École Polytechnique Fédérale de Lausanne. 13

FEUP Faculdade de Engenharia da Universidade do Porto. 70

HGL Hydraulic Grade Line. 59

ISEC Instituto Superior de Engenharia de Coimbra. 13

LHM Laboratory of Hydraulic Machines. 36

NPSH Net Positive Suction Head. 28

PBL Project-Based Learning. 93

SEFI European Society for Engineering Education. 13

UMH Universidad Miguel Hernández de Elche. 13

VFD Variable-Frequency Drive. 37

1 INTRODUCTION

1.1 Motivation and Objectives

The increasingly complex world demands that engineers be at the forefront of technological development, either by responsibly creating new technologies or by using older ones more sustainably, with consequential economic and social impacts. Much of the technical knowledge engineers possess comes from years of experience within their respective fields. However, the current paradigm requires engineers to be aware of real-world challenges at an earlier stage of their professional journey, ideally during their academic training. By providing practical learning experiences such as hands-on projects, internships, case studies, simulations, industry collaborations, and competitions, future engineers develop a comprehensive understanding of engineering concepts and their practical applications. This solidifies their theoretical knowledge, making the learning experience more tangible and memorable, ensuring they are well-prepared to respond to often unexpected technical demands. These experiences also equip them with essential skills to conduct experiments, perform research and development, and process and analyze data using various software tools. At the same time, fostering soft skills such as teamwork, communication, time management, and adaptability through group projects and laboratory work, where individuals learn to share, collaborate, and communicate their findings, often while managing projects with strict deadlines.

Mechanical engineering programs typically include core courses covering areas such as Design and Manufacturing, Mechanics and Materials, Dynamics and Controls, and Thermal Sciences. One of the practical applications within the Thermal Sciences area is Turbomachinery - machines used in fields such as power generation (e.g., hydro, thermal, or wind power plants), aerospace, water supply, irrigation and drainage, HVAC systems, and firefighting and smoke extraction.

However, providing practical learning experiences in the field of turbomachinery as closely as possible to reality is often challenging, as it is difficult to replicate, to provide and to sustain the necessary conditions for its operation with the equipment commonly available in educational institutions; this includes all the auxiliary measuring equipment required to perform detailed tests.

Thus, the present work constitutes a contribution for future insights on hydraulic turbomachinery lectures, serving as a portrayal assessment of current testing methodologies at Instituto Superior de Engenharia de Coimbra (ISEC) and Universidad Miguel Hernández de Elche (UMH). It provides a pedagogical perspective on how to test hydraulic turbomachinery, particularly centrifugal pumps and Pelton turbines, and on developing appropriate setups for their testing, aiming to connect theoretical concepts with real-world challenges, at a smaller scale.

This contribution represents one outcome of the Erasmus+ program that started in April 2023, and since then is being developed and enhanced between UMH and ISEC.

Part of the work presented here was submitted as a paper and presented at the 52nd Annual Conference of the European Society for Engineering Education (SEFI), held at École Polytechnique Fédérale de Lausanne (EPFL), titled *PELTON TURBINE UNDER DIFFERENT OPERATING CONDITIONS: An Experimental Pedagogical Study*.

1.2 Structure of the Dissertation

The structure of the dissertation is described in the following paragraphs. Following this introduction, which situates the general topic of study and defines the objectives and motivation of the work, Chapter 2 provides a background on turbomachinery, defining the different existing turbomachines and their respective classifications based on the different existing criteria. Moreover, a brief overview of energy analysis is presented, focusing on the perspectives offered by the first and second laws of thermodynamics. This enables a discussion on turbine and pump efficiencies concerning thermodynamic losses, as well as Euler's equation for turbomachines.

Chapter 3 presents a comprehensive overview of centrifugal pumps, delving into some theoretical topics not mentioned in Chapter 2, identifying the major aspects of centrifugal pump operation such as cavitation, pump performance, and system characteristics curves, control and adjustment of pump performance, and association of pumps. A brief overview of dimensionless parameters and similarity laws is also presented. Chapter 3 represents a background of Chapter 4, combining theoretical and practical engineering concepts.

Chapter 4 covers the performance testing of centrifugal pumps, providing a practical application of the content mentioned in Chapter 3, especially at the pedagogical level. The experimental setups used at two different educational institutions, ISEC and UMH, with their distinct capabilities and particularities, are described.

Subsequently, the experimental procedures to be followed at each institution, depending on the curves to be obtained and the capacities of each institution, are described. A detailed assessment of the different topics covered in Chapter 3 and the curves obtained are presented, comparing and linking the results to their theoretical foundations whenever possible.

Chapter 5, as Chapter 3, presents a comprehensive overview of the Pelton turbine, beginning with a detailed description of the different forms and transformations of energy that occur in a Pelton turbine, leading to the respective characteristic performance curves, their acquisition, and the influence of the relevant parameters on its behavior and performance. Analogies between the operation of a Pelton turbine in a hydroelectric power plant and its reproduction in a laboratory and pedagogical context are drawn whenever possible.

Chapter 6 covers the performance testing of the Pelton turbine under a wide and detailed range of operating conditions. Once again, the experimental setups used at two different educational institutions, ISEC and UMH, with their distinct capabilities and particularities, are described. Subsequently, the experimental procedures to be followed at each institution, depending on the curves to be obtained and the capacities of each institution, are presented. A detailed analysis and discussion of the results obtained, including the different variables involved in the governing equations, are conducted, comparing them with theoretical predictions and findings from the literature whenever possible. A brief uncertainty analysis is also presented.

Chapter 7 concludes the dissertation, providing final remarks, summarizing the conclusions drawn in the previous chapters, and suggesting some perspectives for future research.

2 INTRODUCTION TO TURBOMACHINERY

2.1 Definition and Classification of Turbomachines

In a broad sense, a turbomachine can be defined as a device that exchanges mechanical energy with a flowing fluid in the form of shaft work. The energy transfer between the continuous flow of a fluid and the machine occurs through a continuously rotating element referred to as a rotor ¹, which is associated with a set of blades responsible for effecting the energy exchange with the fluid upon contact, based on flow-generated forces (Dick, 2015; Oliveira & Lopes, 2020). Figure 2.1 shows some of the many varieties of turbomachines currently available.

Energy may be transferred from the flow to the rotating machine components or vice versa. If the energy transfer occurs from the fluid to the rotor, the device is referred to as power delivering or turbine, irrespective of the fluid, which can be water, steam, air, gas or other fluid, as refrigerants in cooling systems. If the fluid receives energy imparted by the rotor, driven by an external motor, it is called power receiving but different designations can be used depending on the fluid and energy component that is mainly increased: velocity-associated energy (kinetic energy) or pressure-associated energy (pressure potential energy in case of a constant density fluid). For incompressible fluids, such as water, pumps are normally employed, with the fluid registering a dominantly pressure increase, whereas for compressible fluids such as air, gas or vapour, a compressor or a fan can be used depending if the objective is providing the fluid mainly with pressure energy or kinetic energy, respectively (Dick, 2015).

Besides the classification based on energy exchange, another important classification arises from the internal geometry of the turbomachine and the nature of the path described by the fluid as it passes through the rotor. If the main direction of the flow occurs in the plane of rotation, the turbomachine is referred to as a radial-flow machine.

In the case where the main direction of the flow is aligned parallel to the axis of rotation, the machine is considered an axial-flow machine. Turbomachines with partially radial and axial flow characteristics are designated as mixed-flow machines. A particular classification called tangential-flow is one where the flow occurs in a plane perpendicular to the axis of rotation and predominantly in the tangential direction (Oliveira & Lopes, 2020; Viegas, 1989; Munson et al., 2013).

¹Instead of the term rotor, general terms describing the role of the rotor in the machine can be used including the term *impeller* (pumps) or *runner* (hydraulic turbines).

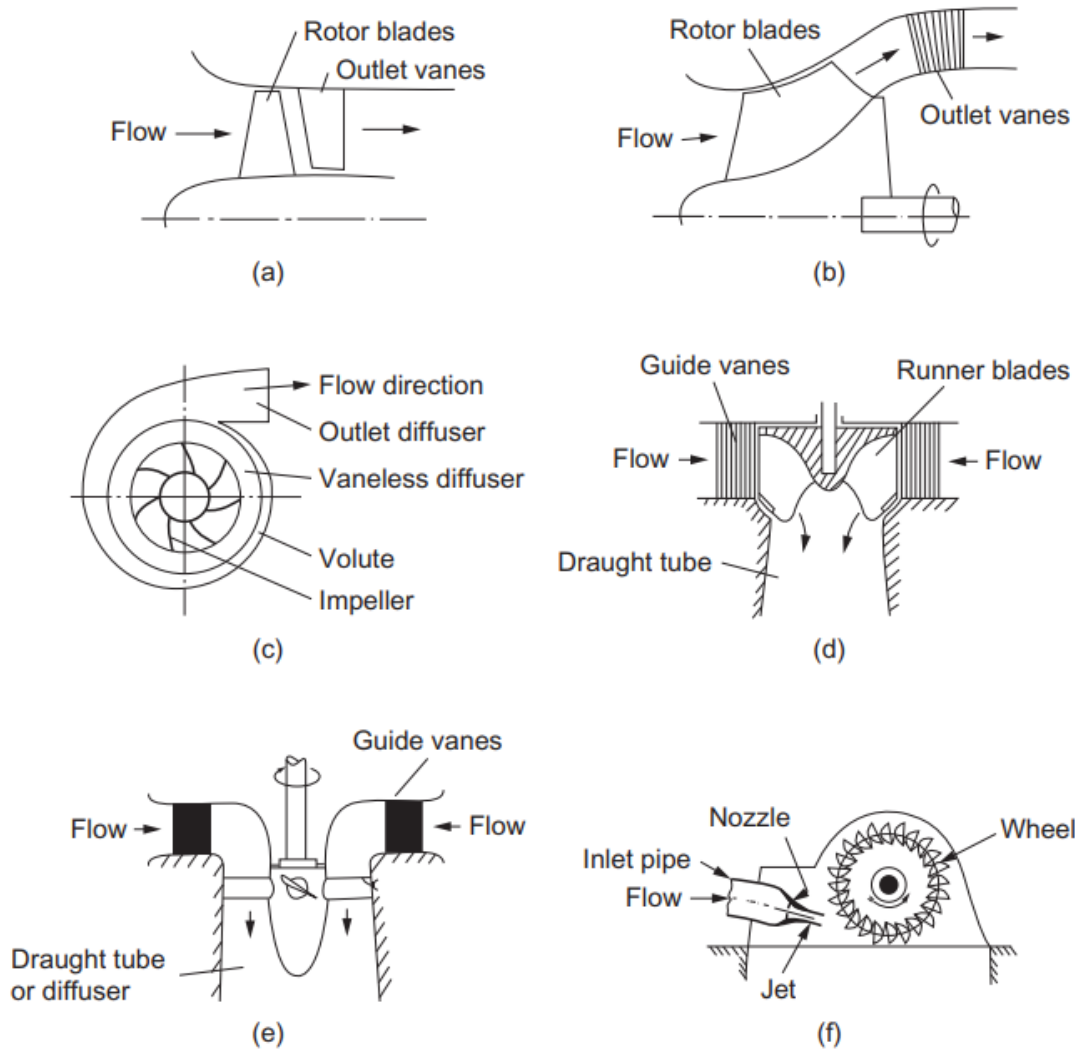


Figure 2.1: Examples of turbomachines. (a) Single stage axial flow compressor or pump, (b) mixed flow pump, (c) centrifugal compressor or pump, (d) Francis turbine (mixed flow type), (e) Kaplan turbine, and (f) Pelton wheel (Dixon & Hall, 2014).

In terms of flow restriction, turbomachines can be classified as open or closed: if the machine contains some type of housing or casing that surrounds the rotating blades or rotor, forming an internal flow passageway through which the fluid flows, it is called open. Otherwise, if unducted, it is called closed.

2.2 Brief Overview on Energy Analysis

This section provides a concise overview of energy analysis, focusing on the perspectives offered by the first and second laws of thermodynamics, enabling a discussion on turbine and pump efficiencies concerning thermodynamic losses, as well as Euler's equation for turbomachines.

While passing through the machine, the fluid exchanges energy with the surroundings, either receiving in the case of pumps or delivering in the case of turbines, evolving from an initial to a final state. The flow through these types of turbomachines is generally adiabatic ($q \approx 0$) since heat transfer is negligible compared to other energy exchanges involved in the process. Despite instabilities within the machine, if a control volume surrounding the turbomachine is considered, the flow through its boundaries can be essentially considered steady, and the first law of thermodynamics can be expressed in the form of the steady-flow energy equation (Dick, 2015; Viegas, 1989)

$$\dot{m}(h_1 + \frac{1}{2}v_1^2 + gz_1) + \dot{Q} = \dot{m}(h_2 + \frac{1}{2}v_2^2 + gz_2) + \dot{W} \quad (2.1)$$

where $\dot{m}$ is the mass flow rate of fluid, $v^2/2$ the kinetic energy and gz the potential energy, with g being the acceleration of gravity and z a height. The enthalpy, h , is given by $u + pv$ where u is the internal energy of the fluid and pv the so called *flow work*. This term represents the work done by the fluid to move the fluid ahead of it into and out of the control volume. The heat transfer rate into the control volume is denoted as $\dot{Q}$ and the rate at which work is delivered is $\dot{W}$. Dividing both sides by $\dot{m}$, the first law of thermodynamics comes in the form, (Korpela, 2011)

$$q - w = (h_2 - h_1) + \frac{1}{2}(v_2^2 - v_1^2) + g(z_2 - z_1) \quad (2.2)$$

admitting ($q \approx 0$) and

$$h = u + \frac{p}{\rho} \quad (2.3)$$

$$w = (\frac{p_2}{\rho} + \frac{v_2^2}{2} + gz_2) - (\frac{p_1}{\rho} + \frac{v_1^2}{2} + gz_1) + (u_2 - u_1) \quad (2.4)$$

Regardless of how the evolution from an initial state to a final state occurred, the equation above encapsulates the energy absorbed or released per unit mass of the fluid while undergoing the referred evolution. The simplification of considering a control volume surrounding the turbomachine is entirely valid and allows for a transition from an extended integral analysis over the control surface to an analysis in terms of average values at the inlet and outlet sections of the control volume.

In these types of systems, there is no conversion of nuclear, chemical, or thermal energy into mechanical energy. Thus, such systems can be conveniently analyzed by considering only the mechanical forms of energy and the effects of friction that cause the loss of mechanical energy, as in equation (2.4). Mechanical energy can be defined as the form of energy that can be completely and directly converted into mechanical work by an ideal mechanical device. The transfer of mechanical energy is typically done by a rotating shaft, thus the mechanical work is referred to as shaft work, w_{shaft} .

A pump receives shaft work and transfers it to the fluid in the form of mechanical energy. A turbine converts the mechanical energy of a fluid into shaft work.

When dealing with incompressible fluids, the shaft power is commonly expressed in terms of *head* (H) and equation (2.4) comes

$$w_{shaft} = g\Delta H + \Delta u \quad (2.5)$$

where H is defined as

$$H = \frac{p}{\rho g} + \frac{v^2}{2g} + z \quad (2.6)$$

In hydraulics, the term *head* is used for mechanical energy divided by the gravity acceleration and is expressed as a height, in meters of flowing fluid. This term is also used for the rise of this quantity by a pump and the drop of it by a hydraulic turbine, as will be discussed later. In equation (2.5) the term $g\Delta H$ corresponds to the maximum work that could be produced in an idealized flow.

Even though the evolution can be considered adiabatic, it cannot be considered reversible, meaning that is non-isentropic due to the presence of viscous forces, turbulence and separations, which are accompanied by an energy loss compared to the scenario where the final state would have been achieved through an isentropic evolution, as postulated in the second law of thermodynamics. In fact, it is possible to observe in equation (2.5) that not all shaft work can be converted into useful energy ($g\Delta H$) per unit of mass, or vice versa, leading to an associated increase in internal energy (Δu) and consequently an elevation in fluid temperature.

Going back to equation (2.2), it can be seen that what distinguishes a real evolution from an ideal (isentropic) one is the variation in enthalpy, given that the remaining parameters will be the same. Figure 2.2 illustrates the real and ideal evolutions in a turbine (on the left) and in a pump (on the right).

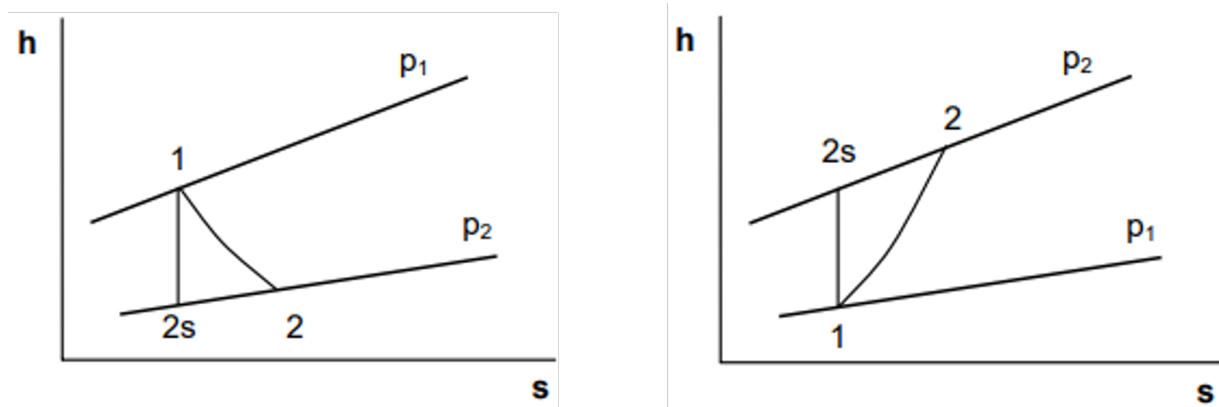


Figure 2.2: h - s diagrams for turbines (left) and pumps (right) (Mendes, 2005).

If both evolutions, in turbines and pumps, are plotted on an h - s diagram (Fig.(2.2)), denoting the energy involved in the actual evolution as E and the corresponding isentropic evolution as E_s , the equation (2.2) can be rewritten as

$$E = (h_2 - h_1) + \frac{1}{2}(v_2^2 - v_1^2) + g(z_2 - z_1) \quad (2.7)$$

$$E_s = (h_{2s} - h_1) + \frac{1}{2}(v_2^2 - v_1^2) + g(z_2 - z_1) \quad (2.8)$$

Since,

$$dh = T \cdot ds + \frac{dp}{\rho} \quad (2.9)$$

and considering an isentropic evolution ($ds = 0$)

$$dh_s = \frac{dp}{\rho} \quad (2.10)$$

for incompressible fluid, comes

$$h_{2s} - h_1 = \frac{p_2 - p_1}{\rho} \quad (2.11)$$

obtaining

$$E_s = \frac{p_2 - p_1}{\rho} + \frac{1}{2}(v_2^2 - v_1^2) + g(z_2 - z_1) \quad (2.12)$$

It can be seen that equation (2.12) only differs from equation (2.4) in the sense that since the evolution is non-isentropic we must consider the work done to overcome the viscous resistance, as expressed in equation (2.4). Consequently, the temperature of the fluid rises; the internal energy increases, and generally, heat (q) is transferred from the fluid to its surroundings. Since the rise in temperature and the subsequent increase in internal energy are not of interest in these type of machines, the increase in internal energy, as well as q are normally treated as a loss of useful energy, being the total loss per unit mass ($u_2 - u_1 - q$). For a constant fluid density, it is normal to represent this loss of useful energy due to friction as an energy per unit weight, denoted by h_f , so that

$$h_f = \frac{u_2 - u_1 - q}{g} \quad (2.13)$$

It is important to notice that h_f is not necessarily an energy loss. It represents a loss in terms of $(-q/g)$, but not in terms of $(u_2 - u_1)/g$, since what is observed is a decrease in the total mechanical energy, hence it is said to be a loss of useful energy (Mendes, 2005).

2.3 Definitions of Efficiency

It is now possible to define the efficiency of turbines and pumps. In fact, delving into the literature on turbomachinery, it can be observed that a considerable number of efficiency definitions can be found (Dixon & Hall, 2014). As discussed, hydraulic turbines convert the available hydraulic energy in a flowing fluid into useful mechanical work in the form of shaft work. The efficiency of this process is denoted as *overall efficiency* η_o and is expressed by the following ratio

$$\eta_o = \frac{\text{mechanical energy available at coupling of output shaft in unit time}}{\text{maximum energy difference possible for the fluid in unit time}} \quad (2.14)$$

The discrepancy between the maximum energy difference possible for the fluid and the mechanical energy supplied to the rotor, in unit time, is due not only to mechanical energy losses but also to the inherent non-isentropic nature of the transformation, as mentioned above. The mechanical energy losses occur between the turbine rotor and the output shaft coupling as a result of the work done against friction at the bearings and seals. Other types of mechanical losses, characteristic of Pelton turbines, will be addressed later in Chapter 5. The magnitude of this loss varies with the size and individual design of the turbomachine and, therefore, is difficult to estimate. According to Dixon & Hall (2014), small machines with power outputs in the range of several kilowatts may exhibit losses of 5% or higher, whereas in medium and large-scale machinery, this loss ratio can decrease to a value as low as 1%.

The non-isentropic nature of the transformation also allows to define an *isentropic efficiency* η_t or *hydraulic efficiency* η_h , in the case of hydraulic turbines, as the following ratio

$$\eta_h = \frac{\text{mechanical energy supplied to the rotor in unit time}}{\text{maximum energy difference possible for the fluid in unit time}} = \frac{E}{E_s} \quad (2.15)$$

which corresponds to the ratio between the energy present in the real evolution and the energy corresponding to the case of an isentropic evolution. Similarly to the mechanical losses, the hydraulic (isentropic) losses, characteristic of the Pelton turbine, will be addressed ahead in Chapter 5.

As discussed before, it is common to associate E_s (energy per unit mass) with a height H (energy per unit weight), defined by

$$E_s = g \times H \quad (2.16)$$

In the case of hydraulic turbines, H is referred to as the net available head after deducting the losses along the pipes².

Combining both expressions, η_o and η_h , it is possible to define the *mechanical efficiency* η_m , which is simply the ratio of shaft power to rotor power

$$\eta_m = \frac{\eta_o}{\eta_h} = \frac{\text{mechanical energy available at coupling of output shaft in unit time}}{\text{mechanical energy supplied to the rotor in unit time}} \quad (2.17)$$

The same approach can be followed in the case of pumps. In this case, the *overall efficiency*, η_o , is given by

$$\eta_o = \frac{\text{useful (hydrodynamic) energy input to fluid in unit time}}{\text{power input to coupling of shaft}} \quad (2.18)$$

Due to external energy losses in the bearings and seals, the power input to the rotor (or impeller) is always less than the power supplied at the coupling. Thus, the *hydraulic efficiency* of a pump η_h is defined as

$$\eta_h = \frac{\text{useful (hydrodynamic) energy input to fluid in unit time}}{\text{power input to rotor}} = \frac{E_s}{E} \quad (2.19)$$

In the case of pumps, the term H in equation (2.16) is referred to as the lift height. In the scenario of a hydraulic pump rising water from one reservoir to another (both large-scale), the height H represents the difference in level between the free surfaces of the reservoirs, plus the height equivalent to the head loss that occurs through the pipes and accessories.

Hence, the *mechanical efficiency* η_m is

$$\eta_m = \frac{\eta_o}{\eta_h} = \frac{\text{power input to rotor}}{\text{power input to coupling of shaft}} \quad (2.20)$$

2.4 Brief Overview on Governing Equations

As seen in the previous sections, the transfer of energy to or from a fluid flowing through a pump or a turbine occurs through a continuously rotating element referred to as a rotor. Due to that, since there is a rotation of an element about a central axis, it is interesting to discuss the performance of these machines in terms of torque and angular momentum, constituting an application of Newton's second law in the form where it applies to the moment of forces (Munson et al., 2013; Korpela, 2011).

²different designations for the term *head* will be detail further on, depending on the energy considered.

Considering a system of mass m , the vector sum of the moments of all external forces acting on the system about a given axis (sum of torques) is equal to the time rate of change of angular momentum of the system about that axis, i.e.,

$$T_{shaft} = m \cdot \frac{d}{dt}(rV_{\theta}) \quad (2.21)$$

where r is the distance from the axis of rotation measured along the normal to the axis and V_{θ} the velocity component perpendicular both to the axis and radius vector r , i.e., the tangential component of velocity.

For a given arbitrary control volume surrounding a generalized turbomachine, as illustrated in Figure 2.3, the fluid enters the control volume at a radial position r_1 with tangential velocity V_{θ_1} and leaves at radius r_2 with tangential velocity V_{θ_2} , leading to

$$T_{shaft} = \dot{m}(r_2 V_{\theta_2} - r_1 V_{\theta_1}) \quad (2.22)$$

where T_{shaft} is the *shaft torque* applied to the contents of the control volume.

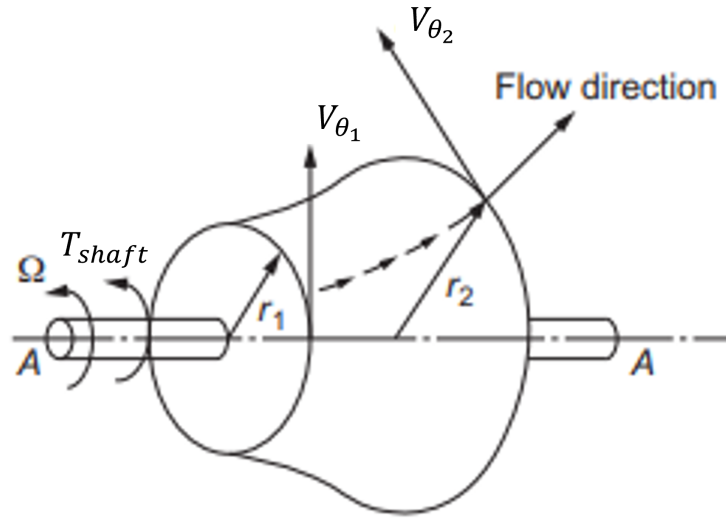


Figure 2.3: Control volume for a generalized turbomachine. Adapted from Dixon & Hall (2014).

It should be noted that the fully moment-of-momentum equation (cf. Munson et al. (2013)) was simplified by considering that the flows are one-dimensional, meaning uniform distributions of average velocity at any section; steady or steady-in-the-mean cyclical flows and only axial component (resolved along the axis of rotation).

Equation (2.22) is the so called *Euler turbomachine equation* and it is considered to be the most important basic equation in turbomachinery theory.

According to Oliveira & Lopes (2020), some considerations regarding the general character of *Euler equation* should be noted. Firstly, it does not impose restrictions on incompressibility (variable ρ). Secondly, it does not account for the trajectory of the fluid between the inlet and outlet sections; instead, it only focuses on the change in velocity, thus neglecting losses such as turbulence and friction between fluid and blades along this trajectory. Finally, it considers the conditions of the inlet and outlet sections to be uniform, which would mean an infinite number of blades, limiting it to being used only as a first approximation. By applying the action-reaction principle, Euler turbomachine equation can be extended to turbines simply by multiplying it by (-1), as will be seen.

For a rotor running at angular velocity w , the rate at which the rotor and the fluid interact, in the form of work, the *shaft power*, $\dot{W}_{shaft}$, is given by

$$\dot{W}_{shaft} = T_{shaft} \cdot \omega \quad (2.23)$$

From Equation (2.22) and (2.23) and considering the blade speed to be $U = w \cdot r$, the *Euler's pump equation* is obtained

$$\dot{W}_{shaft,P} = \dot{m}(U_2 V_{\theta_2} - U_1 V_{\theta_1}) \quad (2.24)$$

By multiplying the above equation by (-1), the *Euler's turbine equation* is obtained

$$\dot{W}_{shaft,T} = \dot{m}(U_1 V_{\theta_1} - U_2 V_{\theta_2}) \quad (2.25)$$

It can be seen that if V_θ and U are in the same direction, then V_θ is positive and the torque exerted by the shaft on the rotor is positive, as well as $\dot{W}_{shaft}$ since shaft torque and w are in the same direction - case of pumps. If the opposite case occurs, the machine is a turbine, since fluid does work on the rotor and the rotor rotates in the opposite direction.

3 CENTRIFUGAL PUMP

3.1 Theoretical Considerations

The centrifugal pump is one of the most common radial-flow turbomachines. It is composed by two main components: an impeller, which consists of a set of blades or vanes, usually curved, arranged in a regular pattern around a rotating shaft, and a stationary casing that surrounds it, also called volute, as illustrated in Figure 3.1.

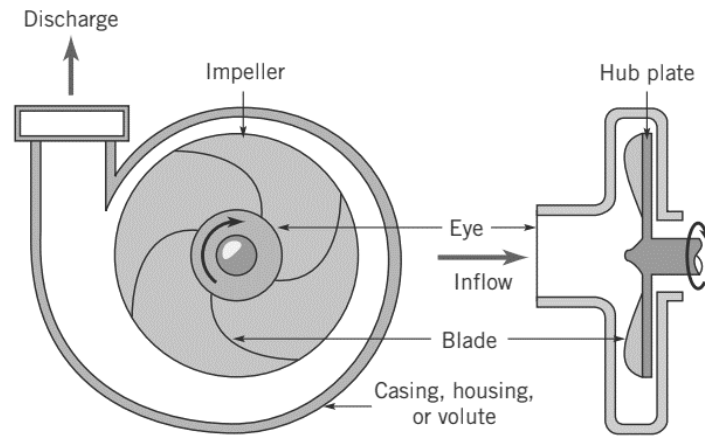


Figure 3.1: Schematic diagram of basic elements of a centrifugal pump (Munson et al., 2013).

