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Research paper

Optimized sizing of a pumping system using the Scilab software

Dimensionamento otimizado de um sistema de bombeamento utilizando o software Scilab

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ABSTRACT: Climate change represents, today, one of the greatest challenges regarding the availability of water resources for the cultivation of food and raw materials of plant origin that the world's population so badly needs. Sugarcane, due to its high added value to produce several products and whose accumulation of sucrose is mainly due to a perfect dosage of water, brought to light the need to obtain new irrigation strategies, thus aiming at a more sustainable, efficient and economical management of water consumption in this sector. As an alternative to this challenge, drip irrigation stood out, which allowed not only a more precise control of water dosage, but also essential nutrients for the development of sugarcane. Taking advantage of this, a sugar and alcohol plant in the interior of the state of São Paulo, whose identity will be preserved, made several investments to implement this system on its farms and one of these projects will be discussed in detail about the dimensioning of the entire piping network. which involves the pumping system, from water collection to discharge into a storage tank, responsible for supplying the dripping, using a pre-dimensioned pump and respecting the specifications defined by the client for this work. In addition, the free software Scilab was used to implement the piping and pumping sizing procedure, as an extra tool, in addition to development in Excel.
Keywords: water; irrigation; drip; suction; discharge.

RESUMO: As mudanças climáticas configuram, hoje, um dos maiores desafios no que se diz respeito à disponibilidade de recursos hídricos direcionados ao cultivo de alimentos e matérias-primas de origem vegetal que a população mundial tanto necessita. A cana-de-açúcar, em razão de seu elevado valor agregado para a produção de diversos produtos e cujo acúmulo de sacarose se deve, principalmente, a uma dosagem perfeita de água, trouxe à tona a necessidade em se obter novas estratégias de irrigação, visando, assim, uma gestão mais sustentável, eficiente e econômica de consumo água nesse setor. Como alternativa a esse desafio, destacou-se a irrigação por gotejamento que permitiu não só um controle mais preciso da dosagem de água como também nutrientes essenciais para o desenvolvimento da cana-de-açúcar. Aproveitando-se disto, uma usina sucroalcooleira do interior do estado de São Paulo, cuja identidade será preservada, realizou vários investimentos para implementação desse sistema em suas fazendas e um desses projetos será abordado detalhadamente no que se diz respeito ao dimensionamento de toda a rede tubulação que envolve o sistema de bombeamento, desde a captação de água até a descarga em um tanque de armazenamento, responsável pelo abastecimento do gotejo, utilizando-se de uma bomba já pré-dimensionada e respeitando as especificações definidas pelo cliente para o presente trabalho. Além disso, contou-se com o auxílio do *software* livre Scilab para implementação do procedimento de dimensionamento da tubulação e bombeamento, como ferramenta extra, além do desenvolvimento no Excel.
Palavras-chave: água; irrigação; gotejo; sucção; recalque.

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INTRODUCTION

Sugarcane, scientifically known as *Saccharum officinarum*, constitutes one of the oldest crops, due to its high adaptability to Brazilian climate and soil, it served as the cornerstone of the colonial economy for a considerable period, with sugar being the flagship product representing Brazil's initial major agricultural and industrial wealth. This directly influenced the formation of Brazilian culture (Yogitha *et al.*, 2020).

Its high economic value and the research and development of more sustainable products and energies, sugarcane derivatives have extended far beyond molasses, sugarcane juice, cachaça, and sugar, gaining prominence in the production of fuels, such as first and second-generation hydrated ethanol and anhydrous ethanol, as well as energy from bagasse mixed with straw. Additionally, residual vinasse and filter cake have found extensive applicability in fertilizing sugarcane itself and other crops (Schmidt Filho, 2016).

The Brazilian scenario regarding the sugarcane harvest in 2022 revealed increased ethanol production until mid-June, driven by the higher price of ethanol compared to sugar, a trend not seen since 2019. As a forecast for the upcoming harvest (2023/2024), sugar production promises to yield better returns, reinstating Brazil as the world's largest sugar producer and exporter (Vital, 2023).

Such forecast stems from a greater amount of sugarcane available for milling, as the rains in 2022 provided recovery after three years of fluctuating levels well below average (Vital, 2023).

With the scarcity of rainfall and the emerging need to keep sugarcane farms irrigated, drip irrigation has emerged as an alternative, supported not only by water resource conservation but also by agricultural input economy, thus promoting maximum crop yield (Carr; Knox, 2011; Dalri *et al.*, 2008; Lamm *et al.*, 2012).