The fluid enters the machine axially through the eye of the casing and it is deflected by 90° to be admitted to the impeller, flowing radially outward and increasing both pressure and velocity. At the impeller outlet, it is forced to pass through the volute, inside of which it describes a spiral trajectory, with a gradually increasing cross-section area in the direction of flow. The kinetic energy gained by the fluid in the impeller is here partially converted into static pressure (Oliveira & Lopes, 2020; Munson et al., 2013).

If the flow at the inlet and outlet of the impeller is observed in detail, it is possible to draw some conclusions about the performance of the pump during its operation. Figure 3.2 shows the relationship between the various velocities for a typical blade passage: the absolute velocity of the fluid at the inlet of the impeller, $\vec{V}_1$, is the vector sum of the velocity of the blade, $\vec{U}_1$, and the relative velocity, $\vec{W}_1$, so that $\vec{V}_1 = \vec{W}_1 + \vec{U}_1$ and at the exit $\vec{V}_2 = \vec{W}_2 + \vec{U}_2$.

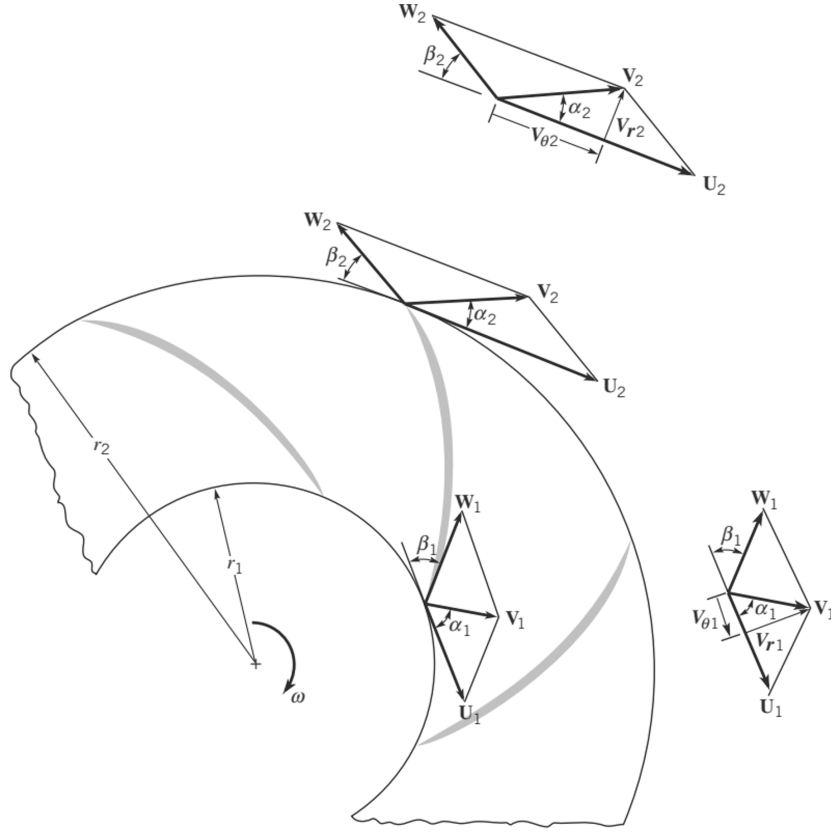


Figure 3.2: Velocity diagrams at the inlet and exit of a centrifugal pump impeller (Munson et al., 2013).

Considering the tangential components of absolute velocity, $V_{\theta 1}$ and $V_{\theta 2}$, the shaft torque required to rotate the pump impeller, given by equation (2.24), is obtained.

From section (2.2),

$$w_{shaft} = \left(\frac{p_{out}}{\rho} + \frac{v_{out}^2}{2} + gz_{out} \right) - \left(\frac{p_{in}}{\rho} + \frac{v_{in}^2}{2} + gz_{in} \right) + loss \quad (3.1)$$

where $w_{shaft} = \dot{W}_{shaft} / \rho Q$, with Q being the volumetric flow rate. Combining this with equation (2.24) per unit mass of flowing fluid and dividing both sides by g

$$\frac{(U_2 V_{\theta 2} - U_1 V_{\theta 1})}{g} = H_{out} - H_{in} + h_f \quad (3.2)$$

where H and h_f are total head and head loss, respectively, as defined earlier in section (2.2).

It is possible to see that the head loss, h_f , reduces the actual head rise, $h_a = H_{out} - H_{in}$, achieved by the fluid, being always less than the ideal head rise, h_i

$$h_i = \frac{(U_2 V_{\theta 2} - U_1 V_{\theta 1})}{g} \quad (3.3)$$

Under normal conditions, the amount of angular momentum carried by the fluid as it enters the rotor is approximately zero, and thus

$$h_i \approx \frac{U_2 V_{\theta_2}}{g} \quad (3.4)$$

After some algebraic manipulation,

$$h_i \approx c_1 - c_2 \frac{Q}{tg\beta_2} \quad (3.5)$$

where c_1 and c_2 are constants. Equation (3.5) shows that the ideal or maximum head rise varies linearly with Q for a given rotation and blade orientation, as illustrated in Figure 3.3. Pumps with blades with $\beta_2 > 90^\circ$ are called forward curved, whereas blades with $\beta_2 < 90^\circ$ are called backward curved, with a normal range of $20^\circ < \beta_2 < 25^\circ$. Since it is not desirable for the $H(Q)$ curve to exhibit a maximum value within the range of Q variation, due to the creation of instabilities in its operating point, pumps are usually designed with backwards curved blades (Munson et al., 2013).

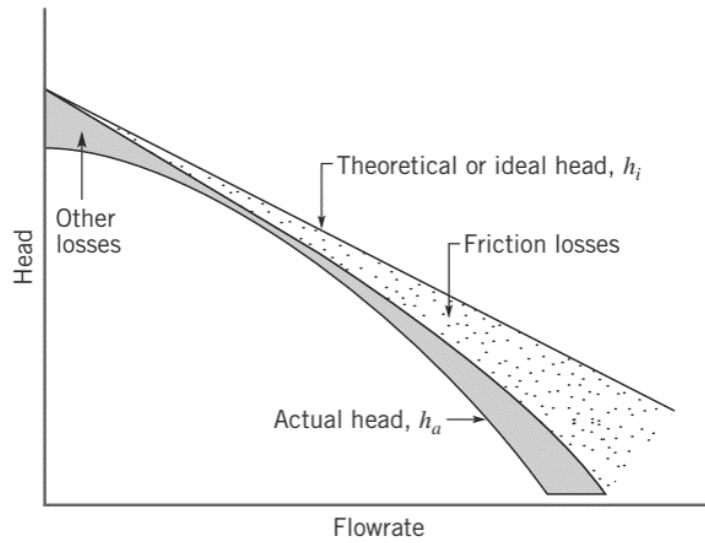


Figure 3.3: Effect of losses on the pump head-flow rate curve (Munson et al., 2013).

In Figure 3.3 it is possible to observe the degradation of the head provided by the pump, from ideal to the actual case. The differences between curves are due to: at the outlet, the flow is not perfectly guided by the blades (slip), since they are not infinite in number; to the existence of losses due to friction, which vary with Q^2 ; impeller blade-casing leakage flows or the relative velocity at the inlet not being tangent to the blades, increasing with the increase in the deviation between the angle of attack and the inlet angle of the blade, β_1 . In fact, for a given blade design, at a given rotation speed, there will be only one flow rate value for which this condition is met, known as the nominal flow rate, resulting in a maximum efficiency (Munson et al., 2013; Viegas, 1989).

3.2 Cavitation and Net Positive Suction Head (NPSH)

Cavitation is caused by the formation and collapse of vapor bubbles when the static pressure of a liquid drops below its vapour pressure, leading to the formation of small vapor-filled cavities in the liquid. When the bubble moves with the flow to an area of higher pressure, water particles rush into the empty cavities, colliding with each other, resulting in extremely high local pressures (in some cases, on the order of 1 GPa, according to Brennen (2013)). If this collapse occurs near solid surfaces, material erosion can be so severe that it may lead to its destruction, as illustrated in Figures 3.4 and 3.5.

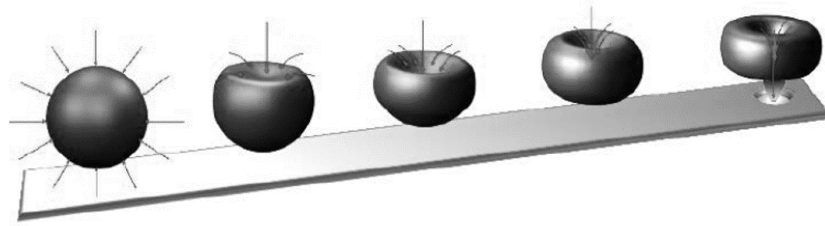


Figure 3.4: Cavitation: implosion of vapor bubbles (Holdhof et al., 2019).

In centrifugal pumps, this phenomenon typically occurs near the leading edge of the impeller blade, where not only is there an increase in fluid velocity, but also separations caused by fluid shock at the leading edge, leading to further reductions in fluid pressure.



Figure 3.5: Destructive effects of cavitation on a pump impeller.

In addition to causing structural damage to the pump, with characteristic noise and vibrations, cavitation reduces the flow rate and head provided by the pump, significantly reducing performance, as illustrated in Figure 3.6.

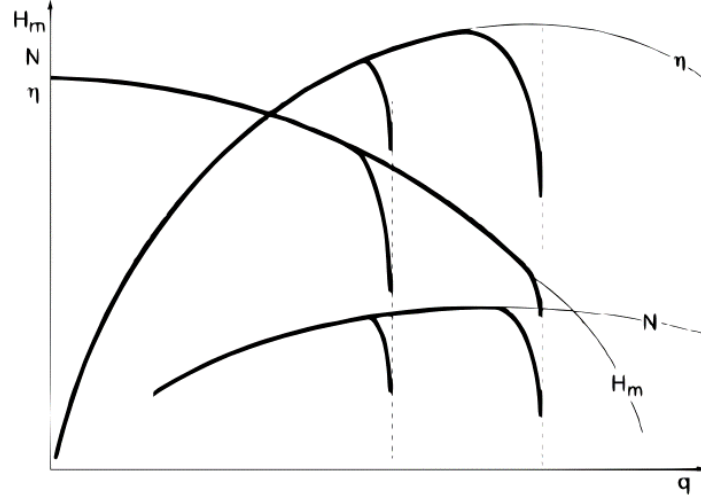


Figure 3.6: Cavitation: effect on pump performance characteristics (Ramírez, 2022).

To ensure that cavitation does not occur, it must be guaranteed that, in the area susceptible to cavitation (immediate vicinity of the impeller, on the suction side in the case of pumps), the absolute stagnation pressure always remains higher than the vapor pressure of the fluid for a given operating temperature. If defined in terms of head, the safety threshold, given by the difference between these values, is called the Net Positive Suction Head (NPSH),

$$NPSH = \left(\frac{p_{in}}{\rho g} + \frac{v_{in}^2}{2g} \right) - \frac{p_v}{\rho g} \quad (3.6)$$

In fact, it is necessary to distinguish between the required NPSH ($NPSH_R$) and the available NPSH ($NPSH_A$). The $NPSH_R$ is defined by the manufacturer and reflects the value that must be maintained or exceeded so that cavitation does not occur. This value does not depend on the installation in which the pump is installed and is usually determined experimentally by the manufacturer. $NPSH_A$ represents the actual head that occurs for the particular flow system where the pump is installed. Therefore, it must be ensured that once the pump is installed, under no circumstances $NPSH_A < NPSH_R$.

With the appropriate experimental conditions, both $NPSH_R$ and $NPSH_A$ values can be determined experimentally or, in the case of $NPSH_A$, analytically if the system parameters are known by using equation 3.8. In the present study, experiments aimed at determining these values will be discussed in Chapter 4, using appropriate experimental setups. A typical flow system is illustrated in Figure 3.7.

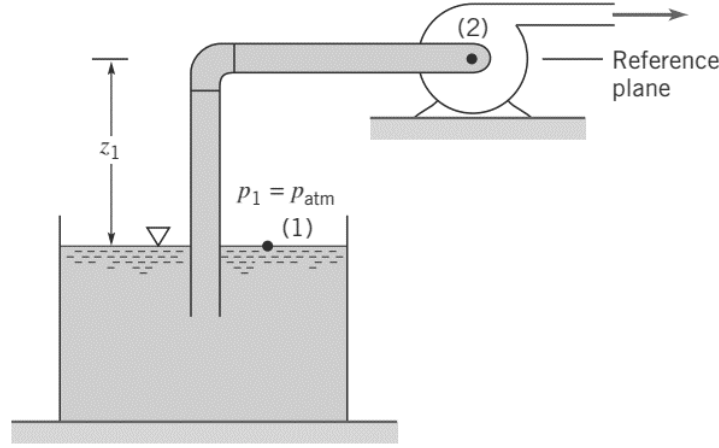


Figure 3.7: Schematic of a pump installation (Munson et al., 2013).

If an energy balance is applied between the free liquid surface and a point on the suction side of the pump near the impeller inlet

$$\frac{p_{atm}}{\rho g} - z_1 = \frac{p_{in}}{\rho g} + \frac{v_{in}^2}{2g} + \sum h_f \quad (3.7)$$

where $\sum h_f$ represents the head losses between the reservoir and the pump inlet. Thus,

$$NPSH_A = \frac{p_{atm}}{\rho g} - z_1 - \sum h_f - \frac{p_v}{\rho g} \quad (3.8)$$

Some conclusions can be made from equation (3.8). For the system considered, as the height, z_1 , between the free surface of the water and the pump inlet increases, the $NPSH_A$ decreases, therefore imposing a critical value for z_1 above which pump starts cavitating. In some cases, the requirement $NPSH_A > NPSH_R$ can only be met if the pump is installed at a lower level than the reservoir, with the gravitational term then constituting a positive contribution to the value of $NPSH_A$.

3.3 Pump Performance and System Characteristics

Since a pump is not supposed to operate alone, but rather as an integral part of a system, it is necessary to understand its performance under different operating conditions, especially regarding variations in the volumetric flow rate to be transported.

3.3.1 Pump characteristic curves

The performance of a pump is determined experimentally and presented as pump performance curves. Usually, performance characteristics are provided for a given diameter impeller and operating speed in the form of plots of H , $\dot{W}_{shaft}$, η , and $NPSH_R$ versus Q , as illustrated in Figure 3.8.

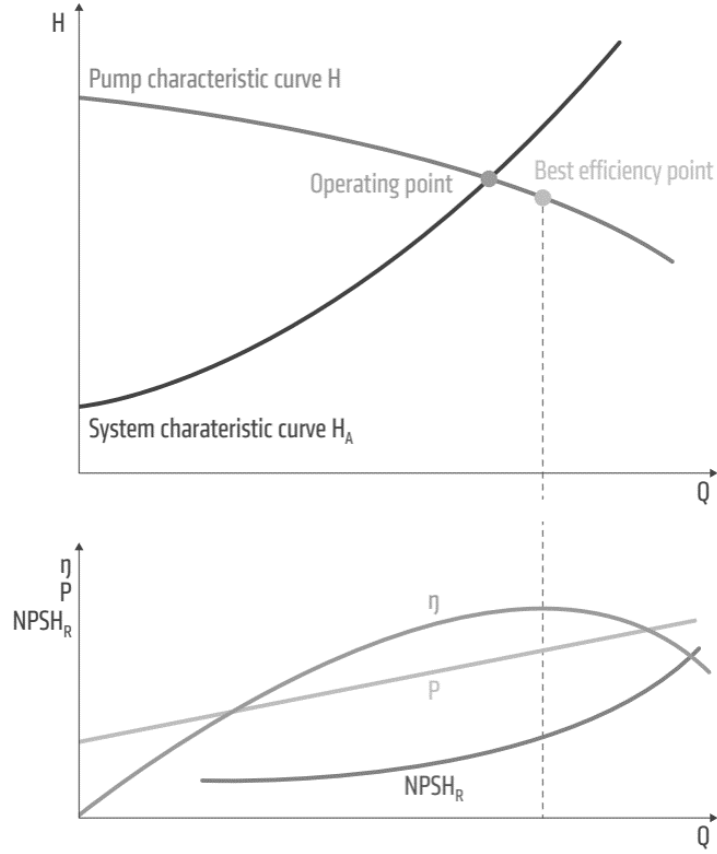


Figure 3.8: Typical performance characteristic curves for a centrifugal pump of a given size operating at a constant impeller speed (Holdhof et al., 2019).

The actual head rise, h_a , referred in section 3.1, and referred to as H from now on, can be experimentally determined by simply measuring the pressure difference between the inlet and outlet of the pump, since typically the differences between elevation and velocity are small. Thus,

$$H \simeq \frac{p_{out} - p_{in}}{\rho g} \quad (3.9)$$

The hydraulic power, P_h , gained by the fluid is given by

$$P_h = \rho Q E_s = \rho Q g H \quad (3.10)$$

and by measuring the torque and the rotation speed, the shaft power driving the pump, $\dot{W}_{shaft}$, can be determined by

$$\dot{W}_{shaft} = T_{shaft} \cdot \omega \quad (3.11)$$

The overall efficiency of the pump, η_o , defined in section 2.3, is then

$$\eta_o = \frac{\text{power gained by the fluid}}{\text{shaft power driving the pump}} = \frac{P_h}{\dot{W}_{shaft}} \quad (3.12)$$

As referred in section 2.3, η_o is affected by hydraulic losses in the pump, as seen in section 3.1, by the mechanical losses in bearings and seals and, in some cases, by leakage of fluid between the impeller and the casing (volumetric loss). Thus, the overall efficiency of the pump comes

$$\eta_o = \eta_h \eta_m \eta_v \quad (3.13)$$

A typical experimental of the arrangement for determining the different performance curves will be detailed in Chapter 4, with special attention to the power and efficiency curves, as often what is measured is the power absorbed by the electric motor driving the pump, which is more interesting from the perspective of the consumer, as it provides an insight into the energy consumption of the motor-pump unit.

3.3.2 System characteristic curve and operating point

As seen in section 3.1, there is only one point for which the efficiency is maximum, as illustrated in Figure 3.8. This point is denoted as the Best Efficiency Point (BEP) and when selecting a pump is desirable for the pump to operate near this region.

In a system where a pump is installed to transport water from a level z_1 to z_2 , at pressures p_1 and p_2 , the actual head gained by the fluid from the pump is

$$H = \frac{p_2 - p_1}{\rho g} + (z_2 - z_1) + \sum h_f \quad (3.14)$$

where $\sum h_f$ are all friction losses in pipe, fittings and valves. Since in majority of cases the flow is turbulent, $h_f \propto Q^2$ and thus

$$H = \frac{p_2 - p_1}{\rho g} + (z_2 - z_1) + KQ^2 \quad (3.15)$$

with the value of K depending on pipes sizes and lengths, friction factors and minor loss coefficients that the fluid has to overcome. Equation (3.15) is known as system equation and relates both the head gained by the fluid and the system parameters, as illustrated in Figure 3.8. The intersection of the pump performance curve and system curve gives the operating point, which can vary for several reasons, requiring for control and adjustment of pump performance.

3.3.3 Control and adjustment of pump performance

According to Holdhof et al. (2019), different approaches can be adopted for the control of pump systems, some of which are illustrated in Figure 3.9.

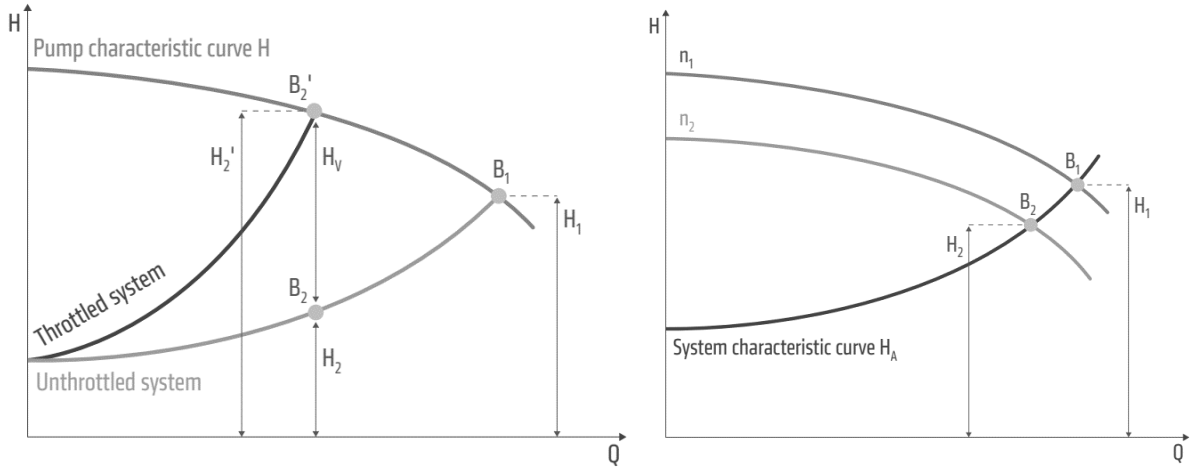


Figure 3.9: Throttle (left) and speed (right) control (Holdhof et al., 2019).

With a throttle control the desired flow rate is achieved via a throttling device in the pressure line of the system which consists on the regulation of the loss of head, leading to higher power consumption in operating in point B_2' , comparing to B_1 . The advantages of this type of control are mainly the simplicity and the low cost of acquisition.

With speed control it is the pump curve that is changed by varying the drive speed of the impeller, enabling for passing from point B_1 to B_2 (figure on the right) with lower power consumption. In the present study, experiments aimed to determine the influence of speed control in the pump performance characteristic curves will be discussed in Chapter 4, using appropriate experimental setups.

3.3.4 Association of pumps

Given that the present work also discusses the association of pumps in series and parallel (sections 4.2.5 and 4.3.3), used to provide additional head or flow capacity, a brief overview of these types of arrangements is also addressed.

It has been shown that a pump cannot be efficiently operated in every operating point due to its efficiency curve. When there are fluctuations of head and flow rate, an individual pump may not be sufficient to met the requirements. This difficulty might be overcome by means of parallel and series connection of several pumps (Holdhof et al., 2019).

Figure 3.10 illustrate the parallel use of several identical (left) or different (right) pumps when the desired flow rate cannot be achieved using a single pump; the system request variable flow or redundancy is desirable in order to increase reliability. It can be seen from Figure 3.10 (left), that for two identical pumps, the combination results in more flow rate (and head), when comparing to a single pump (from point B_1 to B_2).

Even though the addition of two pumps does not double the flow rate, since the pressure losses increase with the flow rate, both pumps overcome the same pressure difference or head. For relatively flat system curves, a significant increase in flow rate can be achieved, hence the advisable use of pumps in parallel for larger piping systems.

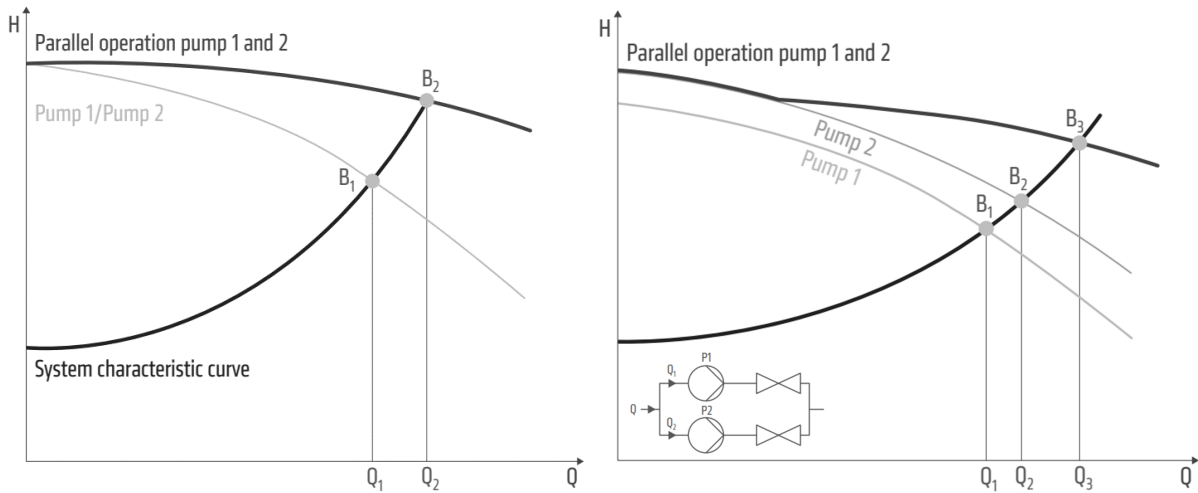


Figure 3.10: Parallel connection of identical (left) and different (right) pumps (Holdhof et al., 2019).

At the right side of Figure 3.10 it can be seen that with two different pumps, pump 1 can only be switched on and contribute to the parallel coupling curve when the head is no longer above the head that pump 1 is capable of supplying. Until then, only pump 2 works and to prevent the fluid from flowing back through pump 1, non-return valves must be installed, as shown in the schematic representation.

Figure 3.11 illustrate the series use of several identical (left) or different (right) pumps when the desired head cannot be achieved using a single pump.

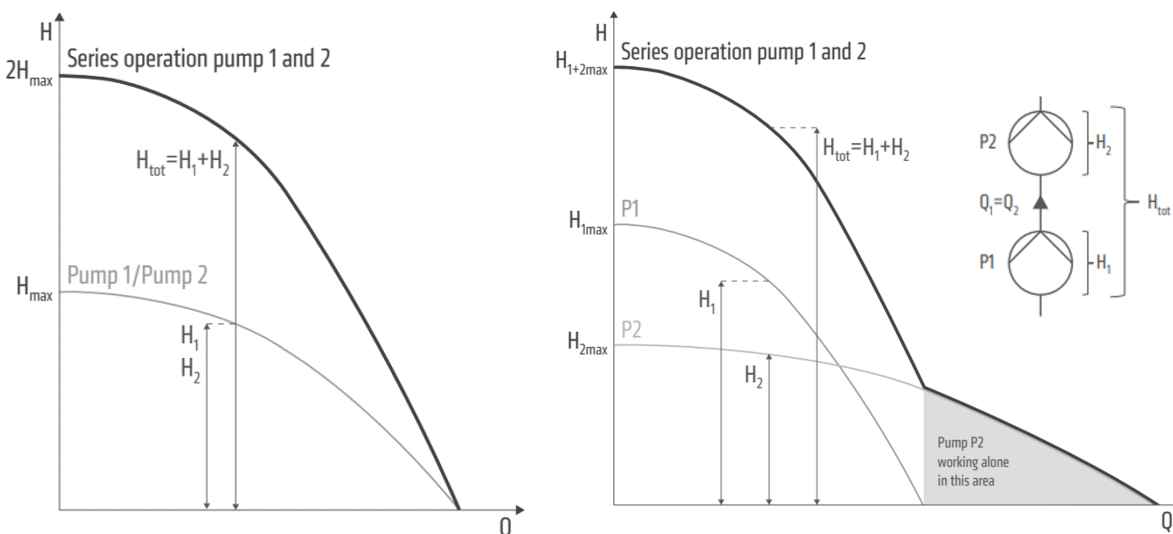


Figure 3.11: Series connection of identical (left) and different (right) pumps (Holdhof et al., 2019).

In this case, the resulting characteristic curve of the pump system is achieved by adding the heads for relevant flow rates. In the case of identical pumps (left), the head that can be achieved is twice as large as the corresponding head of the individual pumps. In the case of use of two different pumps, pump 1 can only be switched on and contribute to the series coupling curve, by supplying more head, until it reaches the maximum flow it can deliver.

When using two different pumps, pump 1 can only be switched on and contribute to the series coupling curve by supplying more head until it reaches the maximum flow it can deliver. From this point on, only pump 2 should work, as shown by the surface colored. Usually, a combination of a pump with a fixed speed and a downstream pump with controlled speed is used.

3.4 Dimensionless Parameters and Similarity Laws

Introduced the dependent variables of practical importance in the previous chapter, H , $\dot{W}_{shaft}$ and η , it is expected that these will depend on other parameters that can be grouped into three groups: geometric variables, such as some characteristic diameter, D , other pertinent length, l , and surface roughness, ϵ ; control variables, flow rate, Q , and shaft rotational speed, w and fluid property variables, such as fluid viscosity, μ , and fluid density, ρ , according to Munson et al. (2013) and Viegas (1989).

The dependent variables referred above can be expressed as

$$\text{dependent variable} = f(D, l, \epsilon, Q, w, \mu, \rho) \quad (3.16)$$

Based on a dimensional analysis using the Buckingham pi theorem (not detailed here for simplicity) that states that *if an equation involving k variables is dimensionally homogeneous, it can be reduced to a relationship among $k - r$ independent dimensionless products, where r is the minimum number of reference dimensions required to describe the variables*, it is possible to rewrite (Munson et al., 2013)

$$\text{dimensionless product} = \phi \left(\frac{l}{D}, \frac{\epsilon}{D}, \frac{Q}{wD^3}, \frac{\rho w D^2}{\mu} \right) \quad (3.17)$$

Since the flow in pumps is normally turbulent, with inertial forces predominating over viscous forces; the geometrical form of the pump chamber is the dominant geometric factor rather than the surface roughness and considering geometrically similar pumps where all dimensions, l , can be scaled up or down by a common length scale, the performance characteristics can now be expressed in terms of dimensionless parameters as

$$\text{head coefficient } (C_H) = \frac{gH}{w^2 D^2} = \phi_1 \left(\frac{Q}{wD^3} \right) \quad (3.18)$$

$$\text{power coefficient } (C_P) = \frac{\dot{W}_{shaft}}{\rho w^3 D^5} = \phi_2 \left(\frac{Q}{w D^3} \right) \quad (3.19)$$

$$\text{efficiency } (\eta) = \frac{\rho g Q H}{\dot{W}_{shaft}} = \phi_3 \left(\frac{Q}{w D^3} \right) \quad (3.20)$$

where $C_Q = Q/wD^3$ is the flow coefficient. Two geometrically similar pumps will be operating under similar conditions if a certain dimensionless coefficient has the same value for both. It is then said that the machines are operating at equivalent points and all other coefficients will be equal due to the relationships that exist between them

$$\left(\frac{Q}{w D^3} \right)_1 = \left(\frac{Q}{w D^3} \right)_2 \quad (3.21)$$

$$\left(\frac{g H}{w^2 D^2} \right)_1 = \left(\frac{g H}{w^2 D^2} \right)_2 \quad (3.22)$$

$$\left(\frac{\dot{W}_{shaft}}{\rho w^3 D^5} \right)_1 = \left(\frac{\dot{W}_{shaft}}{\rho w^3 D^5} \right)_2 \quad (3.23)$$

$$\eta_1 = \eta_2 \quad (3.24)$$

These relationships are usually referred to as scaling laws and are used to predict, after experimentally determining the performance characteristics of one pump in a laboratory, the characteristics for other pumps of the same family under different operating conditions or even for the pump from which the curves were obtained if necessary to assess the performance for different rotation speeds, as will be discussed later.

4 PERFORMANCE TESTING OF CENTRIFUGAL PUMPS

4.1 Experimental setup

4.1.1 ISEC experimental setup

The experimental setup for centrifugal pump testing at ISEC is shown in Figure 4.1, including the corresponding instrumentation.

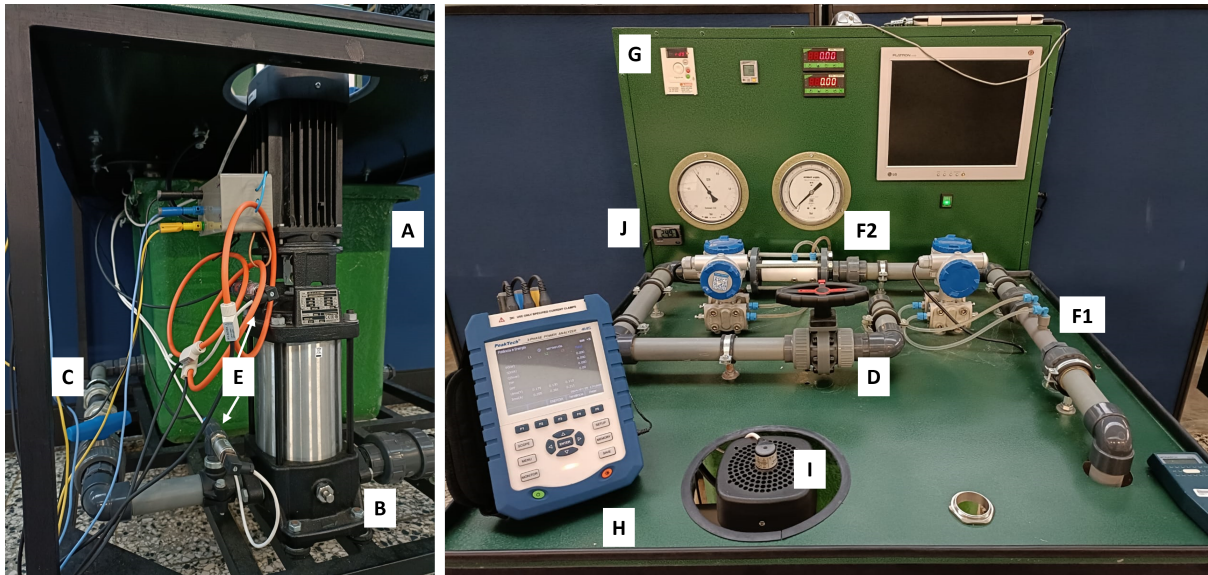


Figure 4.1: Centrifugal pump test bench and instrumentation at the Laboratory of Hydraulic Machines (LHM) of ISEC.

The test bench for centrifugal pump testing at LHM consists of a centrifugal pump from Grundfos, type CR5-8 A-A-A-E-HQQE, model A96542008 P31926 with nominal conditions presented in Table 4.1.

Table 4.1: Nominal conditions of the centrifugal pump CR5-8 at the LHM of ISEC

f	50	Hz	P_2	1.1	kW
n	2853	rpm	$H_{mx.}$	56.5	m
Q	5.8	m ³ /h	H	40.3	m
$p_{max.}/t_{max.} = 16/120 \text{ bar}/^{\circ}\text{C}$					
$\eta_p = 62.1\%$					

The test bench is composed of the following components:

- water reservoir (A);

- centrifugal pump (Grundfos CR5-8) (B);
- regulation valve on the suction side (C);
- regulation valve on the compression side (D);
- pressure transducers on the suction and compression sides (E);
- *multiPitot* or multiple orifice flow meter (F1);
- *venturi*-type flow meter (F2);
- Variable-Frequency Drive (VFD) (G);
- 3-phase power analyser (PeakTech 4145) (H);
- rotational speed meter (I);
- water temperature sensor (J).

The different signals from the various measurement instruments are connected to data acquisition boards, and are subsequently processed and displayed on the screen present on the test bench using *LabView* software.

4.1.2 UHM experimental setup

At UMH, the experimental setup for centrifugal pump testing of individual pumps is shown in Figure 4.2, including the corresponding instrumentation.

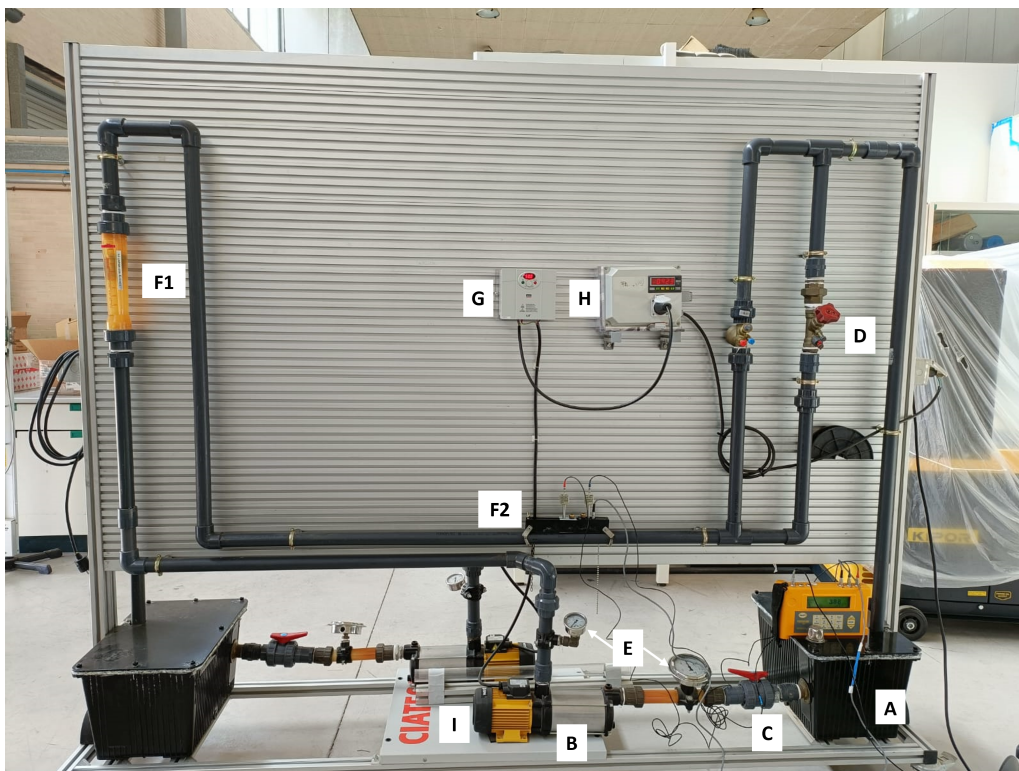


Figure 4.2: Centrifugal pump test bench for individual pump testing, including instrumentation, at the LHM of UMH.

The test bench for centrifugal pump testing of individual pumps is composed of the following components:

- water reservoir (A);
- centrifugal pump (ESPA Prisma 15 5) (B);
- regulation valve on the suction side (C);
- regulation valve on the compression side (D);
- pressure gauges on the suction and compression sides (E);
- *rotameter*-type or variable area flow meter (F1);
- ultrasonic flow meter (F2);
- variable-frequency drive (VFD) (G);
- power analyser (H);
- rotational speed measurement (manual) (I).

The experimental setup for centrifugal pump testing of series and parallel configurations is shown in Figure 4.3, including the corresponding instrumentation.



Figure 4.3: Centrifugal pump test bench for series and parallel configurations, including instrumentation, at the LHM of UMH.

The test bench for centrifugal pump testing of series and parallel configurations is composed of the following components:

- two water tanks treated with an algae growth inhibitor (A);
- two identical centrifugal pumps, (ESPA Prisma 15 5) (B1 and B2);
- regulation valve on the suction side (C);
- regulation valve on the compression side (D);
- two pressure gauges on the suction side (E);
- two pressure gauges on the compression side (G)
- magnetic flow meter (G);
- electrical panel with VFD and power analyser (H).

The test benches for centrifugal pump testing at UMH consists of a centrifugal pump from ESPA, model Prisma 15 5, with nominal conditions presented in Table 4.2.

Table 4.2: Nominal conditions of the centrifugal pump ESPA Prisma 15 5, at the LHM of UMH

f	50	Hz	P_2	0.8	kW
n	2853	rpm	$H_{mx.}$	53.9	m
Q	2.3	m ³ /h	H	35.1	m
$p_{max.suction}/t_{max.} = 0.2/40 \text{ bar/}^{\circ}\text{C}$					
$\eta_p = 36.2\%$					

4.2 Experimental procedure

The test methodology to be followed varies slightly depending on the type of curves and information to be obtained, as well as the equipment available at each institution.

As mentioned previously, the performance of a pump is determined experimentally by pump manufacturers before they are placed on the market, being then presented as pump performance curves, in the form of plots of H , $\dot{W}_{shaft}$, η and $NPSH$ versus Q . Respectively, the equations 3.9, 3.11, 3.12 and 3.6 allow for the determination of the aforementioned curves. Typical arrangements of a test bench for individual pump testing are shown in Figure 4.1 and Figure 4.2. In the case of pump association in series or parallel, it is important to understand the influence of the two configurations on the head and flow rate provided to the system. The typical arrangement of a test bench for testing pump associations is shown in Figure 4.3.

Table 4.3 summarizes the types of tests conducted in each of the facilities to obtain the desired curves.

Table 4.3: Types of centrifugal pump tests conducted at each facility, ISEC and UMH

Type of test	Curves obtained	ISEC	UMH
Individual testing	$H(Q)$	✓	✓
	$\dot{W}(Q)$	✓	✓
	$\eta(Q)$	✓	✓
	cavitation $H(Q)$	✓	×
	cavitation $H(P_{in})$	✓	×
	NPSH(Q)	✓	×
Parallel testing	$H(Q)$	×	✓
Series testing	$H(Q)$	×	✓

4.2.1 Determination of the characteristic curve

It was shown previously that the intersection of the pump characteristic curve and system curve, given by Equations 3.5, 3.9 and 3.15, gives the operating point of the pump and relates both the head gained by the fluid and the system parameters. In this way, the construction of this curve is based on the simple fact that the operating points belong simultaneously to both the pump curve and the system curve, as illustrated in Figure 3.8.

In the simplest case, if there are no elevation differences to overcome or additional pressure differences, as in a didactic setup where the suction and discharge are at the same level and subject to the same pressure (in this case, to the atmospheric pressure), then the system curve of Equation 3.15 will take the form of KQ^2 and the pump will provide only the head necessary to overcome the sum of head losses caused by the pipe and all the components in the system, as well as the corresponding flow rate. These systems are known as closed systems. If the discharge valve is fully open, the head loss caused by it will be minimal, and therefore, the pump will provide the maximum flow rate to overcome the total head loss of the system. This point will be the point of maximum flow rate and minimum head loss observed in the system.

For the maximum rotational speed of the electric motor (50Hz), after verifying that the suction and discharge valves are fully open, readings are taken of the pressure on the suction and discharge sides and of the circulating flow rate. The rotational speed of the electric motor driving the pump is also measured.

The downstream valve is successively adjusted to increase the head loss of the circuit, resulting in a regular decrease in flow rate intervals. After recording the observed values again, the procedure is repeated to obtain an equal distribution of points along the expected curve, in addition to the extreme positions (valve fully open and closed). Once the test is completed for the maximum rotational speed of 50 Hz, the above steps are repeated to test the centrifugal pump at other rotational speeds (e.g., 45 Hz, 40 Hz, and 35 Hz).

4.2.2 Determination of the power curve

To determine the power curve, it is necessary first to clarify some of the different terminologies commonly associated with power. Since pumps are made of several components, according to Grundfos (2016) the power consumption of the different components is designated in the following way:

- P_1 – the power input from the mains, which means the amount of power the consumer has to pay for;
- P_2 – the power input to the pump or the power output from the motor. Often referred to as shaft power ($\dot{W}_{shaft}$);
- P_h – the hydraulic power that the pump transfers to the fluid in the form of flow and head;

Typically, to determine the power transmitted to the shaft, it is necessary to calculate the torque and rotational speed. When the test bench does not have a dynamometer or the pump structure does not allow its installation, as is the case with both tested installations, it is common to use a power measurement equipment that provides the power input to the electric motor P_1 . Therefore, knowledge of the motor efficiency is necessary in such cases in order to determine P_2 .

From the perspective of the user, it is much more practical to know P_1 since it provides information on the energy consumption of the motor-pump assembly as supplied. However, the power curves typically given by manufacturers refer to the power input to the pump or the power output from the motor (usually considered equivalent when the pump is coupled directly to the motor), and therefore for the most common pump types, the term power consumption generally refers to P_2 . Since many manufacturers provide the complete assembly consisting of the motor and pump, it is common for some of them to also provide the P_1 curve. The experimental procedure for obtaining the power curve follows the same method used for obtaining the characteristic curve and can be performed simultaneously, requiring only the additional measurement of the power using appropriate equipment.

4.2.3 Determination of the efficiency curve

Once the two aforementioned curves are known, it is possible to determine the efficiency curve, also as a function of flow rate, using equation 4.1

$$\eta = \frac{\rho g Q H}{\dot{W}_{shaft}} \quad (4.1)$$

allowing for the identification of the best efficiency point of the pump, particularly in terms of the $H-Q$ and $P-Q$ combinations. When designing and selecting a centrifugal pump, it should be chosen in order to operate near this point.

4.2.4 Determination of the NPSH curve

As previously described, the $NPSH_R$ is defined by the manufacturer and reflects the value of absolute stagnation pressure that must be maintained or exceeded at the pump inlet so that cavitation does not occur. This value does not depend on the installation in which the pump is installed and is usually determined experimentally by the manufacturer. The $NPSH_A$ represents the actual head that occurs for the particular flow system where the pump is installed. Therefore, it must be ensured that once the pump is installed, under no circumstances $NPSH_A < NPSH_R$.

According ANSI/HI 9.6.1-2017 (2017), the $NPSH_R$ is defined at the point corresponding to a 3% drop in the total head of the pump. This value was implemented in the industry as early as the 1930s, after it had appeared in published voluntary consensus standards, based on the fact that it was the lowest pressure drop that could be measured. Currently, the NPSH value of a pump is determined by testing according to ISO 9906 (2012), and the $NPSH_{3\%}$ value continues to be used as a reference, since can be well captured and is technically justifiable.

According to Furtado (2019), there are two types of setups that allow for determining the $NPSH_{3\%}$ curve: closed loop installations and open sump/open loop installations. In open loop installations, such as the experimental setups previously mentioned, the pump is supplied by a tank with a constant liquid level, and the suction pressure is controlled by a valve located upstream. After starting the pump, a suction pressure is set by adjusting the valve on the suction side, and the flow rate is varied by regulating the valve on the discharge side. Since the variation in flow rate causes a change in suction pressure, the suction valve must be adjusted simultaneously with the discharge valve to maintain a constant suction pressure as the flow rate is varied. The pump follows its characteristic curve until it reaches a point where it cannot maintain the set suction pressure, deviating from the initial curve and dropping sharply. The flow rate at that point is registered and by knowing the suction pressure the $NPSH_R$ value is calculated using equation (3.6).

4.2.5 Association of pumps

It was shown previously in section 3.3.4 that pumps are often connected in series or parallel to extend some characteristics of the pump in a system.

To obtain the characteristic curves of the pump associations in series and parallel, a typical test is first performed, as detailed previously, to determine the characteristic curves at frequencies of 50, 45, and 40 Hz.

After ensuring that the suction valve is fully open, the valves in the circuit are configured to achieve a closed circuit in series, so that the fluid circulates through pump B1, then pump B2, and returns through the pipe with the flow meter, in that order, as illustrated in Figure 4.3. Next, the frequencies of pump B1 and pump B2 are set according to different combinations, e.g., those indicated in Table 4.4 for the series configuration, and the system is started. The procedure then follows the normal testing procedure for a single pump. The discharge valve, downstream of the second pump, is gradually closed to reduce the flow rate, and the readings from the suction and discharge pressure gauges (at the inlet of the first pump and the outlet of the second pump, respectively) and the circulating flow rate are taken, in order to obtain various operating points. The procedure is repeated for the other frequency combinations shown in Table 4.4.

Table 4.4: Couplings tested and frequencies used in the series and parallel pump tests

Type of association	Pump 1	Pump 2
Series	50 Hz	45 Hz
Series	50 Hz	40 Hz
Series	45 Hz	45 Hz
Parallel	50 Hz	50 Hz
Parallel	50 Hz	45 Hz
Parallel	50 Hz	40 Hz

In the case of the parallel test, the experimental procedure follows the same steps, with the only difference being the initial configuration of the valves in the circuit to achieve a closed parallel circuit so that the fluid circulates through pump B1 and pump B2 simultaneously, with a common return through the pipe with the flow meter, in that order.

4.3 Results and discussion

4.3.1 Individual pump curves

As shown in Figure 3.9, in Section 3.3.3, different approaches can be adopted for the control of pump systems, namely the throttle and speed control. As seen in the experimental procedure, the throttle control is used to obtain the curves for each speed, and the speed control is used to adjust the pump curve to a new speed. The throttle control is then used again to obtain the desired curve for the newly selected speed.

In Figure 4.4, the head versus flow rate curves at several different rotational speeds are plotted, corresponding to motor drive frequencies of 50Hz, 45Hz, 40Hz, and 35Hz. The obtained curve exhibits the expected characteristic shape, with the amount of head that the pump can produce decreasing as the flow rate increases. For 50Hz, the *shutoff head*, i.e., the head that the pump will develop when operating against a closed discharge valve amounts to 55 m, while for 35Hz amounts to 27m. For a fully open discharge valve, the pump is capable of producing a head of 17.5 meters of water column for a flow rate of approximately 8.7 m³/h (50Hz) and 8.7 meters for 6.1 m³/h (35Hz). The point of best efficiency for each rotational speed is illustrated, although these points do not correspond to equal levels of efficiency, as will be discussed further.

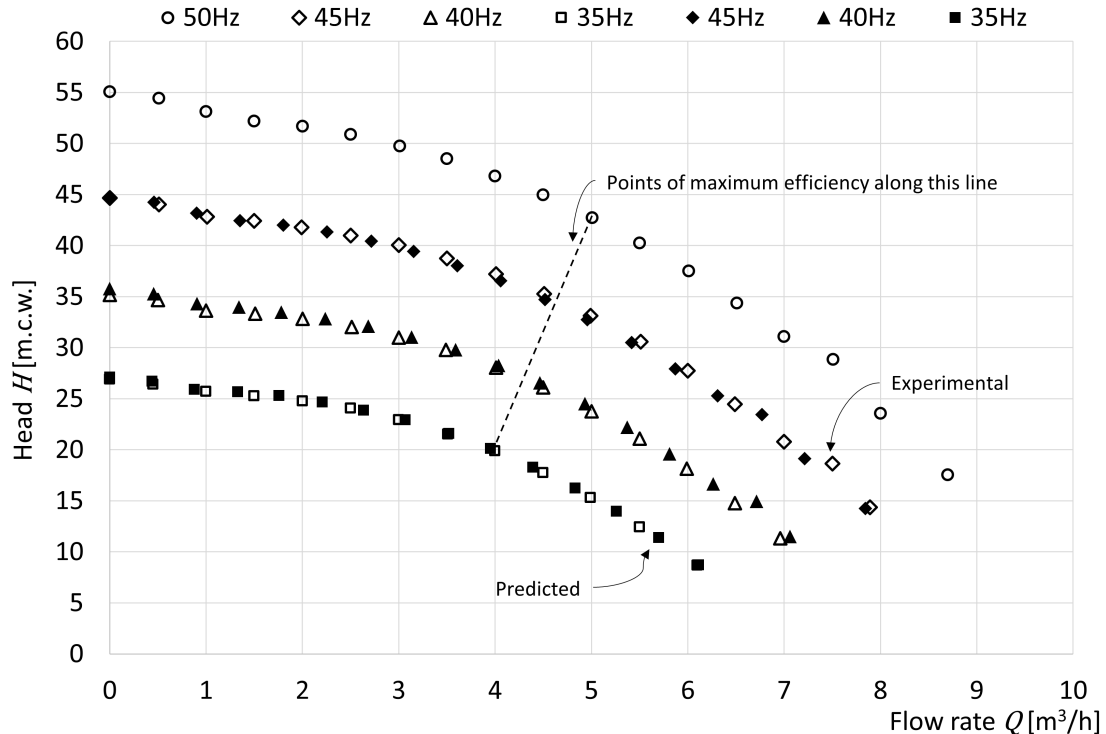


Figure 4.4: Predicted and experimental characteristic curves of the centrifugal pump CR5-8 at various speeds, for motor drive frequencies of 50Hz, 45Hz, 40Hz, and 35Hz.

From the first curve obtained at 50Hz, curves for other rotational speeds can be derived or predicted by means of the similarity laws mentioned in Section 3.4, with $Q \propto n$ and $H \propto n^2$. The predicted curves are presented in Figure 4.4 (solid points), and a good agreement between the experimentally obtained data and the predicted values can be observed. Since, in practice, the rotational speed of the pump varies along the curve, the rotational speed value adopted for calculation purposes is the one corresponding to the maximum flow rate, a criterion also followed by pump manufacturers.

Figure 4.5 shows the power versus flow rate curves at different motor drive frequencies. It is shown both the power absorbed by the electric motor, as well as the hydraulic power that the pump transfers to the fluid in the shape of flow and head.

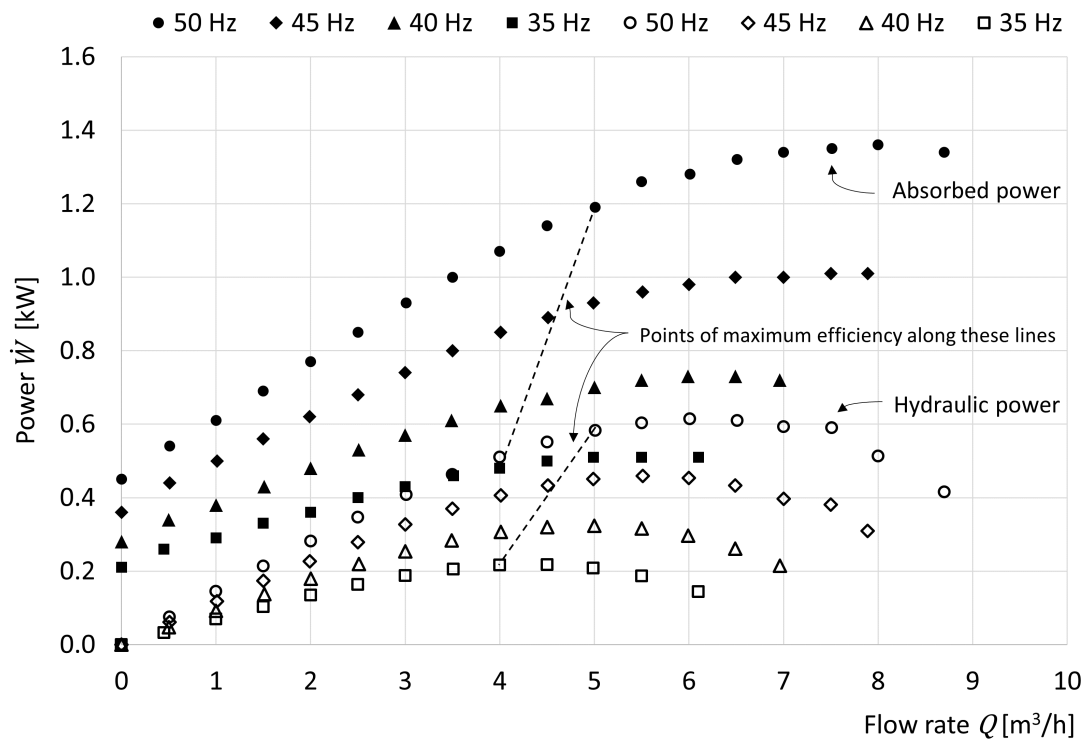


Figure 4.5: Experimental absorbed and hydraulic power curves of the centrifugal pump CR5-8 at various speeds, for motor drive frequencies of 50Hz, 45Hz, 40Hz, and 35Hz.

The obtained curves exhibit the expected characteristic shape, with the amount of power required increasing as the flow rate increases due to the increased hydraulic load imposed, registering a subsequent fall as the maximum flow rate is approached. The hydraulic power is zero for $Q = 0$ with the power supplied by the pump being used to maintain the shutoff head, and is otherwise dissipated as heat. Since the hydraulic power is proportional to both the head and the flow rate, but beyond a certain point the head begins to decrease more rapidly than the flow rate increases, the hydraulic power starts to decrease. As expected from the characteristic curves previously presented, both the absorbed power and the hydraulic power are lower with decreasing pump rotational speed.

For 50Hz, the absorbed power values for zero flow rate, the point of best efficiency, and maximum flow rate were 0.45 kW, 1.19 kW, and 1.34 kW, respectively, while the hydraulic power values were 0 kW, 0.58 kW, and 0.42 kW. The maximum values obtained ranged from 1.36 kW (50Hz) to 0.51 kW (35Hz) for absorbed power and from 0.61 kW (50Hz) to 0.22 kW (35Hz) for hydraulic power delivered to the fluid.

Figure 4.6 shows the pump efficiency versus flow rate curves at several different rotational speeds, corresponding to motor drive frequencies of 50Hz, 45Hz, 40Hz, and 35Hz.

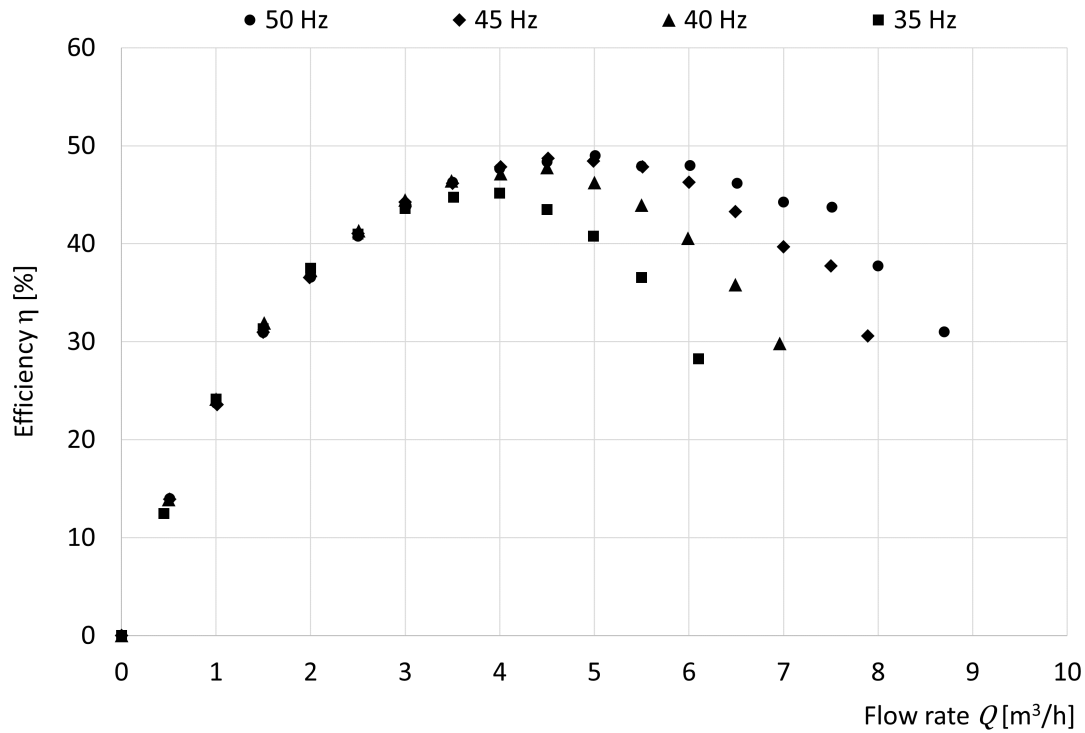


Figure 4.6: Experimental absorbed and hydraulic efficiency curves of the centrifugal pump CR5-8 at various speeds, for motor drive frequencies of 50Hz, 45Hz, 40Hz, and 35Hz.