According to the Sugarcane Journal (2021), drip irrigation not only delivers an ideal volume of water and nutrients tailored to the plant's growth stage but also prevents evaporation and leaching processes, minimizes climate dependence, and enables the full utilization of 100% of arable land area.

Such a system consists of pipes suitable for the type of liquid used, emitters known as drippers, a set of motor pumps, filters, valves, pressure gauges, automation system, and fertilizer injection system. Operating at low pressures, it also offers energy savings (Esteves *et al.,* 2012).

Given the aforementioned, the present study aims to optimize the sizing of a pipeline for pumping water from a stream of the Sapucaí River to a tank built to supply water to the drip irrigation system of a sugarcane and alcohol plant, located in the municipality of São Joaquim da Barra, São Paulo state. Additionally, the material and diameters of suction and discharge pipes will be determined, respecting a desired flow rate of 200 m³/h and a limited power consumption of 25 HP, assuming the reuse of the previously installed structure.

Finally, the sizing process will be simulated using the Scilab software, aiming for greater convenience in obtaining results compared to Excel.

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THEORETICAL FRAMEWORK

Centrifugal pumps

Centrifugal pumps are machines that convert mechanical energy, supplied by a motor source, into hydraulic energy in the form of kinetic, potential, and pressure energy (Araújo; Araújo, 2019).

Among the most common applications are water supply networks, irrigation systems, industrial cooling, firefighting, water treatment, chemical pumping, etc. (Santos, 2019).

Macintyre (1986) states that one of the requirements for the operation of a centrifugal pump is the complete filling of the casing with the fluid to be pumped at startup, a phenomenon known as priming, as the centrifugal movement of the rotor with the blades propels the fluid to an area of two distinct pressures.

The low-pressure zone is responsible for suctioning particles coming from the inlet, while the high-pressure region allows the discharged material to overcome the head losses imposed by fittings and piping (Macintyre, 1986).

The transportation process begins as the fluid is drawn through the pump's nozzle, with the local gauge pressure being either positive or negative (vacuum) relative to atmospheric pressure, depending on whether it's higher or lower, respectively. Subsequently, the fluid is directed to one or more rotors, which impart kinetic energy to it, which is then converted into potential pressure energy (Fox; McDonald; Pritchard, 2014).

Similarly, fluid exit occurs through the discharge nozzle; however, the energy to be imparted to it becomes the result of the pressure difference between the pump suction and discharge. This energy is known as total dynamic head, and it is because of it that fluid elevation, pressurization, or transfer becomes possible (Fox; McDonald; Pritchard, 2014).

Pump characteristic curves

The pump characteristic curve is provided by the manufacturer using test benches specifically equipped for this purpose. These curves describe the pump's operational behavior under certain flow and head conditions (Gomes, 2017).

By utilizing this curve, we can not only analyze and select the ideal pump for each project but also verify if it is operating within the recommended tolerances, aiming to maximize the machine's efficiency and avoid future bottlenecks (Gomes, 2017).

The pump installation characteristic curve, also known as the system curve, represents the energy required per unit weight to be supplied to a fluid as a function of its flow rate. In other words, it depicts the total head corresponding to each flow rate within a certain operating range of the system (Takami, 2011).

The pump characteristic curve tends to exhibit a decreasing behavior. At the pump's startup, as the fluid begins to flow and the discharge valve opens further, the pumping capacity decreases. Conversely, the system curve displays an ascending behavior because the head loss is proportional to the square of the flow velocity, which is directly proportional to the flow rate. Thus, the head loss increases as the system flow rate increases (Lengsfeld *et al.*, 2005).

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Operating point

The operating point or working point of the pump, as shown in **Figure 1**, represents the operational condition of a pump in each system. It is determined by the intersection of the system's characteristic curve with the pump's characteristic curve (Takami, 2011).

This means that at this point, the pump is capable of providing the fluid with a head precisely equal to what the fluid needs to flow through the hydraulic system at a steady flow rate Q (Macintyre, 1986).

Figure 1. Graphical representation of the system's operating point.

Source: Baptista and Lara, 2002.

Cavitation and NPSH

Cavitation is a complex physical phenomenon of vaporization commonly encountered in hydraulic machinery systems operating with liquids, such as in the case of a centrifugal pump (Coelho, 2006).

In cavitation occurrence, there is an intense formation of vapor bubbles in the low-pressure zone of the pump, and after this pressure is restored, these vapor bubbles collapse with a highly intense mechanical micro-shock (Simões, 2020).