As seen in Section 2.3, the overall efficiency is affected by the hydraulic losses in the pump, by the mechanical losses in the bearings and seals and by the volumetric losses due to internal leakage of the fluid. Some of the hydraulic losses, such as those due to fluid friction in the passages between the vanes vary approximately as Q^2 . Thus the decrease in the hydraulic power curve in Figure 4.5 and the abrupt drop in efficiency at higher flow rates are observed. It is also possible to notice that the efficiency of the pump drops off as the rotational speed is moved away from the nominal speed of 50Hz. The maximum efficiency values achieved were 49%, 48.7%, 47.8%, and 45.1% for 50Hz, 45Hz, 40Hz, and 35Hz, respectively.

As seen in Section 3.2, in addition to causing structural damage to the pump, with characteristic noise and vibrations, cavitation reduces the flow rate and head provided by the pump, significantly reducing performance. Figure 4.7 illustrates the zones beyond which the pump can no longer follow its characteristic curve (50Hz), leading to a reduction in the flow rate and head provided, for different pre-established suction pressures. The head versus flow rate curve for the 3% limit is shown in the figure with a dashed line, indicating the zone beyond which, for the referred suction pressure, the pump is considered to be in cavitation.

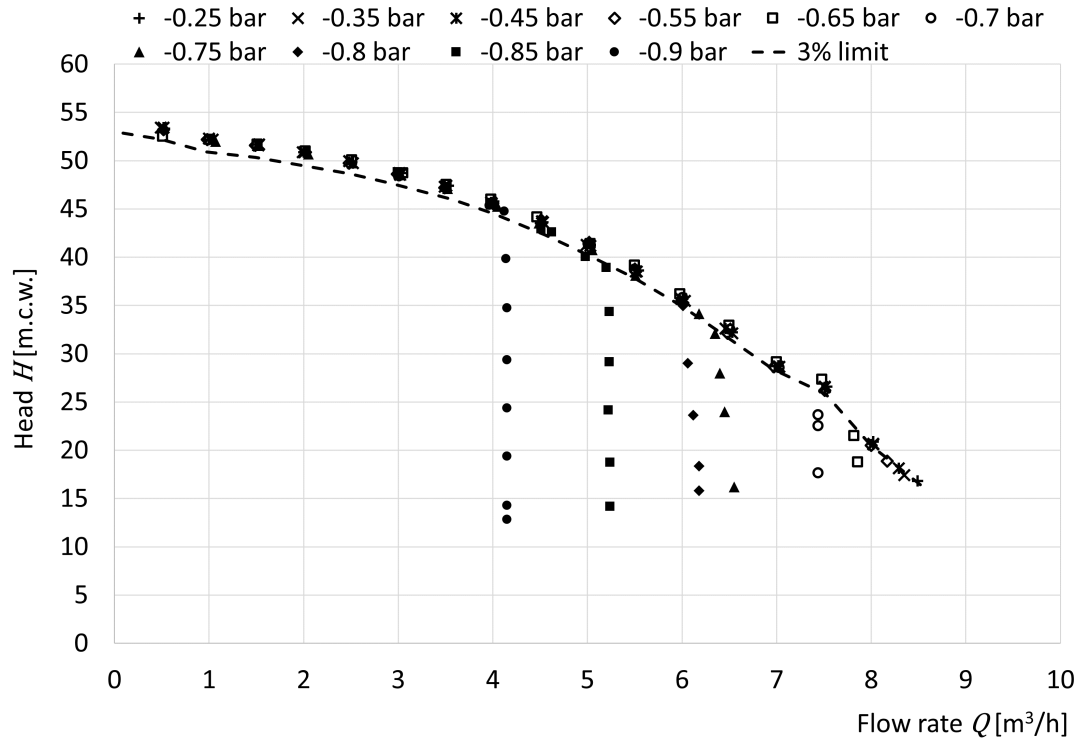


Figure 4.7: Tear-off branches for variable flow rate of the centrifugal pump CR5-8 characteristic curve at $NPSH_A < NPSH_R$, for a motor drive frequency of 50Hz.

The suction pressure value at which cavitation begins to occur is -0.65 bar, for a flow rate of approximately 7.6 m³/h. For lower suction pressure values, the pump enters cavitation more easily due to the increased fluid vaporization effect induced by greater head loss on the suction side. If the flow rate is increased at lower suction pressures, the effect is further amplified.

Figure 4.8 illustrates the different 3% drop curves for different fixed flow rates of 5, 6, 7, and 8 m³/h, as a function of suction pressure. It is observed that at higher flow rates, cavitation occurs more easily, i.e., at higher suction pressures, because high flow rates imply significant velocity gradients and, consequently, significant pressure gradients, thus amplifying the vaporization effect.

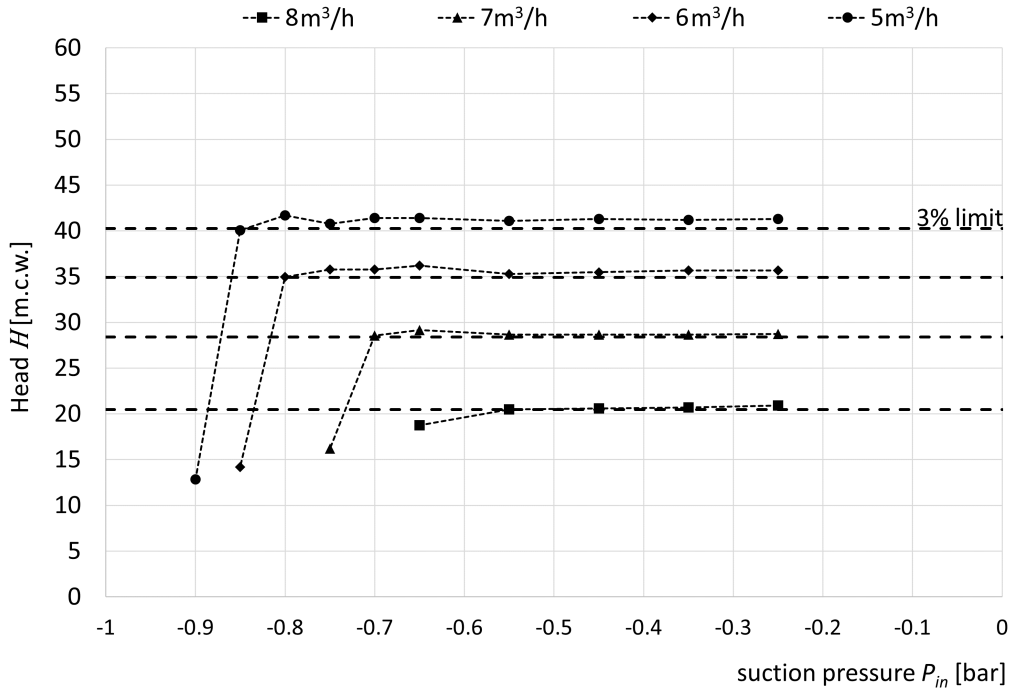


Figure 4.8: Tear-off branches for constant flow rate of the centrifugal pump CR5-8 characteristic curve at $NPSH_A < NPSH_R$, for a motor drive frequency of 50Hz.

For a given suction pressure, the $NPSH$ value is calculated for the corresponding flow rate at which there is a 3% drop in head, resulting in the $NPSH_R$ curve illustrated in Figure 4.9.

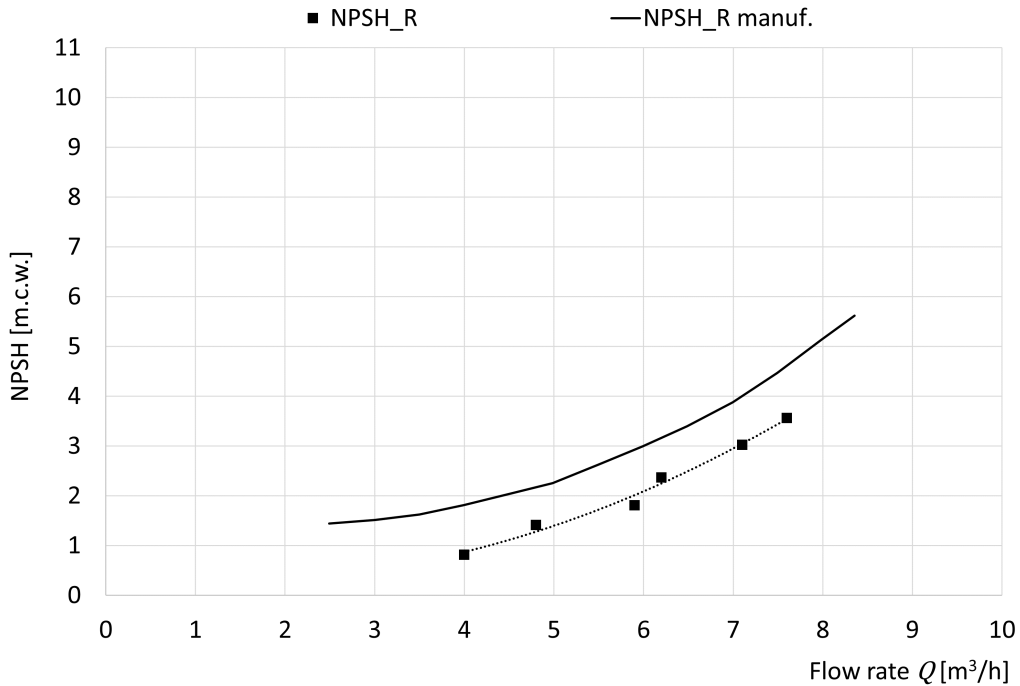


Figure 4.9: $NPSH_R$ curves obtained and specified by the manufacturer for the centrifugal pump CR5-8, at a motor drive frequency of 50Hz.

The required NPSH line provided by the manufacturer of the CR5-8 pump is also plotted with a solid line. The observed difference between the two curves is due to the fact that pump manufacturers use a safety coefficient of approximately 1 m, added to the NPSH value obtained in tests.

Figure 4.10 illustrates the CR5-8 pump $H(Q)$ curves, overlaid with curves of constant $\dot{W}$ and η , based on the data collected for motor drive frequencies of 50, 45, 40, and 35 Hz. Overall, the efficiency increases with the flow rate up to the central zone of best efficiency, regardless of the rotational speed, and then decreases with a continued increase in flow rate. For low flow rate values, the efficiency remains approximately constant as the rotational speed decreases (vertical efficiency isolines), while for higher flow rates, the efficiency starts to decrease as the rotational speed decreases (inclined efficiency isolines). The power isolines suggest that as the flow rate or the rotational speed increases, the absorbed power also increases, which had been previously observed.

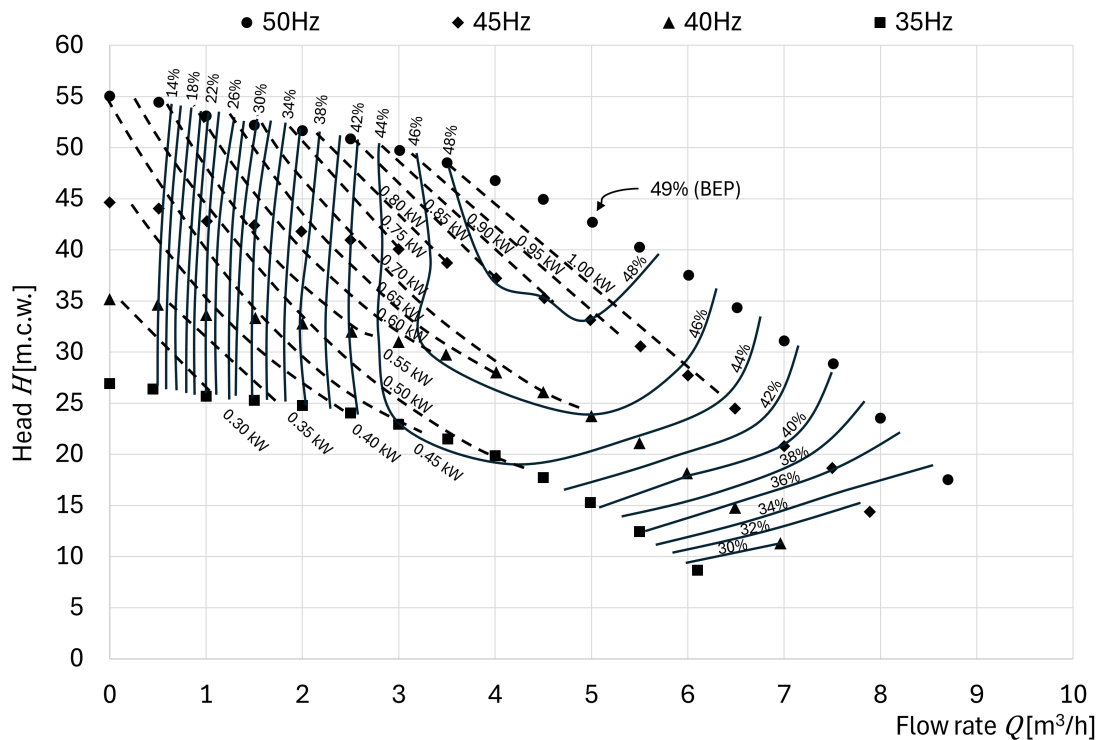


Figure 4.10: Experimental $H(Q)$ curves of the centrifugal pump CR5-8 at various speeds, for motor drive frequencies of 50Hz, 45Hz, 40Hz, and 35Hz, overlaid with curves of constant $\dot{W}$ and η .

Figure 4.11 illustrates the various curves obtained - H , $\dot{W}$, η , and $NPSH_R$ - as function of the flow rate Q , overlaid with the curves provided by Grundfos, the manufacturer of the CR5-8 pump, at a motor drive frequency of 50Hz. In general, there is a good agreement between all the curves. The curve that shows the greatest deviation from the manufacturer's curve, although not significant, is the efficiency curve, since it is determined based on the other curves, and therefore, the effect is slightly amplified.

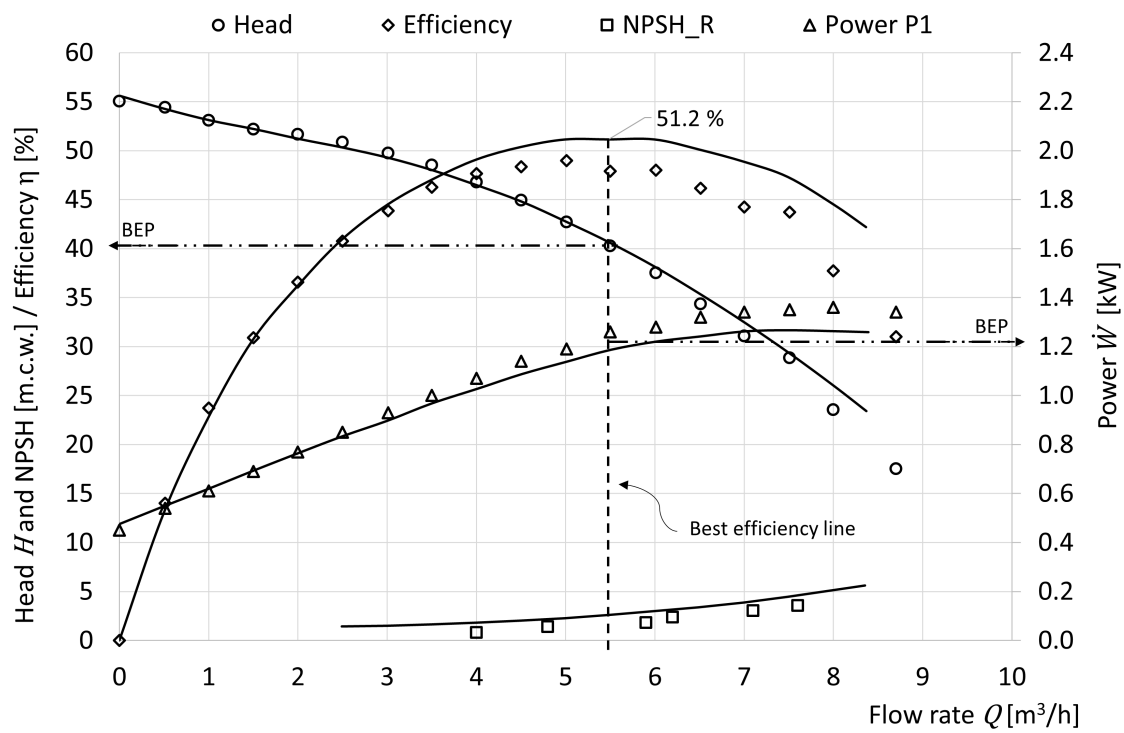


Figure 4.11: Performance curves obtained and specified by the manufacturer for the centrifugal pump CR5-8, at a motor drive frequency of 50Hz.

The performance curves presented so far can be represented in terms of dimensionless parameters, based on the dimensionless parameters presented in Section 3.4, as illustrated in Figure 4.12.

These curves condense the operating characteristics of the CR5-8 pump under various conditions, showing that the obtained curves all incorporate into a single set of non-dimensional pump performance curves. However, for the user who wants a machine for a specific purpose or who, having a specific machine, wants to know under what conditions it will operate in a given installation, the non-dimensional characteristic curves are not very practical.

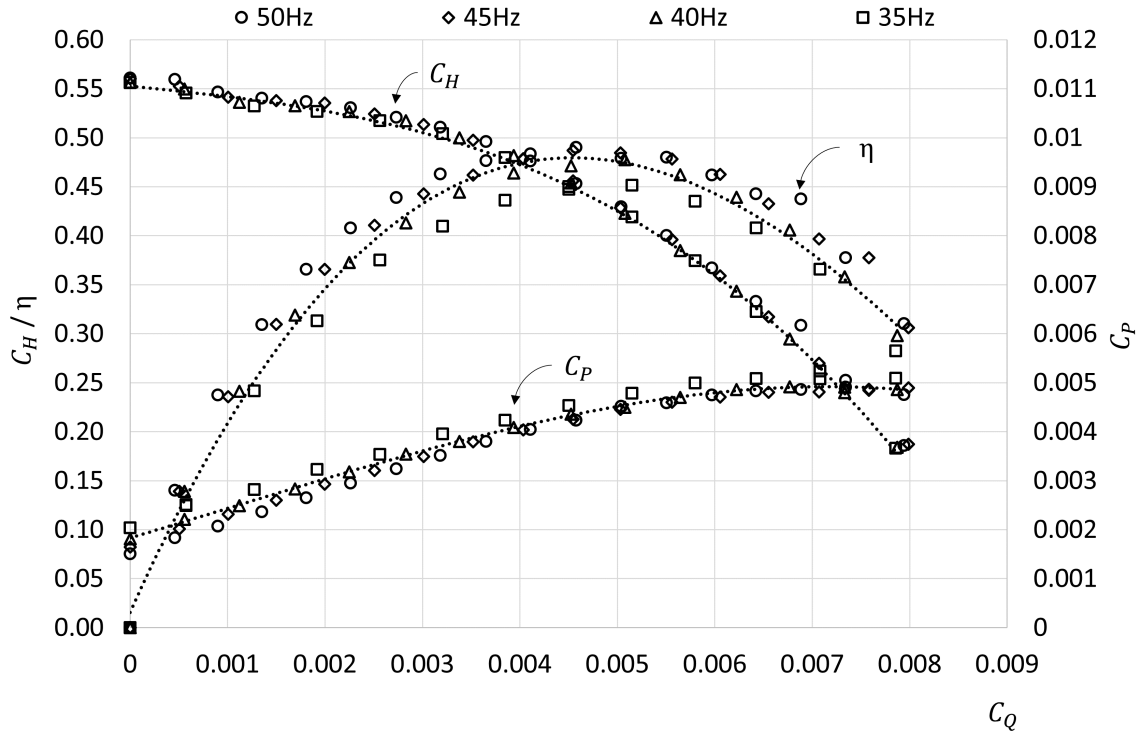


Figure 4.12: Dimensionless performance curves obtained for the CR5-8 centrifugal pump from the motor drive frequency curves of 50Hz, 45Hz, 40Hz, and 35Hz.

4.3.2 Operating behavior - control and adjustment of the pump performance

In the design, selection, and subsequent operation of a centrifugal pump, three types of questions may arise, illustrated in Figure 3.9:

Case A: Given a pump with fixed dimensions and rotational speed, the objective is for it to produce a specific head, regardless of the flow rate. In this case, it is sufficient to check the pump curve to see if it can produce the desired head, and if so, the curve will immediately provide the flow rate that the pump will deliver. For the CR5-8 pump, if a head of 25 m is required, regardless of the flow rate, the pump would be able to produce the desired head while delivering a flow rate of approximately 7.86 m³/h (Figure 4.13, at left). This would occur for a specific installation with an installation curve $H_{i(A)}$ or, in the case of an installation with a curve H_i , the point A could be obtained through throttle control.

Case B: Given a pump under the same conditions, the objective is for it to deliver a specific flow rate, without specifying the head. Similarly, in this case, it is sufficient to check if the desired flow rate is achieved by the pump, and if so, the curve will provide the head delivered by the pump. For the CR5-8 pump, if a flow rate of $3.5 \text{ m}^3/\text{h}$ is required, regardless of the head, the pump would be able to deliver the desired flow rate while providing a head of approximately 48.2 m (Figure 4.13, at left). Likewise, this would only occur for a specific installation with an installation curve $H_{i(B)}$ or, in the case of an installation with a curve H_i , the point B could be obtained through throttle control, as before.

Case C: Given a pump with specific conditions, the goal is for it to produce a specific flow rate under a specific head. Only by chance would this point C fall on the pump curve. For the CR5-8 pump, if a flow rate of $3.5 \text{ m}^3/\text{h}$ under a head of 25 m is required, it would not be possible to reproduce these conditions in any installation with curve H_i or with throttle control, as in Case A and B. Considering an installation H_i , if point C were located along this curve, it would only be necessary to decrease the rotational speed, sliding the operating point along H_i from speed n_1 to n_C , since the static head is zero, and it equals the so-called parabola of equivalent points. This would be the case for an installation of type $H_{i(C)}$. However, since the reference installation is H_i and point C is not located on its curve, adjusting the speed alone would not reproduce the desired conditions. In these cases, as illustrated on the right in Figure 4.13, it is necessary to combine throttle and speed control to obtain an installation $H_{i(C)}$ that passes through point C and then slide along $H_{i(C)}$ from n_1 to n_C , since $H_{i(C)}$ is also a parabola of equivalent points.

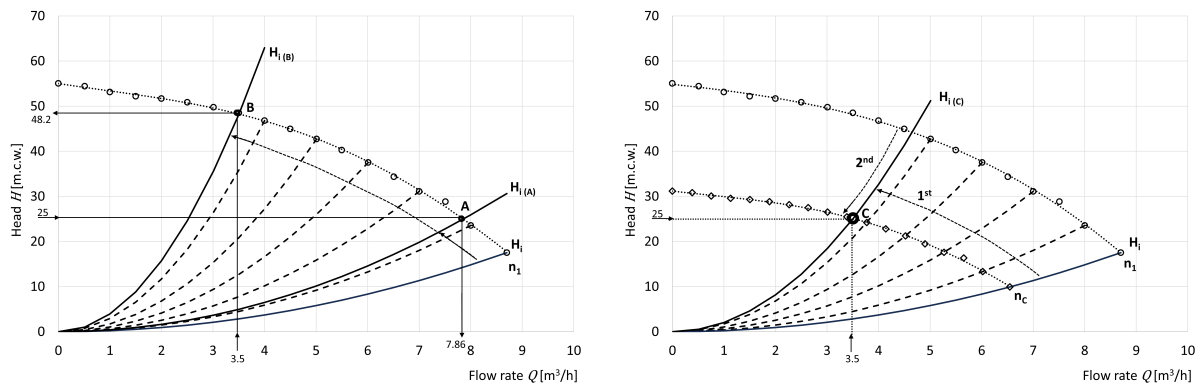


Figure 4.13: Different possible cases of centrifugal pump operation: throttle control (left) and combination of speed and throttle control (right).

4.3.3 Association of pumps

Parallel operation

Figure 4.14 shows the head versus flow rate curves for a combination of two identical pumps operating in parallel: at equal rotational speeds (50 Hz) (Figure 4.14 a)); and at different rotational speeds (P1: 50 Hz, P2: 40 Hz) (Figure 4.14 b)). The characteristic curve provided by the pump manufacturer at 50 Hz is also illustrated. As seen previously, the parallel combination would theoretically provide the sum of the flow rates of each individual pump and the same head. In the first case (Figure 4.14 a)), since the pumps are identical and operate at equal rotational speeds, even though the addition of two pumps does not double the flow rate, as pressure losses increase with the flow rate, both pumps overcome approximately the same pressure difference or head, corresponding to the head of an individual pump at 50 Hz.

Additionally, there is a slight discrepancy between the curve obtained in the individual pump test and the curve provided by the manufacturer, both at 50 Hz. Considering the curve obtained for (P1, P2), the maximum flow rate is $3.78 \text{ m}^3/\text{h}$, whereas the maximum flow rate provided by the parallel association is about $8 \text{ m}^3/\text{h}$, which would be unfeasible, due to head losses. If the curve provided by the manufacturer is considered, the maximum flow rate is about $4.3 \text{ m}^3/\text{h}$, and the pump association (P1 + P2) could result in a flow rate of $8 \text{ m}^3/\text{h}$ (less than double) as is the case. The observed difference may be due to the use of a variable area flow meter of the rotameter type in the individual pump test and a magnetic flow meter for the pump association test.

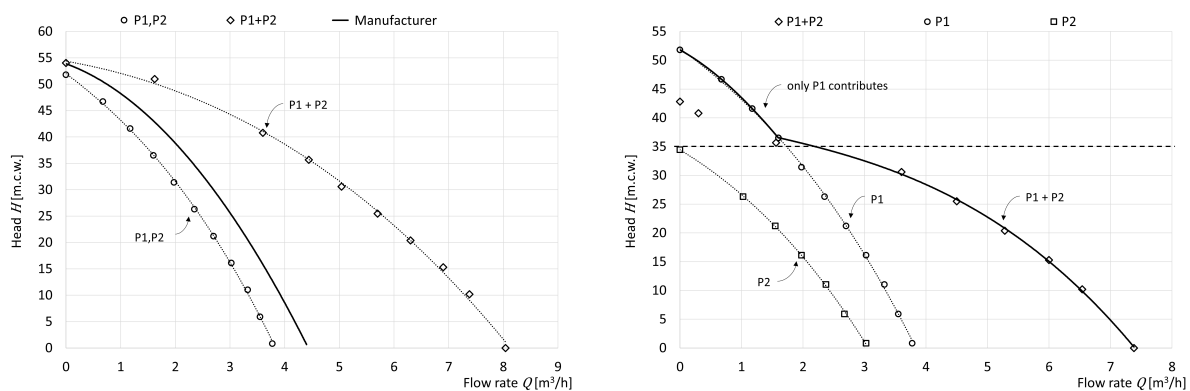


Figure 4.14: Characteristic curves for a combination of two identical pumps operating in parallel: a) at equal rotational speeds (50 Hz) (left); b) at different rotational speeds (P1: 50 Hz, P2: 40 Hz) (right).

In the second case (Figure 4.14 b)), the curves of the pumps operating individually at 50 Hz (P1) and 40 Hz (P2) are illustrated, along with the resulting curve of the parallel combination of both (P1 + P2). Since for low flow rate values the resulting shutoff head is lower than the shutoff head of P1, only P1 contributes in this zone, and P2 only starts contributing when its shutoff head is reached.

Series operation

Figure 4.15 shows the head versus flow rate curves for a combination of two identical pumps operating in series: at equal rotational speeds (45 Hz) (Figure 4.15 a)); and at different rotational speeds (P1: 50 Hz, P2: 45 Hz) (Figure 4.15 b)). The characteristic curve provided by the pump manufacturer at 50 Hz is also illustrated. As seen previously, the series combination would theoretically provide the sum of the shutoff heads of each individual pump. In the first case (Figure 4.15 a)), since the pumps are identical and operate at equal rotational speeds (45 Hz), it is observed that the shutoff head obtained with the series combination (88.2 m) is approximately equal to the sum of the shutoff head of each individual pump (43.6 m). Due to the previously mentioned discrepancy in obtaining the individual curves, the maximum flow rate recorded is not the same in the case of individual operation and series operation, which should be the case, and the sum of the shutoff heads does not exactly equal the head obtained with the series combination.