This phenomenon directly interferes with machine lubrication, potentially causing significant damage to the pump rotor due to increased physical stress, as well as considerable noise in the piping (Simões, 2020).

Therefore, an important parameter that assists in assessing the performance of centrifugal pumps is the NPSH (Net Positive Suction Head), which can be understood as the minimum pressure in absolute terms, in meters of water column, above the vapor pressure of the fluid, in order to prevent the formation of vapor bubbles and thus cavitation occurrence in the pump suction (Lengsfeld et al., 2005).

Following Lengsfeld *et al.* (2005), for study and definition purposes, NPSH can be divided into required and available.

The available NPSH refers to a feature of the installation in which the pump operates and the available liquid pressure on the suction side of the pump, which can be calculated through a mathematical expression (Oliveira, 2017).

On the other hand, the required NPSH represents the energy in absolute height of the liquid at the pump suction above the vapor pressure of this liquid at the pumping

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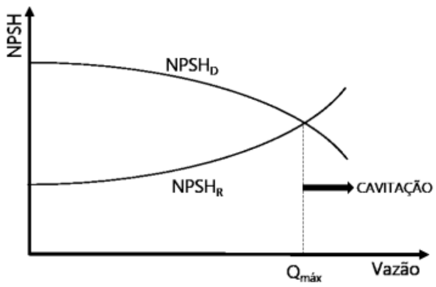
temperature, referred to the pump centerline; being a pump-specific characteristic, it must be obtained experimentally on the manufacturers' test benches (Oliveira, 2017).

Thus, in practical design terms, once the available energy equals or exceeds the required NPSH values, the occurrence of liquid vaporization is ensured to be avoided, consequently preventing cavitation-related issues. Typically, the available NPSH should exceed the required NPSH by at least 10 to 15% and never be less than 0.5 m (Lengsfeld *et al.*, 2005).

In **Figure 2**, it is possible to observe and thus better understand the effect of flow rate on NPSH.

Figure 2. Representation of the effect of flow rate on NPSH.



Source: Oliveira, 2017.

By analyzing **Figure 2**, it is important to note that the intersection point between the available NPSH and the required NPSH corresponds to the maximum flow rate allowed by the system to prevent cavitation from occurring.

Economic diameter

One of the main objectives of any project is to meet the needs of what has been proposed while incurring the lowest possible cost, whether it be for installation or annual maintenance (Gama; Souza; Callado, 2019).

Therefore, to determine the pipeline diameter that meets such requirement in conveyance systems, it is necessary to take into consideration certain parameters, and primarily, the availability of commercial pipe diameters for selection (Corrêa, 2016).

Additionally, in this context, it is necessary to bear in mind the clear separation of the cost of the pipeline to be installed, commonly referred to as capex, which includes the fixed costs or depreciation of the initial investment, and the operational cost of the system, commonly referred to as opex, which encompasses the energy expended in pumping the fluid (Corrêa, 2016). Thus, for determining the pipe diameter that meets such requirement in conveyance systems, certain parameters need to be taken into consideration, and, importantly, the availability of commercial pipe diameters for selection (Corrêa, 2016).

It's worth mentioning that the capex increases proportionally with diameter, as larger diameters require a greater amount of material; meanwhile, opex decreases with increasing pipe diameter due to reduced fluid flow resistance.

Thus, when fixed costs are added to operational costs, they will yield a minimum value, which will be responsible for minimizing total costs, constituting what is known as the economic diameter (Denn, 1980). **Figure 3** provides a visual representation of the definition of economic diameter.

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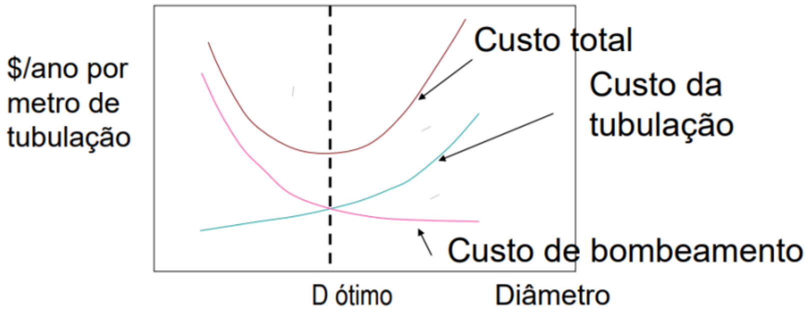
Figure 3. Visual representation of the economic diameter (optimum)



Source: Corrêa, 2016.