In the second case (Figure 4.15 b)), the curves of the pumps operating individually at 50 Hz (P1) and 45 Hz (P2) are illustrated, along with the resulting curve of the series combination of both (P1 + P2). For low flow rate values, the resulting shutoff head (98.4 m) is approximately equal to the combined sum of the head of each pump (P1=51.8 m and P2=43.6 m). For higher flow rates, exceeding the flow rate provided by the pump with the lower capacity (P2), only P1 should contribute (manufacturer curve), with P2 only contributing for flow rates below its maximum capacity.

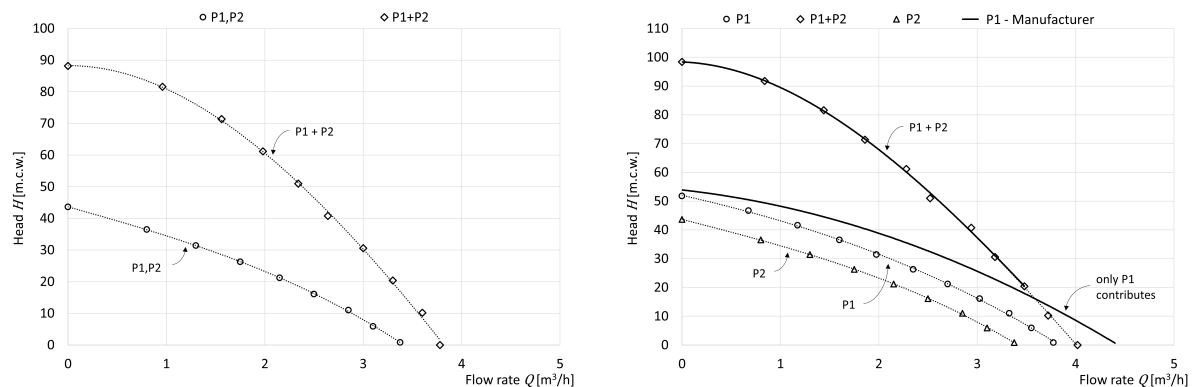


Figure 4.15: Characteristic curves for a combination of two identical pumps operating in series: a) at equal rotational speeds (45 Hz) (left); b) at different rotational speeds (P1: 50 Hz, P2: 45 Hz) (right).

5 PELTON TURBINE

5.1 Theoretical Considerations

Most hydraulic turbines can be classified into two basic types - impulse turbines and reaction turbines. In axial (e.g. Kaplan turbine) and radial flow (e.g. Francis turbine) turbines, two of the most known types of reaction turbines, the fluid completely fills the interior of the rotor, where the pressure gradually decreases in the direction of the flow as the fluid transfers energy to the rotor. The larger the pressure drop that occurs across the rotor, the larger the reaction. Additionally, the available head is only partially converted into kinetic energy (small amount), with the remaining portion (large amount) being converted into pressure energy (Dixon & Hall, 2014; Korpela, 2011).

The Pelton turbine, illustrated in Figure 5.1 and main objective of this part of the work, is a type of impulse turbine in which the total gross available head, H_g , (pressure head, velocity head and elevation head) is fully converted (except for losses) into the kinetic energy in one or more jets (up to 6), using a fixed nozzle.



Figure 5.1: Pelton Turbine from the LHM of ISEC.

The kinetic energy is then harnessed by a series of buckets spaced around the periphery of the rotor, as illustrated in Figure 5.2, generating torque and power. In this type of turbine, all the interaction between the fluid and the rotor occurs at atmospheric pressure, meaning, at constant pressure. Given the operating principle of this type of turbomachine, in order for the jet to be effective, its exit velocity, V_1 , must be as high as possible, hence the application of this type of turbine for high head applications (Viegas, 1989). According to Zang (2016), V_1 can be estimated by

$$V_1 = K_N \sqrt{2gH_n} \quad (5.1)$$

where K_N is the nozzle velocity coefficient, which typically ranges between 0.97 and 0.99, since not all the net energy, H_n , available at the inlet of the nozzle, is fully converted into kinetic energy in the jet, due to losses that occur in the nozzle.

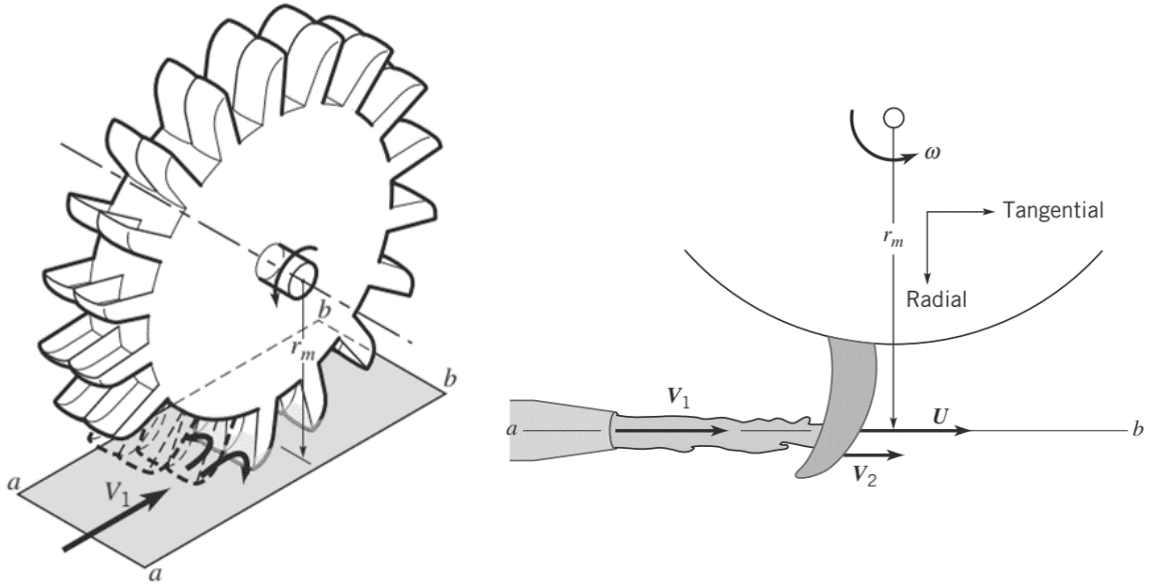


Figure 5.2: Ideal fluid velocities for a Pelton wheel turbine (Munson et al., 2013).

The high speed jet strikes the buckets with an absolute velocity V_1 and, due to their symmetric section, deflect the jet symmetrically on both sides by an angle β . When impacted by the jet, the buckets move with a linear velocity $U = \omega r_m$, and thus the tangential component of absolute velocity of jet, $V_{\theta 1}$, comes

$$V_1 = W_1 + U = V_{\theta 1} \quad (5.2)$$

As in centrifugal pumps, to determine the torque and power, the tangential components of the absolute velocities at the inlet and exit of the bucket should be known.

From Figure 5.3, the fluid particles travel along the bucket with a relative velocity with respect to the bucket of $W_1 = V_1 - U$, leaving it with a relative velocity of magnitude $W_2 = W_1$ (no friction and zero thickness of the separation edge), but at an angle β with respect to the incident jet direction (Munson et al., 2013). Thus,

$$V_{\theta 2} = W_2 \cos \beta + U \quad (5.3)$$

From equations (5.2) and (5.3)

$$V_{\theta 2} - V_{\theta 1} = (U - V_1)(1 - \cos \beta) \quad (5.4)$$

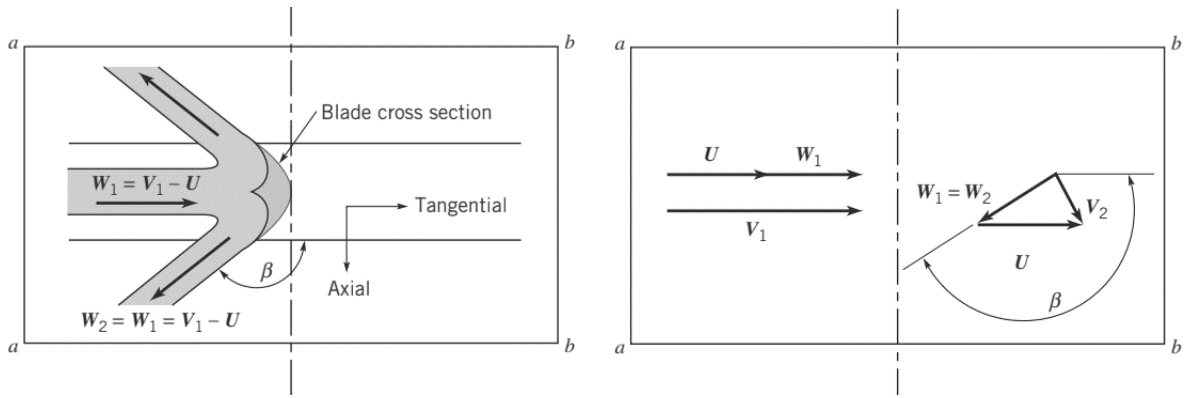


Figure 5.3: Inlet and exit velocity triangles for a Pelton wheel turbine (Munson et al., 2013).

Combining equation (5.4) with (2.22) and (2.23) the ideal torque supplied to the turbine comes

$$T_{shaft} = \dot{m} r_m (U - V_1)(1 - \cos \beta) \quad (5.5)$$

and the theoretical power

$$\dot{W}_{shaft} = \dot{m} U (U - V_1)(1 - \cos \beta) \quad (5.6)$$

This function corresponds to a quadratic function in U , which, with $\dot{m}$ and β being constants, reaches a maximum value when the derivative of the term $U(U - V_1)$ becomes zero, that is when

$$\frac{U}{V_1} = 0.5 \quad (5.7)$$

as can be seen in Figure 5.4, which illustrates typical theoretical and experimental torque and power curves for a Pelton turbine as a function of bucket speed. These curves will be reproduced from the experimental results obtained during the tests. The ratio $\Phi = U/V_1$ is normally referred to as the speed ratio or peripheral speed factor (Munson et al., 2013).

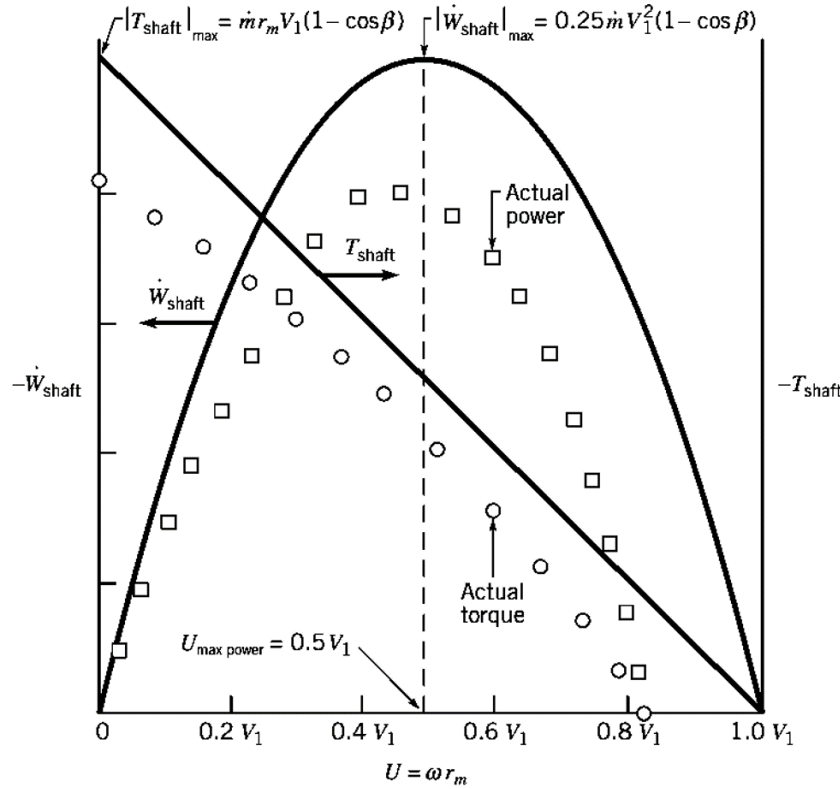


Figure 5.4: Typical theoretical and experimental power and torque for a Pelton wheel turbine as a function of bucket speed (Munson et al., 2013).

It can be seen from equation 5.6 that the maximum efficiency of the turbine would be achieved if each bucket completely reversed the relative velocity of the impinging jet ($\beta = 180^\circ$). However, since the water needs to be deflected to the sides to prevent interference with the subsequent bucket, β is typically around 165° (Zang, 2016).

As discussed in section 2.3, not all the available energy can be delivered to the turbine and the one effectively supplied is not fully converted into useful power. This discrepancy causes the performance curves predicted by theory to diverge from the actual ones. For this reason, it is important to begin by distinguishing between the various terminologies used in the literature as well as the major sources of energy losses. The layout of a Pelton turbine hydroelectric scheme is shown in Figure 5.5.

The water is stored in a dam with a constant level (only minor seasonal variations) from where it feeds the turbine through a pressurized pipe, the *penstock*. This value, the gross head, H_g , is the maximum energy available to the turbine and is given by the height difference between the turbine and the water level in the dam, also known as hydraulic head in the terminology of hydropower (Zang, 2016).

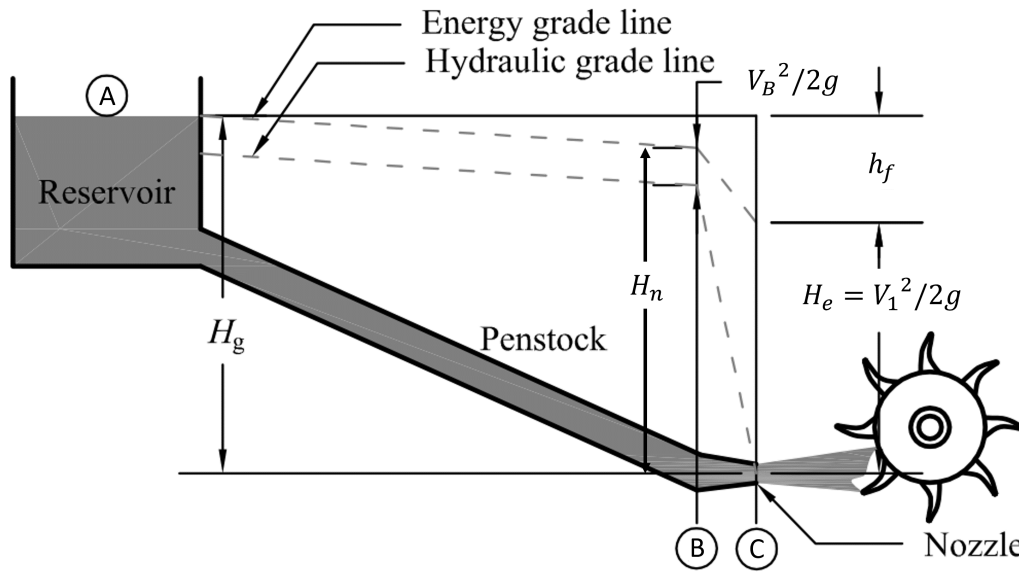


Figure 5.5: Layout of a Pelton turbine hydroelectric scheme. Adapted from Leon & Zhu (2014).

As the elevation decreases, the elevation head gradually transforms into pressure head. However, due to friction losses that occur along the penstock, the Hydraulic Grade Line (HGL) (given by the sum of the elevation head and the pressure head) decreases, thus reducing the energy that could be delivered to the turbine. It can be seen from the Energy Grade Line (EGL) (given by the sum of pressure head, velocity head and elevation head) that the contribution of velocity head is much smaller than the contribution of the remaining forms of head. In fact, and since the figure is not to scale, the contribution of kinetic head is almost negligible, as will be seen further on.

At point B, just before the nozzle, H_n represents all the elevation head that has been converted into pressure head (except for the losses that occurred in the penstock) and including the velocity head (almost negligible). This is the net head available to the turbine and only differs from point C, at the nozzle exit, due to head losses that occur in the nozzle. To be rigorous, the effective head, H_e , actually delivered to the turbine, corresponds to the hydraulic energy contained in the jet velocity if measured after the nozzle exit.

Usually, the turbine is coupled to an electric generator that must run at synchronous speed in order to match the grid values. Since load changes may occur from the grid side, the flow rate, Q , must be changed in order for the turbine rotate at constant speed. This adjustment is made through the use of a spear/needle valve that moves axially within the nozzle, changing the cross-section area, as illustrated in Figure 5.6.

If a sudden loss in load occurs a deflector plate temporarily deflects the jet from the buckets until the spear valve is moved to a new position, preventing the turbine from over-speeding, as shown in Figure 5.6.

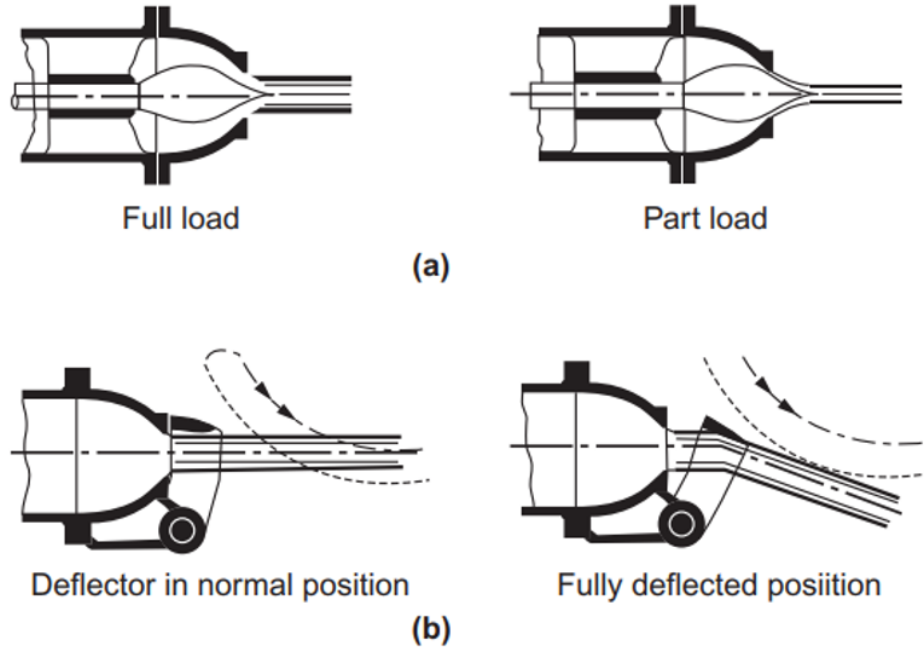


Figure 5.6: Methods of regulating the speed of a Pelton turbine: (a) with a spear (or needle) valve, and (b) with a deflector plate (Dixon & Hall, 2014).

5.2 Turbine Performance Characteristics

As seen in the previous section, in real cases, the water is stored in a dam at a given level. However, throughout the year, that level might change significantly. To ensure the turbine rotates at a constant speed, the useful power must be regulated by adjusting the needle in the nozzle, which changes the flow rate, Q . Therefore, in pedagogical terms, it is important to have a teaching facility capable of simulating those conditions, outside of nominal conditions of the turbine, as frequently occurs in real-life, to understand the associated phenomena and their influence on the parameters T_{shaft} , $\dot{W}_{shaft}$, and η (Calhindo et al., 2024).

The acquisition and plotting of different performance curves depend on the information to be obtained and the resources available at each educational institution. To understand the influence of a particular parameter on the turbine, the remaining ones must be kept constant while varying the control parameter chosen.

If intended to obtain the performance curves as a function of rotational speed, the net head and flow rate must be kept constant while varying the rotational speed. Once the test is conducted for the desired head, it is possible to vary the head (while keeping the flow rate constant) and repeat the test for each selected head, thereby obtaining the same curves as a function of rotational speed for different heads.

A simple change in the presentation of the results allows obtaining the performance curves as a function of head, for the different tested rotational speeds, allowing to understand both the influence of head and rotational speed on the power output and efficiency of the turbine.

On the other hand, if the aim is to understand the influence of the flow rate on the performance of the turbine, the head and rotational speed must remain constant. For a given head and rotational speed, the position of the nozzle needle is adjusted, thus resulting in a flow rate variation. A combination of different parameters allows for a wide range of data to be obtained. For a given constant head, different rotational speeds can be tested by spanning all positions of the nozzle needle for each rotational speed. Therefore, for each rotational speed, the process is repeated by re-adjusting the nozzle needle from the beginning. Similarly, the head can also be varied subsequently, further increasing the number of collected data points, since for another head, different rotational speeds can be tested again, which, in turn, enable the testing of different flow rates.

6 PERFORMANCE TESTING OF PELTON TURBINES

6.1 Experimental setup

6.1.1 ISEC experimental setup

The experimental setup of ISEC is shown in Figure 6.1, as well as the corresponding flowchart scheme in Figure 6.2.



Figure 6.1: Pelton turbine setup at the LHM of ISEC.

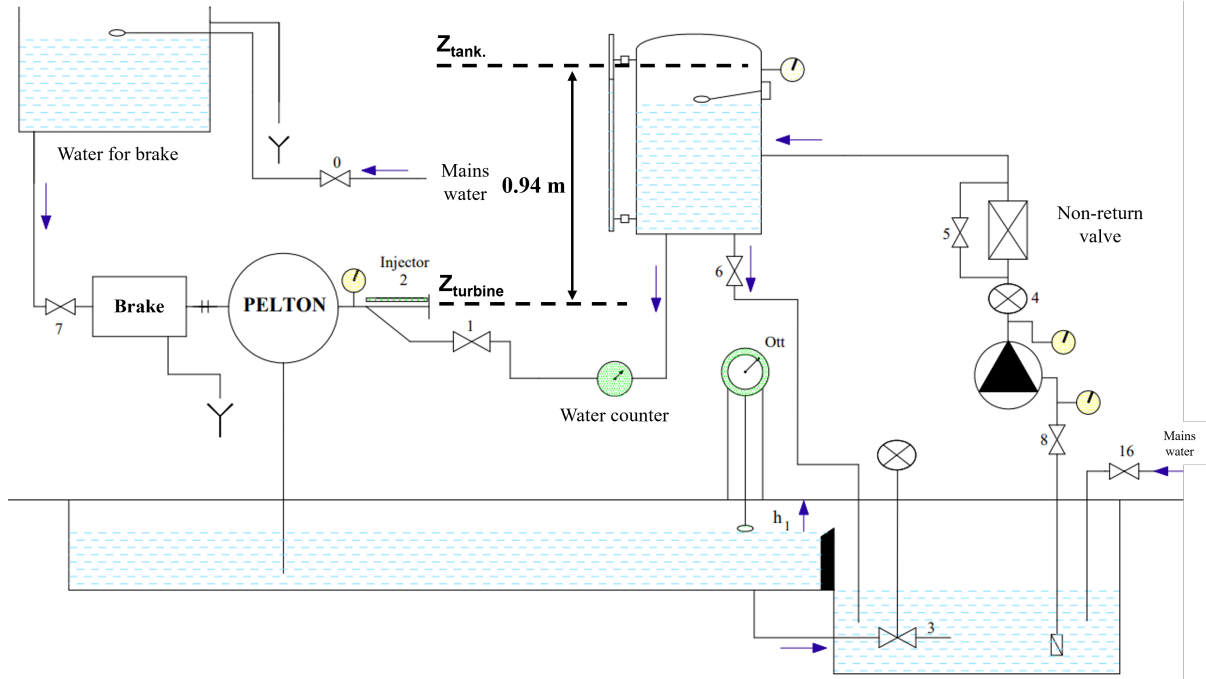


Figure 6.2: Flowchart scheme of Pelton turbine setup at the LHM of ISEC.

The test bench for the Pelton Turbine at the LHM consists of a horizontal shaft Pelton Spiral Turbine manufactured by B. MAIER, dating back to 1971, with a rotor diameter of 0.3 m, $\beta = 160^\circ$ and with nominal conditions presented in Table 6.1.

Table 6.1: Nominal conditions of the Pelton Turbine at the LHM of ISEC

Power	Flow rate	Height	Rotation speed
13.97 kW	0.03 m ³ /s	60 m	1000 rpm
19 CV	108 m ³ /h	-	104.7 rad/s

The turbine is supplied by a pressurized tank through a constant-speed centrifugal pump, model KSB ETA 80-26, manufactured in 1970, with the characteristics presented in Table 6.2.

Table 6.2: Nominal conditions of the KSB ETA 80-26 pump at the LHM of ISEC

Power	Flow rate	Height	Rotation speed
29.4 kW	0.03 m ³ /s	65 m	2900 rpm
40 CV	110 m ³ /h	-	303.7 rad/s

This pump is responsible for simulating the gross head, H_g , available in hydroelectric dams, also known as system head, since for experimental it varies with the head losses in the system.

The energy balance between the feed tank and the exit of the turbine leads to (Mendes, 2005; Munson et al., 2013)

$$H_g = \frac{P_{tank}}{\rho g} + 0.94[m.c.water] \quad (6.1)$$

where P_{tank} corresponds to the feed tank pressure and the value of 0.94 m to the level difference between the pressure measurement level in the feed tank and the turbine outlet. Accordingly, the energy balance between the entrance and the exit of the turbine leads to (Mendes, 2005; Munson et al., 2013)

$$H_n = \frac{P_{turbine}}{\rho g} + \frac{V^2}{2g}[m.c.water] \quad (6.2)$$

where $P_{turbine}$ corresponds to the pressure at the turbine inlet which, for experimental reasons, is kept constant to be in accordance with typical operating conditions of a turbine in a hydroelectric power plant. The shaft power produced by the turbine can be determined by (Mendes, 2005; Munson et al., 2013)

$$\dot{W}_{shaft} = T_{shaft} \times w = 0.735 \times \frac{F \times n}{1000} \quad (6.3)$$

where F corresponds to the force required to brake the turbine at a certain rotational speed, n , in rotations per minute. Finally, the efficiency can be determined by (Mendes, 2005; Munson et al., 2013)

$$\eta_T = \frac{\dot{W}_{shaft}}{\rho g H Q} \quad (6.4)$$

where H can be either H_g or H_n , depending on whether the objective is to calculate the efficiency of the installation or of the turbine, respectively.

The regulation of the flow rate (or its maintenance) is ensured through the adjustment of the needle position of the nozzle, with its measurement being performed using the equipment shown in Figure 6.3, namely, a discharge device, a volumetric and an ultrasonic flowmeter.



Figure 6.3: Equipment available at the LHM of ISEC for flow rate measurement.

The net head is regulated through the gate valve (1), with its reading taken from the manometer at the inlet of the injector nozzle, both illustrated in Figure 6.1 and 6.2. The rotational speed is varied by applying a resisting torque on the turbine shaft using a hydraulic brake, with the braking force read from the dynamometric balance coupled to the brake. The measurement of the rotational speed is performed using different equipment available at the LHM, such as a tachometer, a stroboscope, or by reading the variation in electrical properties, such as voltage, as illustrated in more detail in Figure 6.4.



Figure 6.4: Equipment available at the LHM of ISEC for $\dot{W}_{shaft}$ measurement.

6.1.2 UMH experimental setup

The Pelton turbine test bench at UMH consists of a centrifugal pump connected to a network of PVC pipes plus a flexible section with its corresponding water tank, forming a closed circuit. The scale model of the Pelton turbine, illustrated in Figure 6.5, designed considering the jet diameter and available head of the setup, was modeled using appropriate software, 3D printed, and post-processed to remove supports and achieve the final shape. The model has a rotor diameter of 80 mm, consisting of nine buckets, with an internal width of 22 mm. To enhance the quality of the turbine, the final model was printed using a resin 3D printer. Details of the design process and the methodology followed can be consulted in (Thake, 2000).

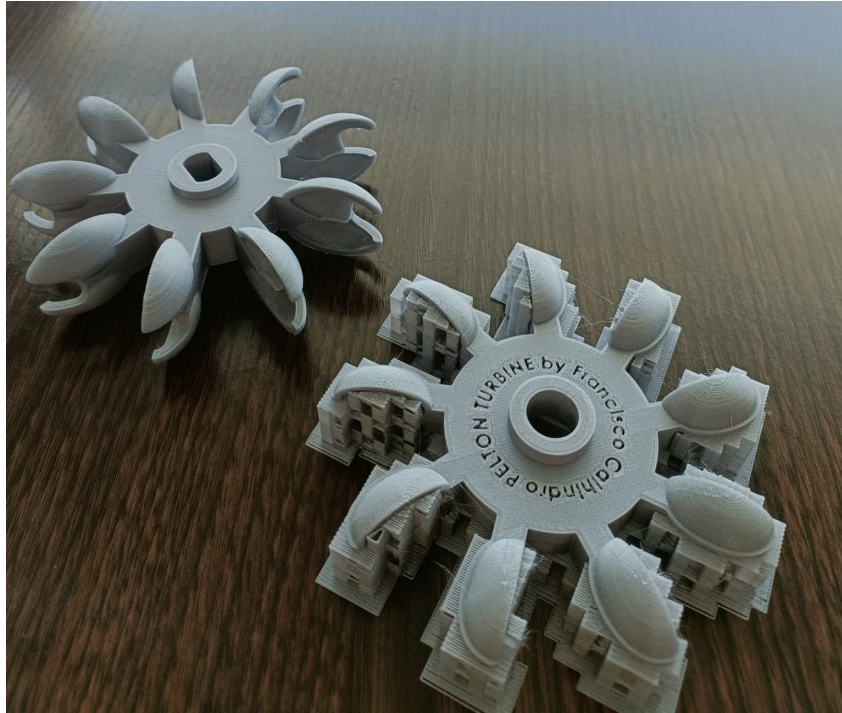


Figure 6.5: Example of a Pelton turbine model designed and 3D printed for the UMH setup, before and after post-processing.