Scilab Software

In recent decades, computational simulations have gained prominence in the industrial sector as they are tools that enable, with the aid of mathematical techniques, the analysis and control of the operation of one or more specific processes (Freitas Filho, 2008).

Thus, the Scilab software is considered one of the most widely used computational tools in this environment, whose development dates back to the 1990s by researchers from INRIA (Institut National de Recherche en Informatique et en Automatic) and ENPC (École des Ponts ParisTech) (Silva; Cunha, 2006).

As of the present moment, this software is free and is characterized by its high performance in solving mathematical problems with a simpler language (Silva; Cunha, 2006).

METHODOLOGICAL PROCEDURES

Base equation

For calculating the average flow velocity in [m/s] inside a cylindrical pipe, **Equation 1** was employed, derived from the Continuity Equation for homogeneous, incompressible fluids with constant properties.

$$V = \frac{4 * Q/3600}{\pi * D^2} \quad (1)$$

Given that Q is the flow rate of the fluid in [m³/h] and D is the internal diameter of the pipe in [m].

To calculate the Reynolds number, we used the definition proposed by Osborne Reynolds at the end of the 19th century. This definition is represented by Equation 2 and is interpreted as the ratio of inertial forces (ρv) to viscous forces (μ/L).

$$Re = \frac{v * D}{\vartheta} \quad (2)$$

Given that ϑ represents the kinematic viscosity of the fluid in [m²/s].

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It is important to note that **Equations 1** and **2** were taken from the book "Transport Phenomena" by Bird, Stewart, and Lightfoot (2004).

To calculate the friction factor, we used Equation 3, proposed by Serghides (1984), which provides a more precise optimization compared to the Colebrook-White Equation.

$$F = \left[A - \frac{(B-A)^2}{A - 2 * B + C}\right]^{-2} \tag{3}$$

$$A = -2 * \log_{10}\left(\frac{\varepsilon/D}{3{,}7} + \frac{12}{Re}\right) \tag{3ª}$$

$$B = -2 * \log_{10}\left(\frac{\varepsilon/D}{3{,}7} + \frac{2{,}51 * A}{Re}\right) \tag{3b}$$

$$C = -2 * \log_{10}\left(\frac{\varepsilon/D}{3{,}7} + \frac{2{,}51 * B}{Re}\right) \tag{3c}$$

For the calculations of localized head loss and abrupt narrowing, we used **Equations 4** and **5**.

$$Hp_L = K * \frac{v^2}{2 * g} \tag{4}$$

$$K = \frac{4}{9} * \left(1 - \frac{B}{A}\right) \tag{5}$$

Let K be a dimensionless coefficient obtained experimentally, g the acceleration due to gravity in [m/s²], and B/A the ratio of the smaller area to the larger area, respectively.

To calculate the distributed pressure loss, we used **Equation 6**, proposed by Darcy-Weisbach (1845).

$$Hp_D = f * \frac{L}{D} * \frac{v^2}{2 * g} \tag{6}$$

Let f be the friction factor and L the total length of the straight sections in meters.

To verify the occurrence of pump cavitation based on the absolute static pressure, we used **Equation 7**, proposed by Simões (2020).

$$Pe_{abs} = P_{atm} - \frac{\gamma}{1000}\left(\frac{v^2}{2g} + Hgeo + Hp_L + Hp_D\right) \tag{7}$$

Given that P_{atm} is the atmospheric pressure in the municipality in [kgf/cm²], γ is the specific weight of the fluid in [kgf/dm³], and Hgeo is the geometric height in [m].

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To calculate the available NPSH (Net Positive Suction Head) in [m] during the design phase, we used **Equation 8**.

$$\mathrm{NPSH_d} = \frac{\mathrm{P_{rs}} + \mathrm{P_{atm}} - \mathrm{pv}}{\gamma} * 10 \pm \mathrm{Hgeo} - \mathrm{Hp} \qquad (8)$$

Let $\mathrm{P_{rs}}$ be the pressure in the suction reservoir in [kgf/cm²], and pv the vapor pressure of the fluid in [kgf/cm²].

For the calculation of suction head (Hs) and discharge head (Hd), we used **Equations 9** and **10**.

$$\mathrm{Hs} = \pm\mathrm{Hgeo} + \frac{\mathrm{P_{rs}}}{\gamma} - \mathrm{Hps} + \frac{\mathrm{vrs}^2}{2\mathrm{g}} \qquad (9)$$

$$\mathrm{Hd} = \mathrm{Hgeo} + \frac{\mathrm{P_{rd}}}{\gamma} + \mathrm{Hpd} + \frac{\mathrm{vrd}^2}{2\mathrm{g}} \qquad (10)$$

Let $\mathrm{P_{rd}}$ be the discharge reservoir pressure in [kgf/cm²] and $\mathrm{vrs/vrd}$ the average flow velocities in the suction and discharge pipes in [m/s].