The experimental setup at UMH is shown in Figure 6.6. The instrumentation mounted on the test bench enables the measurement of pressure, circulating flow rate, resisting torque, and the rotational speed of the turbine, being composed in more detail by:

- a water tank (A);
- centrifugal pump (B);
- nozzle (C);
- manometer on the discharge side (D);
- protective Pelton turbine box (E);
- variable area flowmeter of the rotameter type (F);
- VFD for regulating different rotational speeds while maintaining the set pressure (G).

The turbine model is housed in a watertight PET box with a thickness of 6 mm and dimensions of 15x15x15 cm. The box is fixed to the reservoir. The injector element consists of a cylindrical nozzle with an outlet section of 7 mm² (jet diameter of 3 mm).

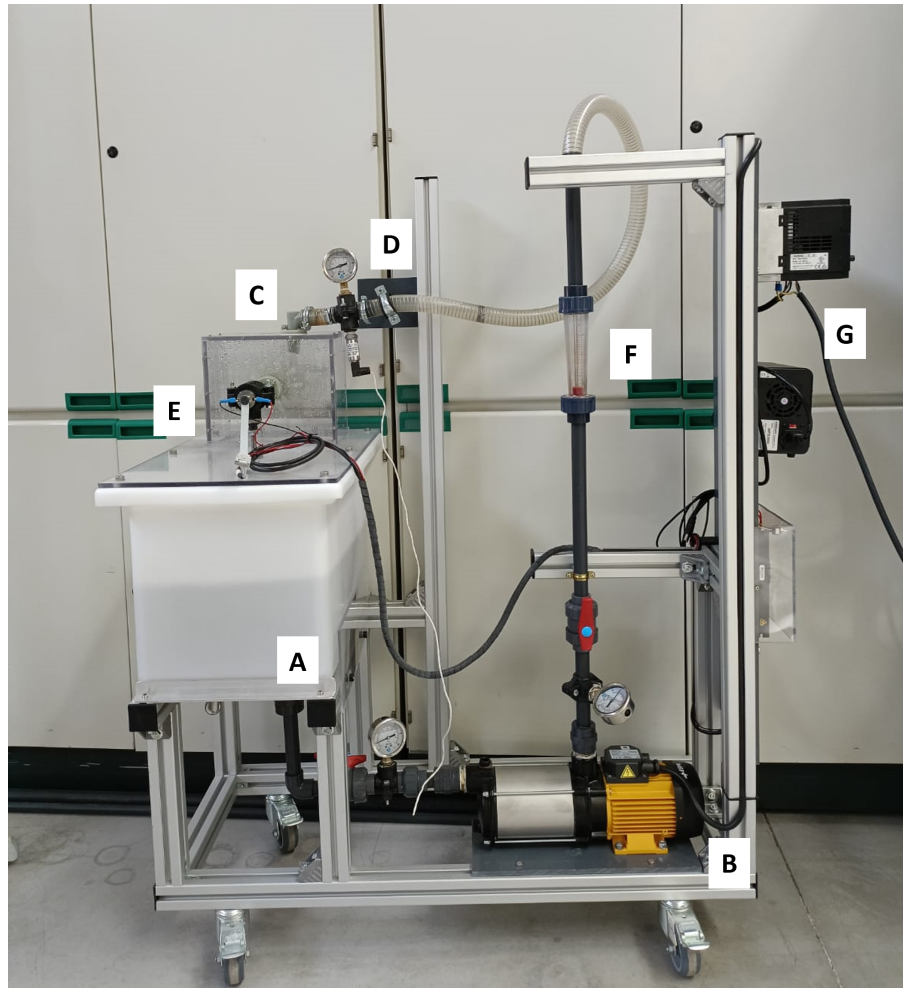


Figure 6.6: Pelton turbine setup at the LHM of UMH.

The turbine is supplied by a VFD centrifugal pump, model ESPA Prisma 15 3, with the characteristics presented in Table 6.3.

Table 6.3: Nominal conditions of the ESPA Prisma 15 3 pump at the LHM of UMH

Power	Flow rate	Height	Rotation speed
0.5 kW	0.000583 m ³ /s	25.1 m	50 Hz
0.68 CV	2.1 m ³ /h	-	-

The turbine rotor is coupled via an adapter piece to the shaft of a MAXON DCX32L motor, a brushed DC motor/generator capable of operating in four quadrants. Additionally, it is equipped with an encoder that allows for the measurement of its rotational speed. The motor/generator is connected to an ESCON 50/5 control and data acquisition card, which has a maximum supply voltage of 50 V_{DC}. For the test development, the card is powered at 13 V_{DC} through a Shunt type regulator, model DSR 70/30 12-75 V, designed to dissipate the power generated by the turbine generator. The acquisition and control card is connected to a PC, allowing for control and data acquisition of the turbine's electrical generation using *ESCON Studio* software (Ramírez, 2022).

The MAXON DCX32L generator will produce a direct current, I , proportional to the torque exerted by the rotor on the shaft, T_{shaft} , such that

$$T_{shaft} = \frac{I \cdot k_M}{\eta_e} \quad (6.5)$$

where k_M is the torque constant (0.0273 Nm A^{-1}) and η_e is the electrical efficiency of the motor (86.6 %).

6.2 Experimental procedure

6.2.1 ISEC experimental procedure

The test methodology to be followed varies slightly depending on the type of curves and information to be obtained.

Curves as a function of rotational speed or head

To obtain curves as a function of the rotational speed, the net available head and the flow rate must be kept constant throughout each test. The constant flow rate is ensured by maintaining the position of the needle of the nozzle unchanged, while the net head is kept constant by regulating the gate valve (1), shown in Figure 6.2. The rotational speed is varied by applying a resisting torque on the turbine shaft using a hydraulic brake coupled with a dynamometric balance, as shown in Figure 6.4.

Once the value of head and flow rate is established, the rotational speed is varied within the desired ranges. When valve (1) is fully open, the available head for the turbine amounts to 60 m, and the opening of the nozzle that ensures the maintenance of the 60 m of available head delivers a flow rate of approximately $80 \text{ m}^3/\text{h}$, representing the maximum conditions that can be replicated in the LHM.

Upon completed the first test for the specified conditions, the available head measured at point (2) is varied by inducing a pressure drop through valve (1) to the desired head for the subsequent test. However, since this valve simultaneously induces a reduction in the value of the flow rate, the position of the needle valve and valve (1) should be simultaneously adjusted to ensure the desired conditions: the new value of constant head (e.g., 55 m) and a constant flow rate, equivalent to the previous test.

Curves as a function of the flow rate

To obtain the curves as a function of flow rate, the methodology to follow presents a slight variation and can take two distinct paths, depending on the desired information to be obtained.

One of these paths, involves simulating what occurs in a real hydropower plant. For a fixed head, the nozzle opening is varied according to the requirements of the grid, thereby varying the supplied flow rate. After sweeping through all needle positions for a particular rotational speed, the same process can be repeated for another rotational speed. Furthermore, if desired, different heads can also be explored, repeating all the process from the beginning. A sweep of all rotational speeds for all nozzle openings allows demonstrating the influence of the same degree of opening on the value of the useful power for the various rotational speeds.

Another path to explore, corresponds to simulating the absence of the nozzle needle, not allowing the desired conditions to be fully simulated, being often the case in didactic installations due to its increased complexity, as it will be seen in the case of UMH laboratory.

To initiate this type of test, the wedge valve (1) is adjusted to simulate the desired head height, for example, 60 m, in the fully open position, with a flow rate of approximately $80 \text{ m}^3/\text{h}$ (the maximum at LHM). The position of the nozzle needle that provides this flow rate value is kept constant, and the desired range of rotational speeds is tested.

Upon completion of the test, valve 1 is adjusted to the desired head height (e.g., 55 m), while maintaining a constant position for the nozzle needle. As the head height is provided by a pump and the position of the nozzle needle is kept constant between different head heights, the flow rate obtained is, barring losses, the maximum flow rate that the pump can provide based on its characteristic curve. In other words, it is always the maximum flow rate available for the reproduced head height.

6.2.2 UMH experimental procedure

Since this setup does not include a regulation needle, it is not possible to control the flow rate through the nozzle.

Therefore, for a given pressure level at the nozzle inlet, the flow rate is constant, but it is not possible to maintain a fixed head while varying the flow rate. As will be demonstrated later, while the curves may appear similar, different testing methodologies are employed, particularly in maintaining a constant flow rate between the different curves. Since the regulation of the head and flow rate through throttle control results in a decrease in head with an increase in flow rate, and vice versa, regulation is done through VFD, thus continuously increasing the head and flow rate along the system curve with an increase in motor drive frequency.

This type of test corresponds to the last one described in the experimental procedure of the ISEC setup, as it can reproduce various conditions, including the absence of a regulation needle. The results will be addressed later, taking into account the different operating conditions and existing similarities.

6.3 Uncertainty analysis

The present section is based on the lecture slides from the Fluid and Energy Laboratories course (Coelho, 2024) taught by Professor Paulo Coelho at Faculdade de Engenharia da Universidade do Porto (FEUP). This analysis was performed solely for the turbine test due to the higher number of intervening parameters, the greater complexity of the system, the age of the equipment, and to avoid extending the scope of the document further. Nevertheless, the same methodology could be applied to the pump test as well.

Uncertainty is typically used to describe the quality of a measurement or experimental result. It represents the estimated error, considered as the error that would be measured through calibration, thus providing an estimation of the experimental error.

The uncertainty of the result consists of two types of errors: those determined through statistical analysis of a series of observations (random error) and those determined through previously measured data, general knowledge of the behavior and properties of relevant materials or instruments, manufacturer specifications, data provided in calibration certificates or other sources, and uncertainties attributed to reference data obtained from the literature (systematic error or bias).

If one intends to make a statement about the value of X based on a specific set of measurements, the best that can be said is that there is a $C\%$ confidence level that the true value of X lies within the interval

$$X_{best} \pm U_x \quad (6.6)$$

where, X_{best} is typically taken as the mean value of N readings or the single reading if $N = 1$. U_x represents the uncertainty in X , which corresponds to the estimate (with $C\%$ confidence) of the combined effects of random (P) and systematic (B) errors.

Random error

When it comes to random error, increasing the number of measurements towards infinity, if possible, would result in a smooth distribution following the normal or Gaussian distribution, a consequence of the Central Limit Theorem. The mean of the population distribution, μ , would be defined as

$$\mu = \lim_{N \rightarrow \infty} \frac{1}{N} \sum_{i=1}^N X_i \quad (6.7)$$

and the standard deviation of the population, σ , as

$$\sigma = \lim_{N \rightarrow \infty} \left[\frac{1}{N} \sum_{i=1}^N (X_i - \mu)^2 \right]^{1/2} \quad (6.8)$$

where σ^2 is known as variance, with higher values of σ indicating greater dispersion in readings and thus greater random error.

By reducing the number of variables through the use of a dimensionless variable, τ , known as the normalized deviation

$$\tau = \frac{X - \mu}{\sigma} \quad (6.9)$$

it can be said that the value of $\text{Prob}(\tau)$ corresponds to the area under the Gaussian curve between $-\tau$ and τ . In this way, in a Gaussian distribution of the population of readings of X , with a mean μ and a standard deviation σ , it can be said with, for example, 95% confidence that the next reading X_i will fall within the interval

$$\text{Prob}(\mu - 1.96\sigma \leq X_i \leq \mu + 1.96\sigma) = 0.95 \quad (6.10)$$

i.e., within the interval $\pm 1.96\sigma$ around the population mean, μ .

Since the population mean, μ , is generally not available, the concept of a 95% confidence interval allows to estimate the limits within which μ should lie. The mean of a sample of N individual readings, X_i , from the population is defined as

$$\bar{X} = \frac{1}{N} \sum_{i=1}^N X_i \quad (6.11)$$

and the standard deviation of the sample, S_x , as

$$S_x = \left[\frac{1}{N-1} \sum_{i=1}^N (X_i - \bar{X})^2 \right]^{1/2} \quad (6.12)$$

As the sample means also follow a normal distribution, similar to the population from which they were taken, it can be written that

$$\text{Prob} \left(\bar{X} - 1.96 \frac{\sigma}{\sqrt{N}} \leq \mu \leq \bar{X} + 1.96 \frac{\sigma}{\sqrt{N}} \right) = 0.95 \quad (6.13)$$

meaning with 95% confidence that the population mean, μ , lies within the interval of the sample mean calculated from N readings, with the interval now reduced by a factor $1/\sqrt{N}$.

From the definitions of σ and S_x , it is possible to observe that when $N \rightarrow \infty$, $\sigma = S_x$. However, for finite values of N , it is necessary to find a value of t that satisfies

$$Prob \left(-t \leq \frac{\bar{X} - \mu}{S_x/\sqrt{N}} \leq t \right) = 0.95 \quad (6.14)$$

with t being no longer equal to 1.96, since σ is actually unknown and only S_x is available to estimate σ , based on a finite sample of N readings ($t = 1.96$ when $N \rightarrow \infty$).

Instead of a normal distribution, the analysis follows a t -Student distribution, or simply t -distribution, with $N - 1$ degrees of freedom, denoted as v . Therefore, the random uncertainty, P , is defined as the 95% confidence limit of a sample of N measurements of the variable X obtained from a Gaussian distribution, where

$$P_{\bar{X}} = tS_{\bar{X}} = \frac{tS_x}{\sqrt{N}} \quad (6.15)$$

with t known from statistical tables for different degrees of freedom.

Since determining a 95% confidence interval for a sample of N readings depends on the sample size N , in practice, in many experiments, sample sizes are sufficiently large so that the value of t used in determining random uncertainty can be considered constant and close to the Gaussian value of 1.96. A study presented by Coleman & Steele (2018), shows that it is typically sufficient to take $t = 2$ in calculating random error when the number of readings is equal to or greater than 10, i.e.,

$$P_i = \begin{cases} t \cdot S_i & \text{if } N < 10 \\ 2 \cdot S_i & \text{if } N \geq 10 \end{cases} \quad (6.16)$$

Systematic error

While the estimate of the random component of uncertainty can be reasonably well-founded on statistical basis, the component resulting from systematic errors in a measured variable corresponds to what remains after calibrations, i.e. it is the same for each reading and will only be known if the true value of X is also known.

In real experiments where the true value X_{true} is unknown, a useful approach to estimate the magnitude of systematic error is to assume that, in a given scenario, systematic error is a value taken from a parent statistical distribution (typically Gaussian) of possible systematic errors.

If the estimate X_{best} is obtained from manufacturer specifications, a calibration certificate, a book, or another source, it may also be stated that the respective uncertainty U_x defines an interval with a confidence level of 90%, 95%, or 99%.

Unless stated otherwise, it can be assumed that a normal distribution was used to calculate this uncertainty. In this case, the standard deviation S_B associated with X_{best} (also known as standard uncertainty) can be calculated by dividing the stated uncertainty by the appropriate factor of the normal distribution, 1.64, 1.96 (2), and 2.58, respectively, for the three confidence levels mentioned.

In other cases, as will be seen later, it is only possible to estimate upper and lower bounds for X_{true} . Specifically, it can be said that the probability of X_{true} falling within a certain interval $[a_-, a_+]$ is, for all practical purposes, equal to one, and the probability of X_{true} being outside this interval is essentially zero.

This distribution corresponds to a uniform or rectangular distribution of possible values, with the expected value of X_{true} , X_{best} , being the midpoint of the interval: $X_{\text{best}} = \frac{a_- + a_+}{2}$. The standard deviation S_B associated with X_{best} is given by $S_B = \frac{a}{\sqrt{3}}$.

Global uncertainty of a measured variable

The overall uncertainty of a measured variable X is the interval around the best estimate X_{best} , within which the true value X_{true} is expected to lie with a given level of confidence, correctly combining estimates of systematic and random errors. According to ISO (1995), the method of combining uncertainty estimates is by adding the variances of the estimates (the squares of the standard deviations), resulting in the combined standard uncertainty, u_c , for a given variable i

$$u_c^2 = S_{B_i}^2 + S_i^2 \quad (6.17)$$

being the estimate of the standard deviation for systematic uncertainty, S_{B_i} , given by $B_i/2$, where B_i is the interval of a normal distribution with 95% confidence (B_i remains the same regardless of whether X is the result of a single reading or an average value, since the calculation process of X does not affect systematic error). The estimate of the standard deviation for random uncertainty, S_i , is given depending on the cases by equation (6.12) or by $S_X/\sqrt{N}$.

To associate a confidence level with the uncertainty of the variable, ISO (1995) recommends using the expansion factor k such that

$$U_{\%} = k_{\%} u_c \quad (6.18)$$

where $U_{\%}$ is the overall expanded uncertainty at a given confidence percentage, usually 95%. Since the combined error, u_c , is somewhat an estimate of the standard deviation of the total error distribution, and the central limit theorem states that the error distribution of the result, r , can often be assumed to approximate a Gaussian distribution, it is possible to use the t -distribution to obtain $k_{\%}$. Therefore, ISO (1995) recommends using values from the t -distribution to obtain the expansion factor $k_{\%}$, i.e., $U_{\%} = t_{\%} u_c$.

General uncertainty analysis

In the general case where an experimental result, r , is a function of J variables X_i , the combined standard uncertainty (u_r) of the result is given by

$$u_r^2 = \left[\left(\frac{\partial r}{\partial X_1} u_{X_1} \right)^2 + \left(\frac{\partial r}{\partial X_2} u_{X_2} \right)^2 + \dots + \left(\frac{\partial r}{\partial X_J} u_{X_J} \right)^2 \right] = \sum_{i=1}^J \theta_i^2 u_{X_i}^2 \quad (6.19)$$

where u_{X_i} are the combined standard uncertainties of the measured variables X_i and $\theta_i = \partial r / \partial X_i$ are the absolute sensitivity coefficients.

The equation for calculating u_r can be made dimensionless by dividing it by r ,

$$\frac{u_r^2}{r^2} = \left(\frac{X_1}{r} \frac{\partial r}{\partial X_1} \right)^2 \left(\frac{u_{X_1}}{X_1} \right)^2 + \left(\frac{X_2}{r} \frac{\partial r}{\partial X_2} \right)^2 \left(\frac{u_{X_2}}{X_2} \right)^2 + \dots + \left(\frac{X_J}{r} \frac{\partial r}{\partial X_J} \right)^2 \left(\frac{u_{X_J}}{X_J} \right)^2 \quad (6.20)$$

where u_r/r represents the standard relative uncertainty of the result, u_{X_i}/X_i denotes the relative uncertainties of each variable, typically much less than 1, and the coefficients $\frac{X_i}{r} \frac{\partial r}{\partial X_i}$ are referred to as uncertainty amplification factors. A specific and highly useful form of the equation (6.20) can be obtained if the data reduction equation takes the following form

$$r = k X_1^a X_2^b X_3^c \dots \quad (6.21)$$

resulting in

$$\frac{u_r^2}{r^2} = a^2 \left(\frac{u_{X_1}}{X_1} \right)^2 + b^2 \left(\frac{u_{X_2}}{X_2} \right)^2 + c^2 \left(\frac{u_{X_3}}{X_3} \right)^2 + \dots \quad (6.22)$$

where X_i represents only the variables that are directly measured. Thus, the result of the overall uncertainty analysis can be directly written without the need for calculating partial derivatives and algebraic manipulations.

In many cases, the experimental result is not measured directly, as previously implied. Instead, various variables are measured and combined into a data reduction equation. In the present case, $\dot{W}_{shaft}$ and η are not measured directly, but rather the respective variables involved in the formulation of equations 6.3 and 6.4. Considering the efficiency equation, and that

$$w = \frac{2 \times \pi \times n}{60} \quad (6.23)$$

$$T_{shaft} = F \times r \quad (6.24)$$

Assuming that the uncertainties in the values of gravitational acceleration and water density are negligible, there are 5 variables to be estimated — rotational speed n , brake force F , arm r , flow rate Q , and head H — propagating through the sub-functions — angular velocity w and resistant torque T_{shaft} — eventually contributing to the uncertainty in the turbine efficiency function η .

In each test, the parameters H_n and n were fixed ($H_n = 55$ m and $n = 1000$ rpm), while force (F) and flow rate (Q) varied, enabling the statistical analysis of random errors. For H , n , and r , random errors are assumed to follow a rectangular distribution with a range of $\pm$ half the smallest resolution of the measuring instruments. The systematic uncertainty standard deviation (S_{B_i}) is estimated as $B_i/2$, where B_i corresponds to a 95% confidence interval, unless additional manufacturer-provided information is available.

6.4 Results and discussion

As mentioned in the previous chapter, the tests conducted in both laboratories constitute a significant amount of experiments and collected data that allow understanding the influence of the various parameters involved in the previously mentioned performance equations. For better comprehension, these tests can be divided or classified as follows:

- Constant flow rate tests: a constant flow rate is maintained between tests, allowing the influence of the available head and rotational speed on the variation of parameters T_{shaft} , $\dot{W}_{shaft}$, and η_T to be understood. The flow rate is maintained by simultaneously regulating the injector needle and valve (1).
- Variable opening tests: a constant available head is maintained between tests, allowing the influence of flow rate and rotational speed on the variation of parameters T_{shaft} , $\dot{W}_{shaft}$, and η_T to be understood.
- Constant opening tests: it is not possible to maintain either the head or the flow rate constant between tests, due to the particularity of the absence of an injector needle element.

Table 6.4 summarizes the types of tests conducted and their respective characteristics, as well as which ones are possible to perform in both facilities.

Table 6.4: Summary of the tests conducted in each of the laboratories and their respective characteristics

Designation	Constant parameter between curves	Curves as a function of	ISEC	UMH
constant flow rate	flow rate	rotational speed	✓	×
		head	✓	×
variable flow rate	head	rotational speed	✓	×
		flow rate	✓	×
constant aperture	none	rotational speed	✓	✓
		head	✓	✓

Figure 6.7 shows the variation of the pump head, system head, net head and velocity head as a function of the flow rate, for a net head of 60 m, the net nominal head of the turbine.

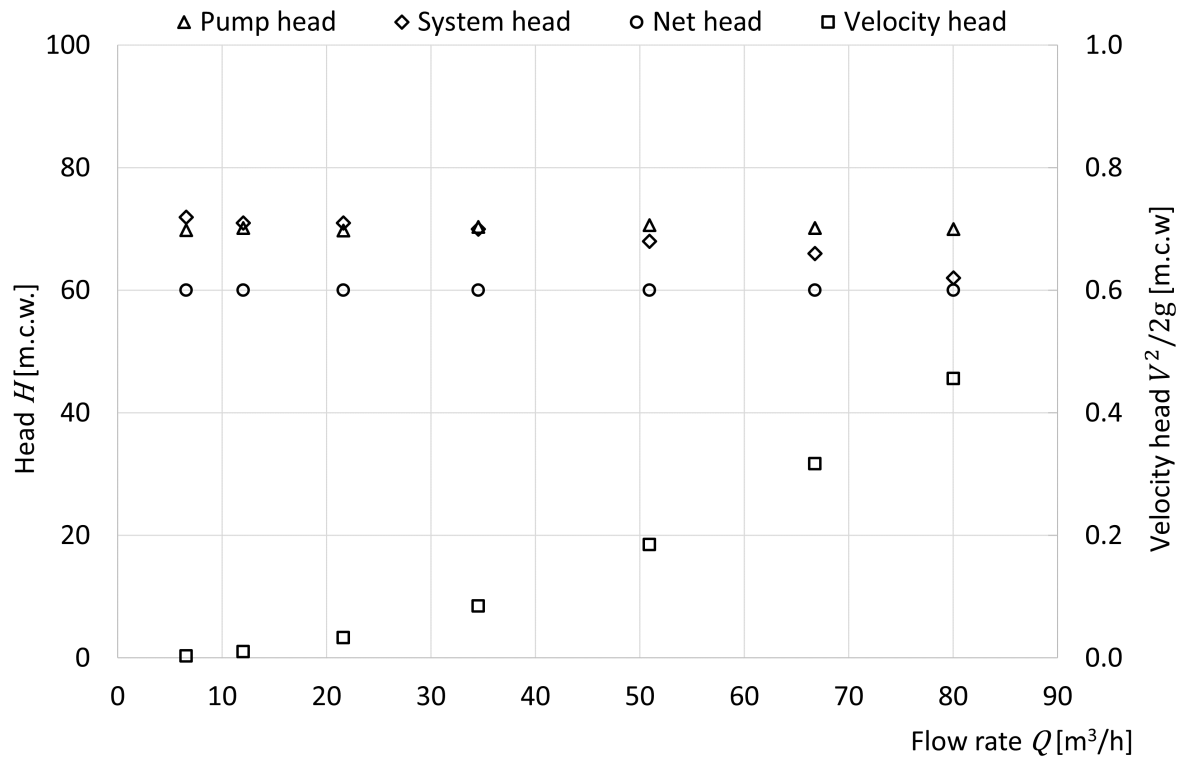


Figure 6.7: Variation of the pump head, system head, the net head and of the velocity head as function of the flow rate, for a net head of 60 m and a rotational speed of 1000 rpm.

Due to the short length of the piping between the pump and the reservoir and the presence of few components, the system head curve only starts to diverge from the pump head curve at higher flow rate values, due to the increased head loss observed.

When the gate valve is fully open, i.e. for 80 m³/h, the pressure at the reservoir almost equals the pressure at the injector (since the length of the pipe is small and there are only four components throughout the pipe), as can be observed by the near coincidence of the system head with the net head.

Figure 6.8 shows the variation of the useful (shaft) power, hydraulic power and both the efficiency of the turbine and installation as a function of the flow rate. The test was carried out in order to reproduce the manufacturer's nominal conditions - 108 m³/h, 60 m, 1000 rpm - despite the impossibility of reproducing such high flow rates in the LHM, where the maximum flow rate is 80 m³/h for the net head of 60 m.

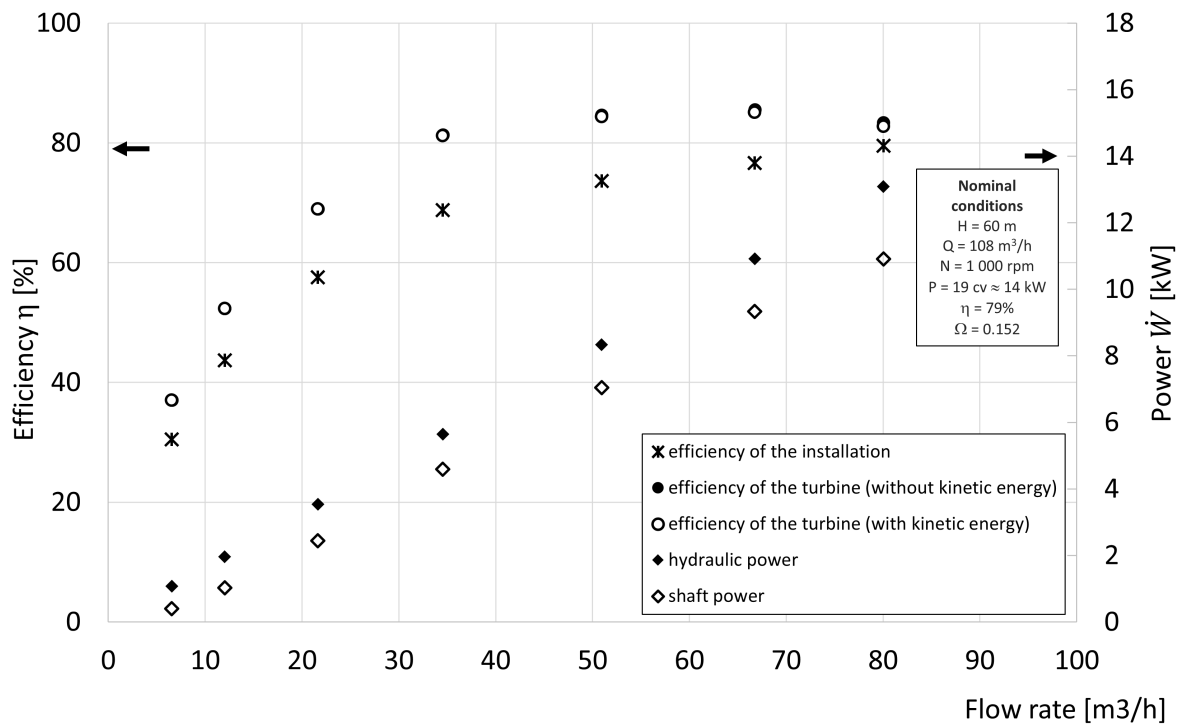


Figure 6.8: Variation of the useful (shaft) power, hydraulic power and both the efficiency of the turbine and installation as a function of the flow rate, for a net head of 60 m and a rotational speed of 1000 rpm.

Figure 6.7 and 6.8 correspond to tests of the *variable flow rate* type, where the constant parameter is the net head (60 m), and the curves are shown as a function of flow rate, with rotational speed maintained constant at 1000 rpm.

Figure 6.7 and 6.8, highlight that almost all of the energy available for the turbine is in the form of pressure, with a relative contribution of kinetic energy below 2%, before being fully converted into kinetic energy in the form of a jet in the injector. In this test the maximum values of the useful power and of the efficiency of the turbine were 10.9 kW and 85%, respectively.

6.4.1 Constant flow rate type tests

The curves shown in Figure 6.9 and 6.10 correspond to a reproduction of the curves presented in the previous section, in Figure 5.4, which illustrates typical theoretical and experimental torque and power curves, respectively, for a Pelton turbine as a function of the blade speed ratio $\Phi = U/V_1$, corresponding to tests of the *constant flow rate* type, where the constant parameter is the flow rate (80 m³/h), and the curves are shown as a function of rotational speed (blade speed ratio) and available net heads.

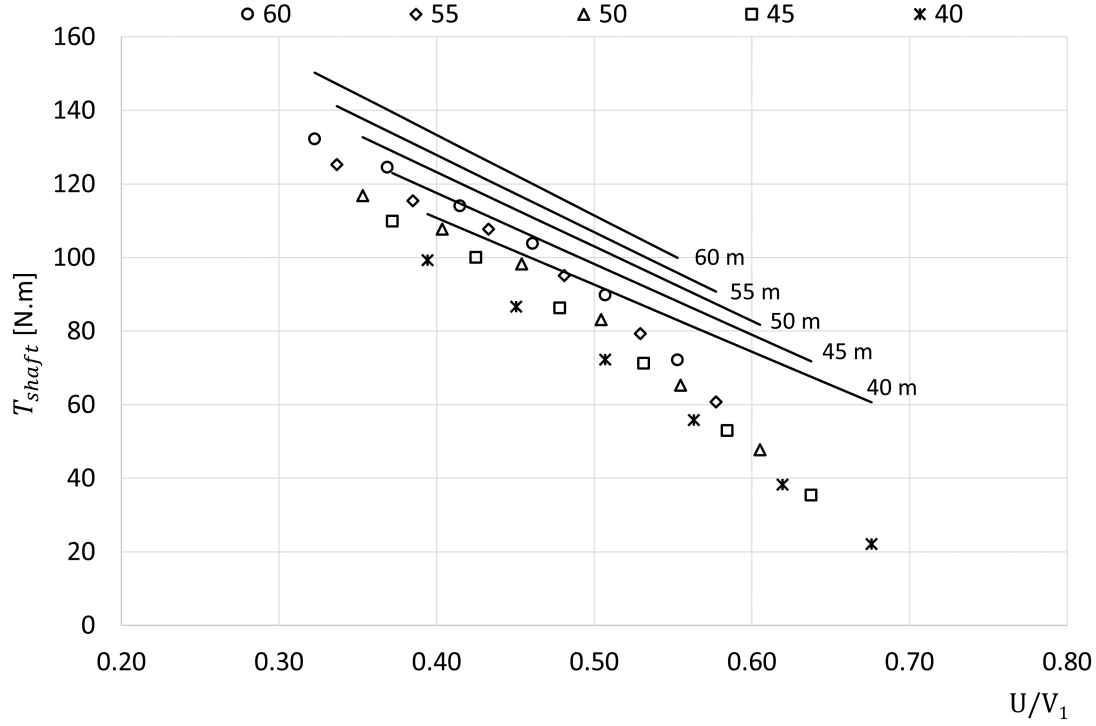


Figure 6.9: Theoretical and experimental variation of T_{shaft} with the ratio $\Phi = U/V_1$ for different net heads, with a constant flow rate of $80 \text{ m}^3/\text{h}$.

Although it was not possible to obtain the curves over the entire range of Φ for experimental reasons, the same trend observed in Figures 6.9 and 6.10 can be seen, for different net available heads. Although the torque tends to its maximum when the turbine is stopped, i.e., for $U = 0$ (Figure 6.9), there is no power under this condition and the curve should decrease to zero as Φ decreases (Figure 6.10), since there is no force and motion. When the bucket speed is one-half of the speed of the incoming fluid, i.e., $\Phi = 0.5$, the maximum theoretical power is observed and is equal to the power of the jet (in the Figure 6.10 is shown the power delivered to nozzle - dashed lines, with the jet power being slightly lower). The speed increases as T_{shaft} decreases, and it is expected that the maximum speed occurs when $T_{shaft} = 0$, i.e., the load from the resisting torque is completely removed from the turbine. At this moment the power will also be zero. For this case, theoretically $U = V_1$, and the turbine is freewheeling, with the water simply passing across the rotor without putting any force on the buckets.

However, it is possible to observe some differences between the curves obtained experimentally and those predicted by theory. The difference between the theoretical torque shown in Figure 6.9 and the actual brake torque, for the different net available heads and constant flow rate, is usually due in part to the effect of k (a factor accounting for friction loss in flow through the buckets), the angle β being different from the ideal one (180° vs. 160°), and the fact that the average angle between the incident absolute jet velocity V_1 and the linear velocity U , normally assumed to be zero, usually is not, especially at very low values of Φ , approaching to zero as Φ increases.

Although it is not possible to observe the full evolution of the curves, the torque curves seem to follow the shape found in the literature, i.e., for the lower speeds they tend to be more straight lines, but at higher speeds the torque decreases more rapidly, tending to zero for ϕ much less than 1 (cf. with Zang (2016), Daugherty & Franzini (1977) and Munson et al. (2013) or with Cardoso (2016) and Martins (2021)). This evolution is partly due to the fact that at speeds above the design value some of the water fails to complete its action upon the buckets or even overtakes them. Additionally, at higher speeds, the back of the bucket hits the water that has previously been intercepted by it and throws it up around the case, causing the back of the bucket to do work upon some of the water. In addition, there is some normal mechanical friction and windage between the wheel and the air.

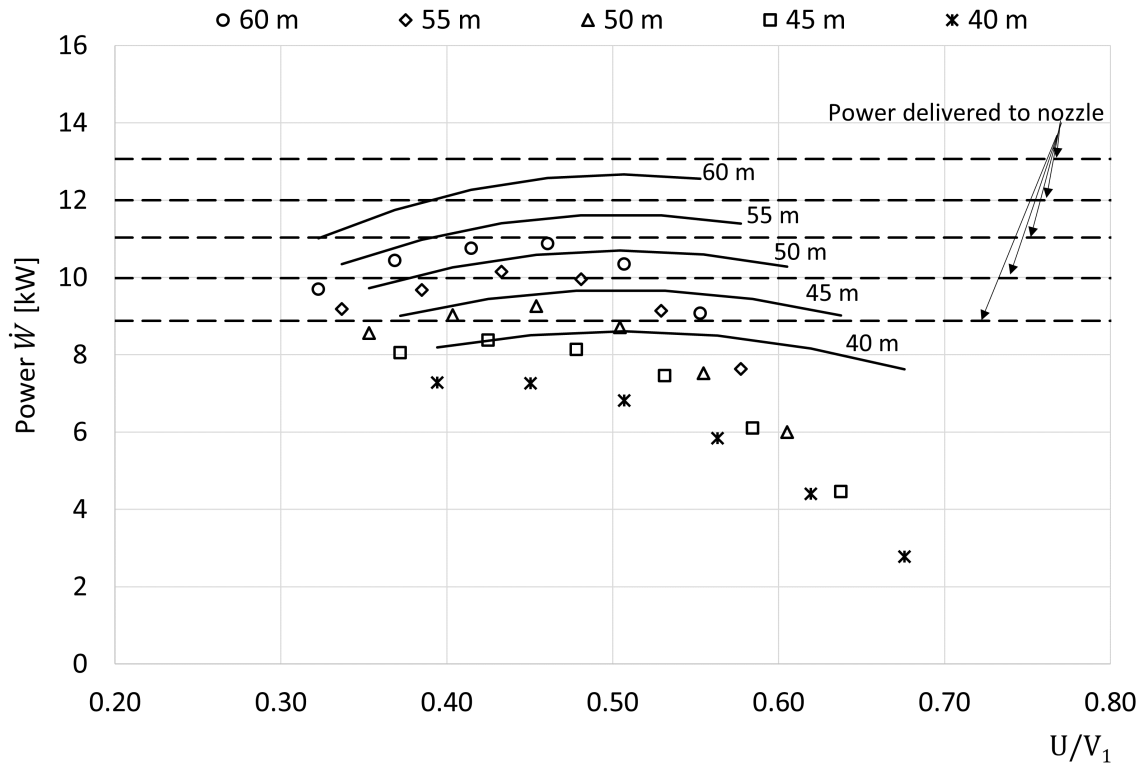


Figure 6.10: Theoretical and experimental variation of $\dot{W}_{shaft}$ with the ratio $\Phi = U/V_1$ for different net heads, with a constant flow rate of $80 \text{ m}^3/\text{h}$.

Due to the fact that the actual brake-torque curves are not straight lines, the actual power curves presented in Figure 6.10 do not have a parabolic shape either, with the right-hand side of the curve being steeper than the left-hand side.

Figure 6.11 illustrates the turbine theoretical and experimental efficiency as a function of Φ , for different net heads, with a constant flow rate of $80 \text{ m}^3/\text{h}$. Together with Figure 6.10 it can be seen that the value of Φ for maximum efficiency is not 0.5, as in the ideal case, but slightly less, between 0.43 and 0.47, as suggested in various literature (cf. with Zang (2016), Daugherty & Franzini (1977) and Munson et al. (2013) or with Cardoso (2016) and Martins (2021)).

Since the rotational speed is being normalized with respect to the head, through the bucket linear velocity and the jet velocity, respectively, the direct influence of rotational speed and head on the efficiency curves is removed, allowing the dependence of turbine efficiency on this velocity ratio to be observed. This results in the overlapping of the curves seen in Figure 6.11, showing that the hydraulic efficiency is independent of the head and depends only upon dimensionless quantities.

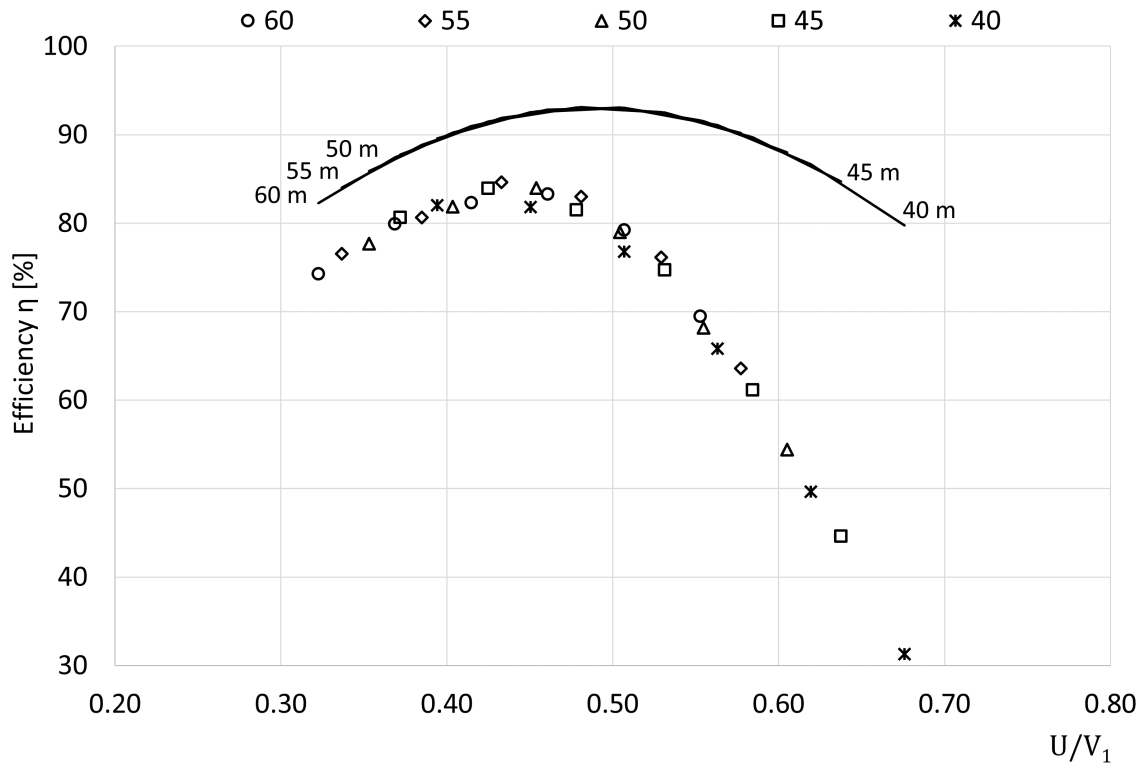


Figure 6.11: Theoretical and experimental variation of turbine efficiency with the ratio $\Phi = U/V_1$ for different net heads, with a constant flow rate of $80 \text{ m}^3/\text{h}$.

Figure 6.12 and Figure 6.13 show the variation of useful power and of the turbine efficiency, respectively, as a function of rotational speed for different heads, with a constant feed flow rate of $80 \text{ m}^3/\text{h}$, providing a better understanding of the previously presented curves, particularly the influence of rotational speed and net available head.

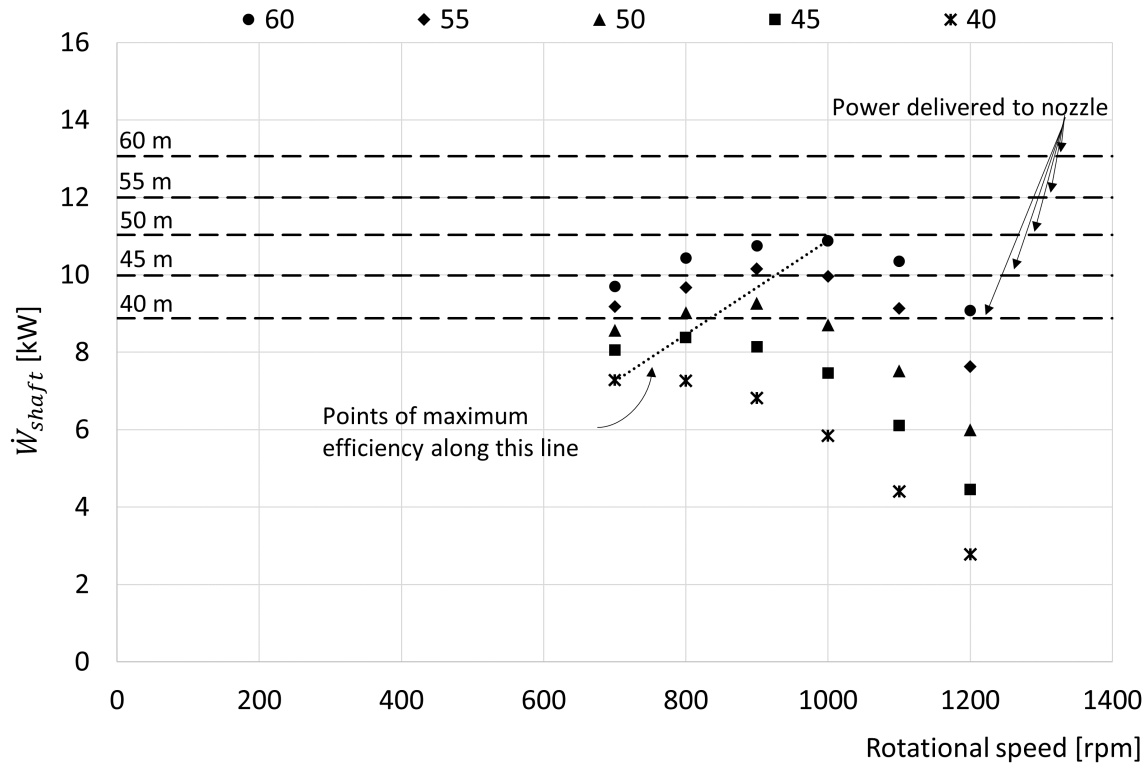


Figure 6.12: Variation of useful power $\dot{W}_{shaft}$ with rotational speed for different heads, with a constant flow rate of $80 \text{ m}^3/\text{h}$.

Since it is not possible to replicate in the LHM the nominal conditions indicated by the turbine manufacturer, the nominal useful power $\dot{W}_{shaft}$ value of 14 kW is not achieved. The maximum values of the useful power ranged between 10.9 kW (for a net head of 60 m) and 7.3 kW (for a net head of 40 m) (vd. Figure 6.12). As can be seen, since the flow rate is constant, an increase in the net available head results in an increase in the hydraulic power supplied to the turbine and consequently an increase in useful power, notwithstanding the importance of the quality with which this transformation is performed, as shown by the efficiency curves in Figure 6.13.

Although greater hydraulic power translates into greater useful power, does not mean that the transformation occurs with the highest quality, i.e., with the greatest efficiency. Figure 6.13 shows that as the rotational speed of the turbine increases, the head that maximizes efficiency shifts from 40 m (700 rpm) to 60 m (1200 rpm) due to the fact that, for increasingly lower heads, the force required to brake the turbine at higher speeds becomes progressively smaller. Thus, depending on the rotational speed, different heads can lead to the maximization of turbine efficiency. Additionally, it is observed that the best efficiency point (84.6%) is obtained for a head of 55 m and a rotational speed of 900 rpm, corresponding to a useful power of 10.2 kW vs. 10.9 kW for 60 m.

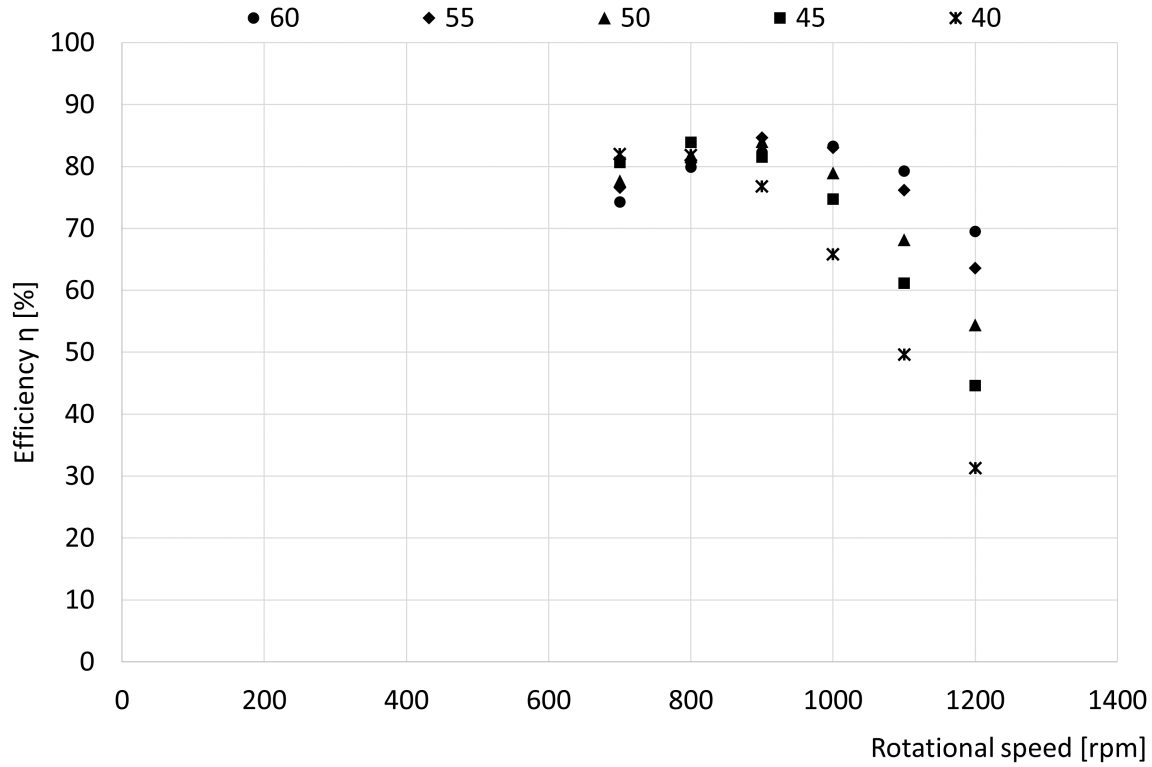


Figure 6.13: Variation of turbine efficiency with rotational speed for different heads, with a constant flow rate of $80 \text{ m}^3/\text{h}$.

6.4.2 Variable flow rate type tests

The curves shown in Figure 6.14 and 6.15 also correspond to a reproduction of the curves presented in the previous section, in Figure 5.4, which illustrates typical theoretical and experimental torque and power curves for a Pelton turbine as a function of the blade speed ratio $\Phi = U/V_1$. The tests are of the *variable flow rate* type, where the parameter kept constant is the head that provided the best efficiency in the *constant flow rate* type tests (55 m), and the curves are shown as a function of rotational speed (blade speed ratio) and for different injector apertures, from 20% to 80%.

Within the tested range of Φ , the same trend can be observed as in the *constant flow rate* tests, i.e., for a given injector aperture, the torque and power curves presented in Figure 6.14 and Figure 6.15, respectively, exhibit roughly the same shape as those observed in the *constant flow rate* tests: the torque tends to its maximum when the turbine is stopped; the speed increases as T_{shaft} decreases, being expected that the maximum speed occurs when $T_{shaft} = 0$ (Figure 6.14); the power curves tends to zero as Φ decreases, with no power existing under this condition since there is no force and motion; the theoretical maximum power occurring approximately at $\Phi = 0.5$ (not exactly since the theoretical points were calculated based on discrete experimental data) and the maximum experimental power in the remaining cases shifting to the left of $\Phi = 0.5$ (Figure 6.15).

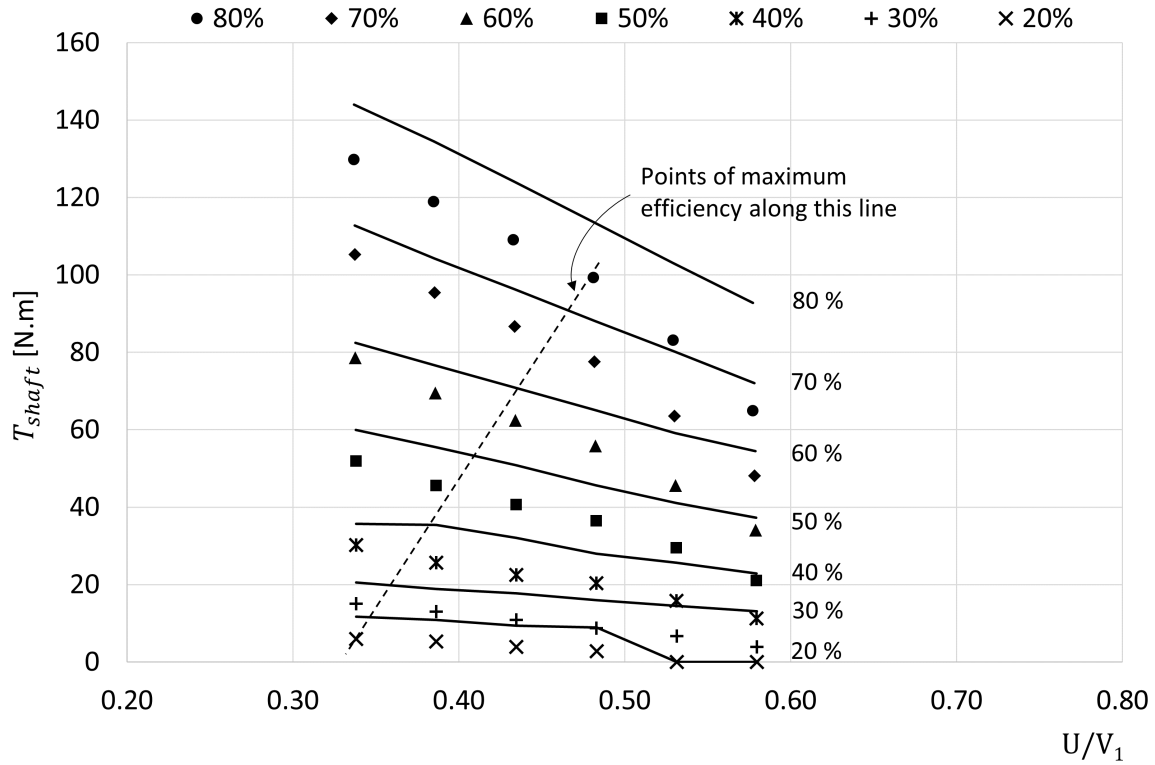


Figure 6.14: Theoretical and experimental variation of T_{shaft} with the ratio $\Phi = U/V_1$ for different injector apertures, with a net available head of 55m.

Figure 6.14 shows that the maximum brake torque occurs when the position of the needle, which is controlled by a screw thread, provides an 80% opening. Since for a given injector aperture, i.e., for each curve, the flow rate and head are constant, the reasons for the deviation between the experimental curve and the corresponding theoretical curve are the same as those stated in the *constant flow rate* type tests.

It can also be seen in Figure 6.14 and Figure 6.15 that Φ for maximum efficiency decreases slightly for smaller injector apertures.

As expected, the maximum power is obtained for the maximum injector aperture, with the power curves for smaller apertures having lower values, since the hydraulic power delivered to the turbine is also lower in these cases. The maximum power points are also found at lower values of Φ .

As the power input to the turbine is independent of the turbine speed and is therefore constant for the same aperture, the efficiency curve is the same as the power curve, but to some suitable scale. In fact, since the net head is maintained constant while varying the injector opening and for each opening the rotational speed is varied between 700 and 1200 rpm, the value of Φ remains constant for each tested rotational speed, independent of the aperture. Therefore, the power and efficiency curves will be the same whether expressed in terms of Φ or rotational speed.

The maximum value of the useful power registered was 10.4 kW for a flow rate of 81.4 m³/h, corresponding to an aperture of 80%, $\Phi = 0.48$ and 1000 rpm. The maximum values of the useful power ranged between 10.4 kW (for an aperture of 80%) and 0.85 kW (for an aperture of 20%).

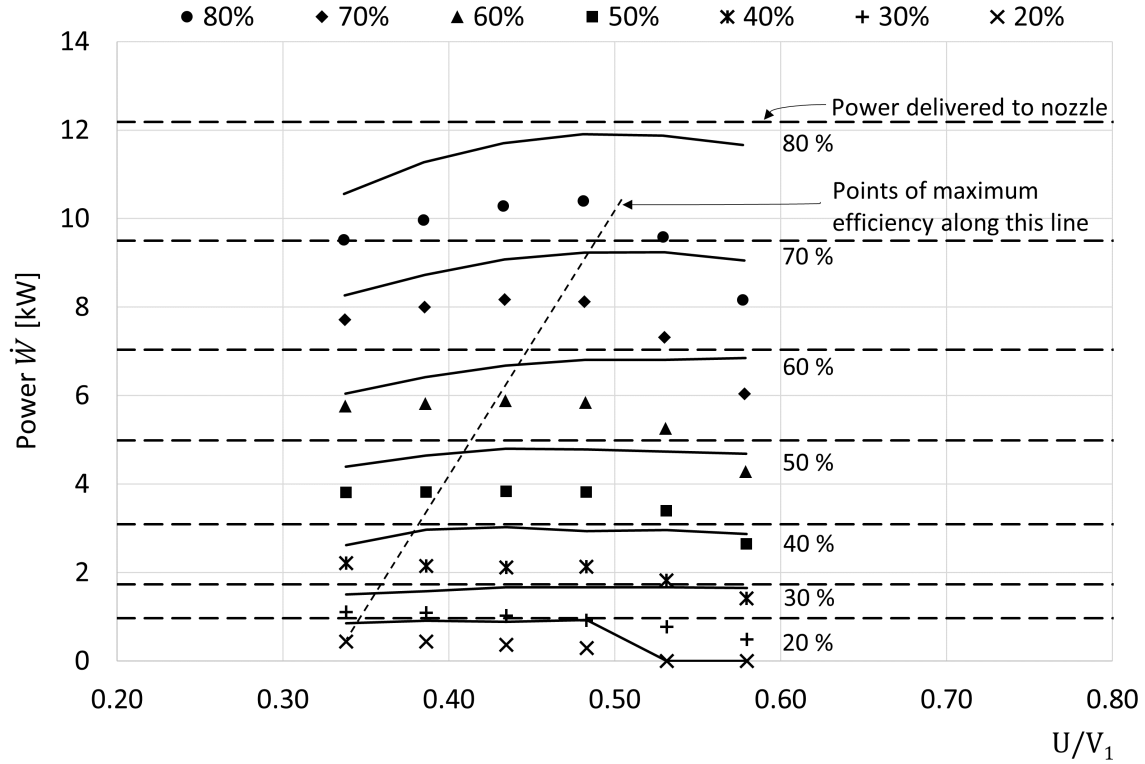


Figure 6.15: Theoretical and experimental variation of $\dot{W}_{shaft}$ with the ratio $\Phi = U/V_1$ for different injector apertures, with a net available head of 55m.

Figure 6.16 shows the variation of turbine efficiency, η_T , with rotational speed for different injector apertures, for a net available head of 55m, and it is observed that the best efficiency point (86%) is obtained for a flow rate of 63.4 m³/h, corresponding to an aperture of 70% (900 rpm) and not for 80%, and a useful power of 8.2 kW vs. 10.3 kW for 80 %, both at 900 rpm.

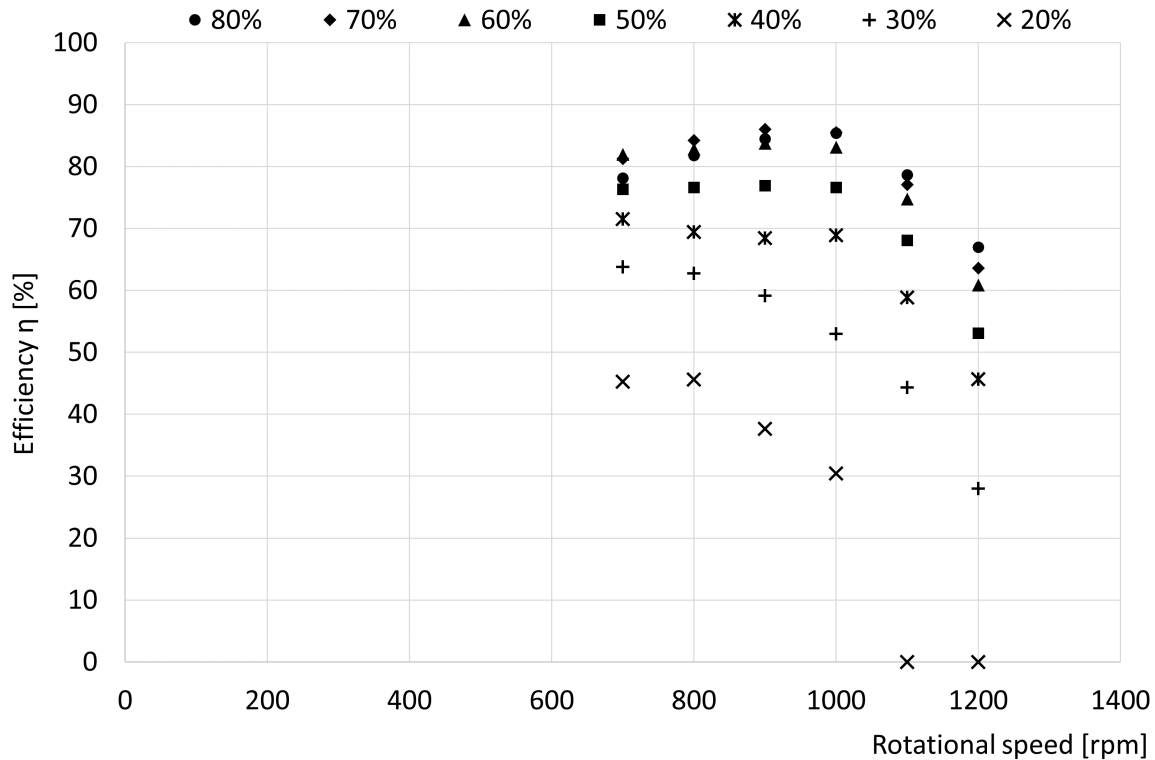


Figure 6.16: Variation of turbine efficiency with rotational speed for different injector apertures, with a net available head of 55m.

6.4.3 Operation at constant speed

As seen previously, a turbine usually operates at a constant rotational speed in order to meet grid requirements. Also, in the previous section, similar to results found in the literature, it was shown that there is one value of Φ that maximizes efficiency for any injector aperture but that one value Φ changes slightly with the injector aperture. Since the gross head, H_g , varies during the year, the net head, H_n , varies accordingly but also with the flow rate through the pipe. Therefore, Φ will vary with turbine speed, but even if rotational speed is constant, it will vary with head changes. Due to that, as also seen, the maximum efficiency at some constant rotational speed might not be the maximum efficiency of the turbine.

Figure 6.17 shows the efficiency curves for the turbine rotating at constant speed with varying load, for a constant net available head of 55m. It can be seen that if the rotational speed is changed for the same head, the resulting curve is different. The figure also shows that the efficiency curve is relatively flat over a wide load range, being the reason why this type of turbines are desirable for variable loads, especially if the load is low over long periods of time. For a constant head and rotational speed, there is a slight plateau where the load can vary below or above the load that leads to the best efficiency without causing a significant drop in turbine efficiency. Furthermore, this plateau is observed for various rotational speeds.

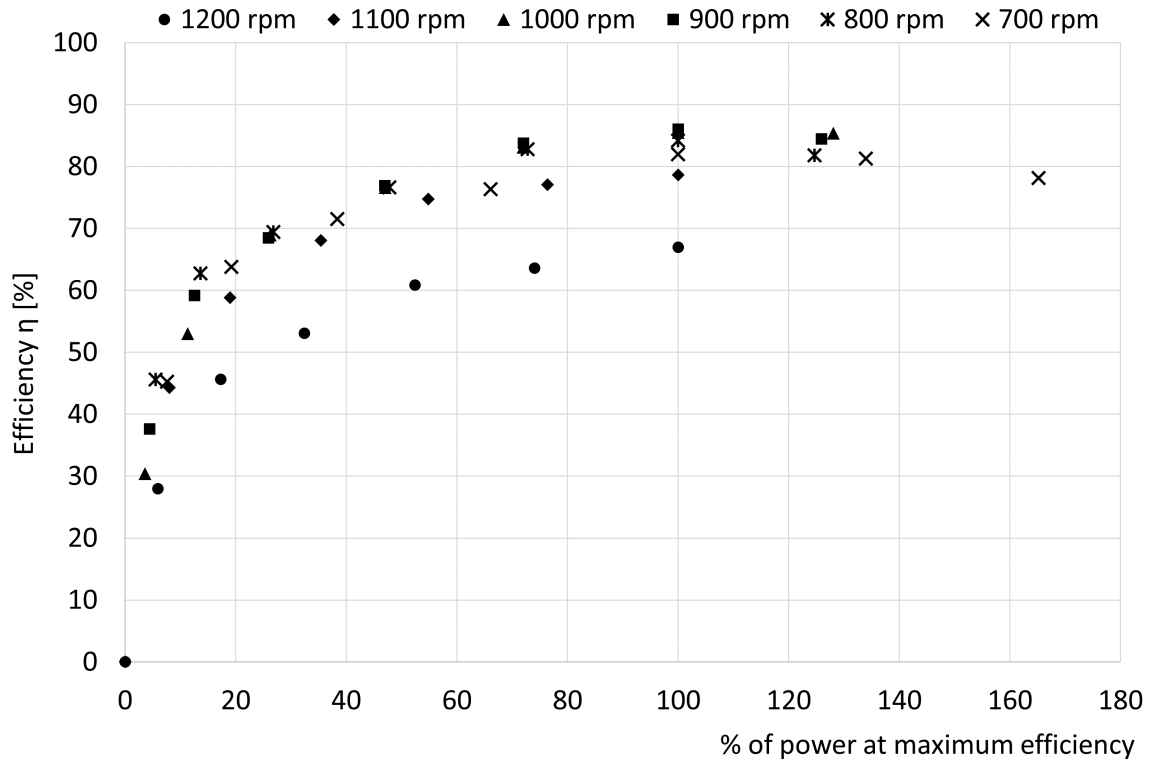


Figure 6.17: Variation of turbine efficiency with load for different rotational speeds, with a net available head of 55m.

Due to the hydraulic losses decreasing in proportion to the power output, a nearly constant efficiency is observed over a wide range of the load spectrum. However, for lower values of the load ratio, the overall efficiency rapidly decreases toward zero because the windage and bearing friction losses, which have not diminished, assume a relatively greater importance in this region.

Figure 6.18 shows the efficiency curves for the turbine rotating at constant speed with varying load, for a constant flow rate of $80 \text{ m}^3/\text{h}$. It is observed that, for a constant flow rate of $80 \text{ m}^3/\text{h}$, except for rotational speeds of 1200 rpm and 1100 rpm, for which the efficiency vary rapidly with the load, the load can vary (depending on the rotational speed considered) $\approx \pm 40\%$ below or above the load that leads to the best efficiency without causing a significant drop in turbine efficiency.

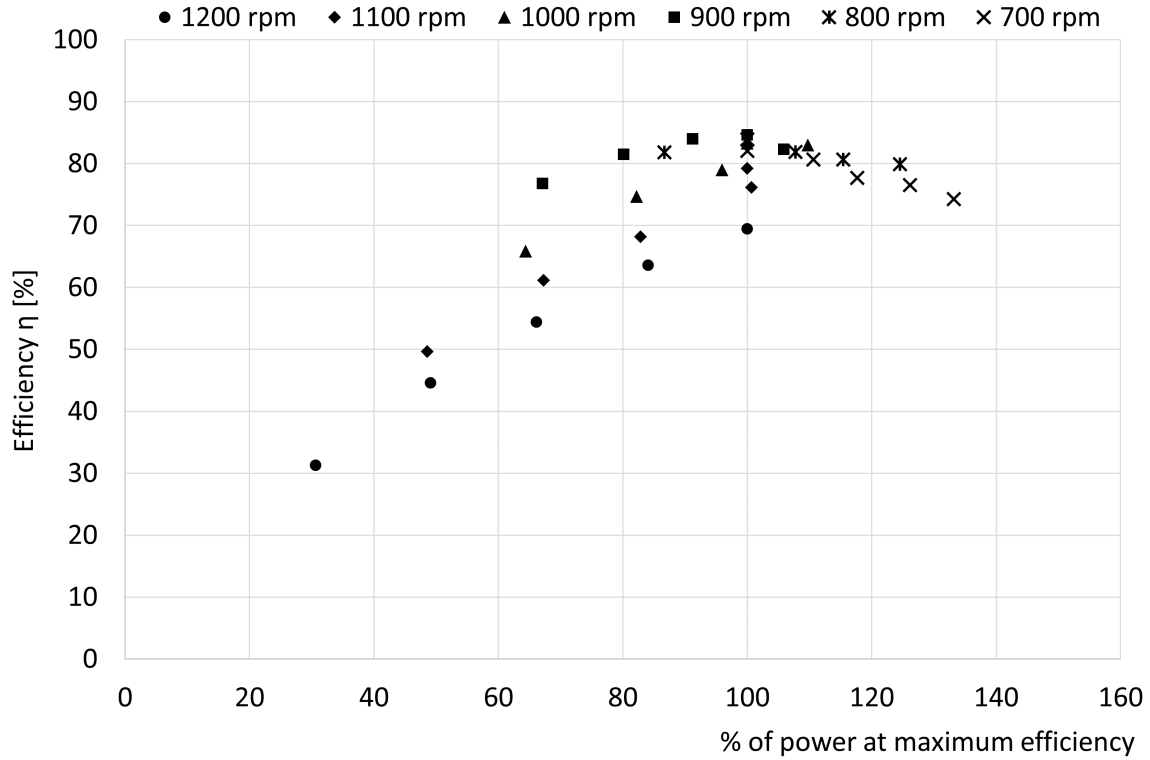


Figure 6.18: Variation of turbine efficiency with load for different rotational speeds, with a constant flow rate of $80 \text{ m}^3/\text{h}$.

6.4.4 Constant aperture type tests

The curves shown in Figure 6.19 illustrate the power (left) and efficiency (right) curves obtained for the 3D-printed model Pelton turbine as a function of the rotational speed, corresponding to tests of *constant aperture* type.

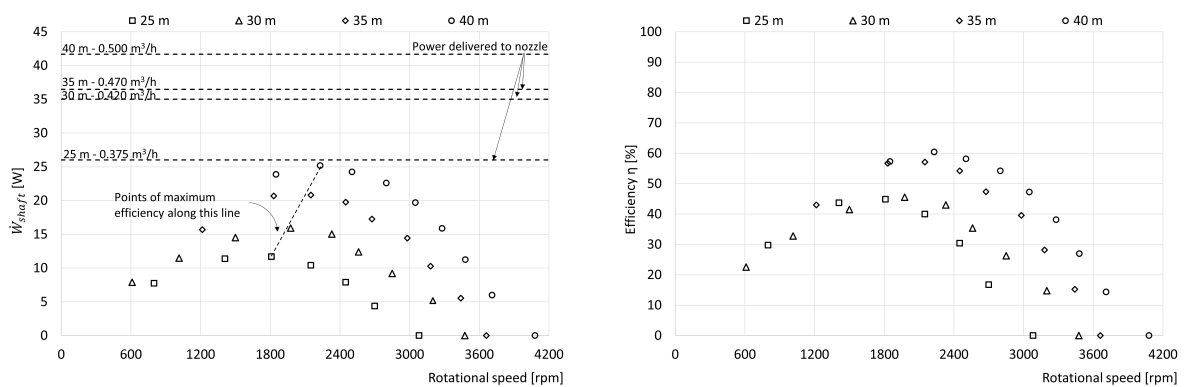


Figure 6.19: Variation of $\dot{W}_{shaft}$ (left) and turbine efficiency η (right), with rotational speed for a constant nozzle aperture (UMH).

To ensure that the head and flow rate increase together, regulation was performed using VFD instead of throttle control, in order to achieve net available heads of 25, 30, 35, and 40 meters.

The supply frequencies used to provide the respective heads were 38.61, 42.37, 45.5, and 48.77 Hz, resulting in flow rates of 0.375, 0.420, 0.470, and 0.500 m³/h, respectively. Since the current, I , produced by the MAXON DCX32L generator is proportional to the torque exerted by the rotor on the shaft, the rotational speed was varied between values corresponding to increments of 0.5 A, ranging from 0 to 4 A of current.

The obtained curves show approximately the same trend as those presented earlier; however, while the curves may appear similar, in this type of test it is not possible to maintain a constant head or flow rate from curve to curve. Since the head is provided by a pump and the aperture is kept constant between different heads, the flow rate obtained is, barring losses, the maximum flow rate that the pump can provide for the respective motor drive frequency. In other words, it is always the maximum flow rate available for the reproduced head, not allowing for a fixed head while varying the flow rate. Due to the small size of the model, the maximum recorded power varied between 11.7 W for $H_n = 25$ m and $Q = 0.375$ m³/h, and 25.5 W for $H_n = 40$ m and $Q = 0.500$ m³/h. These values correspond to hydraulic powers of 26 W and 41.7 W, respectively. Between these values of head and flow rate, the turbine efficiency ranged from 44.9% to 60.4%, respectively.

6.4.5 Uncertainty quantification tests

The test chosen for uncertainty evaluation was the *variable flow rate* type test, as it presented the highest number of regulation parameters, making it the most disadvantageous one in terms of the associated uncertainty. Therefore, the tested head was 55 m, rotational speed 1000 rpm, and the injector opening settled to 60%. For all 11 tests performed, the procedure followed was always the same: turn on the installation, adjust the desired head value (55 m) through the gate valve for the selected injector opening (60%); adjust the rotational speed (1000 rpm) using the brake handle; wait approximately 3 minutes for the values to stabilize and record all observed values, including braking force and flow rate. After recording the values, all regulator handles and valves were returned to their initial positions (or other random positions) and the installation was turned off. The process was repeated again after about 5 minutes.

Table 6.5 summarizes the characteristics of the equipment used for measuring the referred variables, as well as the estimates of the standard deviation of systematic and random uncertainty.

Half of the smallest division of each equipment's scale is considered the 95% confidence interval, and is divided by the expansion factor of 2 to estimate the standard deviation for systematic uncertainty. Since the ultrasonic flowmeter is of the "clamp-on" type, the measured variable is the average velocity, and the user must input the internal diameter of the pipe as additional data. In this case, the flowmeter error must be determined by propagating the uncertainty of the function $Q = V \cdot A$.

Given that the flowmeter manufacturer provides an accuracy of $\pm 1\text{-}3\%$ of reading within the velocity range for fully developed flow, a conservative estimate of 3% was considered. For the estimates of standard deviation for random uncertainty, statistical treatment is performed for the variables Q and F . In other cases, a rectangular distribution is assumed.

Table 6.5: Variables involved, their values, and estimates of standard deviation for systematic and random uncertainties

Variable	X_i or $\overline{X}_i$	Smallest resolution	S_{Bi}	S_{Pi}
F [kgf]	8.463	0.05	$\pm 0.025/2$	± 0.007639
H_n [m]	55	1	$\pm 0.5/2$	$\pm 0.5/\sqrt{3}$
n [rpm]	1000	1	$\pm 0.5/2$	$\pm 0.5/\sqrt{3}$
r [m]	0.716	0.001	$\pm 0.0005/2$	$\pm 0.0005/\sqrt{3}$
V [m/s]	1.497	N.A.	$\pm (0.03 \times 1.497)/2$	N.A.
D [m]	0.1053	0.0001	$\pm 0.00005/2$	N.A.
Q [m ³ /h]	46.95	N.A.	N.A.	± 0.09181

To facilitate the calculations, the aforementioned data and the efficiency calculation equation were input into the Engineering Equation Solver (EES) software, and the "Uncertainty Propagation" function was subsequently used for the corresponding calculations. Table 6.6 shows the values of random and systematic uncertainty for each of the variables, as well as for the objective function η , presented at 95% confidence and the corresponding total uncertainty. The table also includes the percentage contribution to the uncertainty of each variable, both in the random and systematic components.

Table 6.6: Values of random, systematic, individual contribution to uncertainty of each variable and total uncertainty for the objective function η , at 95% confidence

Variable $\pm$ Uncertainty		Partial derivative	% of uncertainty	
Systematic	Random		Systematic	Random
$F = (8.463 \pm 0.0125)$	(8.463 ± 0.007639)	$\partial\eta/\partial F = 10.45$	0.88 %	2.51 %
$H_n = (55 \pm 0.25)$	(55 ± 0.2887)	$\partial\eta/\partial H_n = -1.609$	8.30 %	84.94 %
$n = (1000 \pm 0.25)$	(1000 ± 0.2887)	$\partial\eta/\partial n = 0.08847$	0.03 %	0.26 %
$r = (0.716 \pm 0.00025)$	(0.716 ± 0.0002887)	$\partial\eta/\partial r = 123.6$	0.05 %	0.50 %
$V = (1.497 \pm 0.0225)$	-	$\partial\eta/\partial V = -59.09$	90.66 %	-
$D = (0.1053 \pm 0.000025)$	-	$\partial\eta/\partial D = -1680$	0.09 %	-
-	$Q = (46.95 \pm 0.09181)$	$\partial\eta/\partial Q = -1.885$	-	11.79 %
$\eta = (88.47 \pm 1.396)\%$	$\eta = (88.47 \pm 0.5039)\%$			
$\eta_{95\%} = (88.5 \pm 2.8)\%$	$\eta_{95\%} = (88.5 \pm 1)\%$			
$u_{cr} = \sqrt{1.396^2 + 0.5039^2} = \pm 1.484\%$				
$\eta_{95\%} = (88.5 \pm 2.97)\%$				

At a 95% confidence level, the objective function value obtained for the selected point was 88.5%, with a total associated uncertainty of $\pm 2.97\%$. The variable that contributed most to the systematic error was the velocity, V (90.66%), while the variable that contributed most to the random error was the net available head, H_n (84.94%). In relative terms, since the most unfavorable point was chosen for testing, this will initially represent the largest error associated with the turbine test. However, since only a single measurement is taken during the tests conducted by students in class, it is expected that the error at this point will be higher, directly influencing only the random error, as the systematic error, as previously mentioned, does not depend on the number of measurements.

It is also important to note that the force measurement scale shows a zero deviation, which was corrected in the results, but it is unknown to the author whether this deviation is solely static or also dynamic, potentially contributing, along with the previously mentioned reason, to the difference between the efficiency values obtained and those provided by the manufacturer.

Some proposals to reduce the error include increasing the number of measurements at each point to address the random error, and in cases where a rectangular distribution was considered due to a lack of better information, attempting to video-record multiple measurements for later analysis, allowing for more accurate and comfortable value extraction, frame by frame, would be interesting. As for systematic errors, these could be improved through calibration or by using instruments with higher precision.

7 CONCLUSION

7.1 Concluding Remarks

At this stage, it is important to review the main conclusions of the work carried out in relation to the outlined objectives, make some comments on aspects of special interest, and define future objectives, aiming to the continuity of the research presented here on engineering education.

The current paradigm requires engineers to be aware of real-world challenges at an earlier stage of their professional journey, ideally during their academic training. Practical learning experiences at the academic level allow future engineers to develop a comprehensive understanding of engineering concepts and their practical applications, equipping them with the hard and soft skills necessary to face the often unexpected challenges of the industry.

Thus, the present contribution provides insights on hydraulic turbomachinery lectures, serving as a portrayal assessment of current testing methodologies at ISEC and UMH, encompassing also several other dimensions of turbomachinery studies, aiming to connect theoretical concepts with real-world challenges.

The comprehensive overview of some theoretical topics of centrifugal pumps presented at Chapter 3, along with some of the major practical aspects of centrifugal pump operation such as cavitation, pump performance and system characteristics curves, control and adjustment of pump performance, and association of pumps serve as a background to Chapter 4, where the performance testing of centrifugal pumps is addressed. Typical turbomachinery courses usually cover, in a more detailed way, the topics of Chapter 3. Chapter 4 represents therefore a practical application of the topics presented at the preceding chapter, representing practical insights to the field of turbomachinery engineering education.

The existing experimental setups at ISEC and UMH for centrifugal pump testing of individual pumps are described, as well as the test bench for centrifugal pump testing in series and parallel configurations available at UMH. The testing methodologies used at each institution are detailed, depending on the available equipment and the type of test to be performed, with the objective of obtaining the relevant curves for different rotational speeds.

The results obtained from tests of pumps operating individually and in series and parallel combinations are presented and discussed, comparing and linking the results to the theoretical foundations presented at Chapter 3 whenever possible. For individual operation, a good agreement is observed not only in terms of the expected shape of the curves but also in comparison with those predicted by theoretical similarity laws, in cases where these apply. The NPSH curves, drawn based on the 3% limit, provide a clearer and deeper understanding of the influence of flow rate and suction pressure on the potential occurrence of cavitation, presenting good and coherent results, differing from those provided by pump manufacturers only due to the safety coefficient normally applied. The development of the $H(Q)$ curves for the centrifugal pump CR5-8 at various speeds, for different motor drive frequencies, overlaid with constant $\dot{W}$ and η curves (Figure 4.10), serves as an interesting example of how pump providers usually supply the performance information. Regarding the control and adjustment of pump performance, typical cases encountered during pump operation are presented, along with illustrative graphs developed to better understand the use of different control methods for achieving a specific operating point. In terms of pumps operating in series and parallel, graphs are developed to better understand the effects of their combination in comparison with the individual operation of these pumps, yielding coherent and satisfactory results, with slight variations only due to the different equipment used in individual tests.

In the case of turbines, especially the Pelton turbine, typical turbomachinery courses usually cover, in a more detailed, the topics of Chapter 5, where a comprehensive overview of some theoretical concepts is initially presented, followed by a concise introduction to turbine performance characteristics. Chapter 5 thus represents a practical application of the topics presented in Chapter 4, offering practical insights into the field of turbomachinery engineering education, especially due to the unique facilities available at ISEC and the interesting reproduction of phenomena on a smaller scale at UMH.

The existing experimental Pelton turbine setups at ISEC and UMH are described, including the respective measuring equipment and referring, when necessary, to some theoretical aspects regarding the calculation of various objective functions, based on the measured variables. The testing methodologies to be followed are outlined, depending on the type of curves and information to be obtained. In the case of the UMH installation, it is worth highlighting, from an educational perspective, the interesting setup and the methodology for developing, printing, and testing the various Pelton turbine models.

This approach not only improves the understanding of the underlying phenomena of their operation but also creates links with other subjects commonly found in engineering courses, such as Design, Manufacturing, and Materials. Due to the experimental nature of the work, and the significant amount of tests and adjustment parameters involved in Pelton turbine testing, a brief uncertainty analysis is performed to quantify both systematic and random errors, providing justification for the results obtained from conventional tests. From an educational perspective, the uncertainty analysis also links back to fundamental engineering courses like Statistical Methods.

The results are presented and discussed, showing generally coherent outcomes, both quantitatively and qualitatively, in line with predictions from the literature. A more detailed discussion is included to address the reasons for the discrepancies that were identified. The initial categorization of the different tests that can be performed at each institution clearly illustrates the influence of each parameter on turbine performance.

An interesting difference between the two institutions is that the UMH setup does not include a nozzle needle, leading to the classification of tests as *constant aperture* type. Despite challenges in maintaining constant values for head and flow rate, the UMH installation serves as an excellent example of the Project-Based Learning (PBL) concept, a methodology that has gained prominence in engineering education in recent years due to its effectiveness in training more responsible and capable engineers to address the complex problems of the real world. Additionally, the collaboration between both institutions provides insights into the strengths, weaknesses, opportunities, and threats that each presents, enabling the improvement of engineering education in the field.

An aspect of special interest that deserves to be mentioned is that this contribution represents one outcome of the Erasmus+ program that started in April 2023 and since then is being developed and enhanced between UMH and ISEC. The importance of practical engineering education and studies like the one presented here were highlighted at the 52nd Annual Conference of the SEFI, held at EPFL, where part of the work presented here, titled *PELTON TURBINE UNDER DIFFERENT OPERATING CONDITIONS: An Experimental Pedagogical Study*, was submitted as a paper and presented to a considerable group of researchers, professors, and industry members.

7.2 Future Development Perspectives

Due to the scope of the present work, which aimed to address both main categories of hydraulic turbomachines - pumps and turbines - and although the studies conducted already present some detail, there are aspects of each turbomachine that could be further explored and developed in the future, both from a scientific and pedagogical perspective, fields that are naturally interconnected.

Both institutions have setups for centrifugal pump testing of individual pumps that are well-equipped and properly instrumented, allowing for the control and measurement of relevant variables during pump testing. Nevertheless, while the use of traditional instruments for measuring quantities is pedagogically valuable, complementing them with automatic data acquisition systems, ideally with an integrated interactive dashboard, can reduce operator reading errors and enable faster data acquisition and potential processing, as well as a better real-time understanding of the phenomena involved during operation, particularly in obtaining operating points corresponding to cases A, B, and C outlined in Chapter 4.

In cases where a VFD is integrated, it is useful to measure the power before and after the device to quantify the power absorbed by it, as well as its efficiency. Furthermore, conducting a quantitative and qualitative analysis from an economic and energy perspective on the integration of VFDs in pumping systems, and determining whether this integration is always beneficial or not, would be of interest. The integration of torque and rotational speed measurement equipment on the shaft would also allow the power transmitted to the shaft to be determined and compared with the hydraulic power that the pump transfers to the fluid.

In the case of cavitation phenomenon evaluation tests, it would be interesting to maintain the water temperature constant throughout the entire test, something that was not always possible. Exploring the influence of water temperature on the occurrence of cavitation would also be a valuable aspect to investigate. Additionally, linking it to other engineering disciplines, the use of sensors to analyze the generated vibrations would enable more precise monitoring and detection of the onset of cavitation, allowing for a comparison with the results provided by the 3% limit.

The development of a test bench for the series and parallel association of pumps at ISEC would also be an interesting aspect to explore, both from the standpoint of developing the bench itself and from the perspective of engineering continuing education, given the significant presence of such systems in the industry.

Regarding the setups for Pelton turbine testing, both institutions also have setups that are pedagogically interesting, although they differ significantly in size, equipment, and operating principles. Given these differences, individual suggestions are made for future work at each institution.

For the setup at the LHM of ISEC, a detailed test of the Francis and Kaplan turbines, similar to the one conducted in this work on the Pelton turbine, would produce an excellent pedagogical outcome in terms of understanding the operation of the three main types of turbines used and their respective fields of application.

As discussed earlier, when calculating the absolute velocity of the jet, it is common to use a coefficient to account for losses in the injector. Therefore, it would be interesting to determine the actual jet velocity at the outlet of the injector to more accurately assess the efficiency of the turbine. Some studies on efficiency measurements on Pelton turbines using thermodynamic and acoustic methods suggest that these methods are worth exploring, as they reduce the uncertainty associated with results obtained through traditional methods.

Although the current process of measuring shaft power is pedagogically valuable, the incorporation of a load cell and an optical tachometer connected to the same acquisition board would yield more precise results on the same time basis.

At an opportune moment, such as during turbine maintenance, it would be worthwhile to conduct a detailed dimensional and geometric survey to recreate the model computationally. This would open the door to numerical simulation or even experimental analysis at different scales through dimensionless analysis.

For the setup at LHM of UMH, the incorporation of an adjustable axial needle injector would allow the turbine to be tested under a broader range of operating conditions, leading to a better understanding of the various parameters involved in its operation and the conditions that optimize turbine efficiency. Given the design component associated with Pelton turbine testing at UMH, conducting a sensitivity analysis to identify the influence of design parameters on turbine performance would be of interest. A detailed dimensional and geometric survey of the Pelton turbine at ISEC, followed by its design, manufacturing, and testing at UMH, would allow for an interesting study from the perspective of model theory through dimensionless analysis, thereby encouraging continued cooperation between the institutions.

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