For the calculation of the total head, we used **Equation 11**.

$$\mathrm{H} = \mathrm{Hd} - \mathrm{Hs} \qquad (11)$$

Finally, we used **Equation 12** to calculate the power consumed by the pump.

$$\mathrm{P} = \frac{\gamma * \mathrm{Q} * \mathrm{H}}{270 * \eta} \qquad (12)$$

Let γ represent the specific weight of the fluid in [kgf/dm³], Q the fluid flow rate in [m³/h], H the total manometric head in [m], and η the efficiency obtained from the supplier's pump catalog.

It is important to note that **Equations 6**, **8**, **9**, **10**, **11**, and **12** were all taken from the "Training Manual" by the pump manufacturer KSB, published by Lengsfeld *et al.* (2005).

Design data: motor

We chose to purchase the motor manufactured by WEG, a general-purpose model known as the W22 with premium IR3 efficiency. It operates at 1750 rpm with a power output of 25 HP, meeting the maximum limit set by the client to accommodate the existing structure. The motor is three-phase, with voltage options of 220/380/440 V, 4 poles, and it comes with feet but without flanges.

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Design data: terrain

For suction, we measured 7.5 meters of straight sections and 4.5 meters of geometric height, both with clearance, also estimating the presence of one 90° bend and one butterfly valve in the line.

For the pressure head, we measured 144 meters of straight sections and 5 meters of geometric height, both with clearance. We also estimated the presence of 4 90° bends and two valves, one butterfly-type and the other check valve, in the pipeline.

Additionally, atmospheric pressure for the altitude of São Joaquim da Barra municipality, equivalent to 625 m, was defined using **Table 1**, according to the Google Earth Pro program for the given coordinates of the power plant in question.

Table 1. Variation of atmospheric pressure according to altitude**.**

Altitude [m]	Pressure [mmHg]	Altitude [m].	Pressure [mmHg]
0	760	1000	674
200	742	1200	658
400	724	1400	642
600	707	1600	627
800	690	1800	612

Source: Galvani, 2018.

Design data: fluid

As a fluid, we consider liquid mineral water. Among its properties, at a temperature of 30°C, as noted by Houghtalen, Hwang, and Akan (2012), the mass and specific weight are 996 kg/m³ and 9771 N/m³, respectively. The vapor pressure is equivalent to 0.041831 atm, and the kinematic viscosity is 0.800 x 10^{-6} m²/s.

Iterative procedure

As a calculation procedure, we initially employed an iterative strategy divided into suction, discharge, and verification of consumed power.

For suction, we used the criteria of NPSH (required and available) and cavitation concepts for analysis, as previously presented, relating them to flow through a graph, as shown in **Figure 2**.

To obtain it, we defined an external diameter of the pipe in inches for a specific flow range, whose values in millimeters, as well as the wall thickness deduction for each, were extracted from **Table 2** following the standard typically adopted by the Plant, according to the chosen material.

Table 2. Steel pipes according to ANSI B.36.10 and B.36.19 standards.

D [pol]	De [mm]	5S [mm]	10S [mm]	40/40S [mm]
4	114,30	2,11	3,05	6,02
6	168,28	2,77	3,40	7,11
8	219,08	2,77	3,76	8,18
10	273,05	3,40	4,19	9,27

Source: Vedatech, 2012.

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From this estimate, parameters such as velocity (**Equation 1**), Reynolds number (**Equation 2**), friction factor (**Equation 3**), localized and distributed head losses (**Equations 4 and 6**) were calculated to obtain the suction head loss (**Equation 9**).

As a note, it's worth mentioning that the calculation of localized head loss was performed using the method provided by the empirical coefficient K, whose values are listed in **Table 3**.

Table 3. Values of the coefficient K obtained experimentally.

Accessory/Valve	K	Accessory/Valve	K
90-degree long radius bend curve.	0,40	Globe valve	10,00
90-degree short radius bend	1,50	Gate valve	0,20
45-degree long radius bend	0,20	Angle valve	5,00
45-degree short radius bend	0,40	Straight-through tee	0,60
30-degree bend	0,10	Side outlet tee	1,30
Foot valve	1,75	Double outlet tee	1,80
Check valve	2,50	Strainer	0,40
Butterfly valve	0,30	Outlet	0,50

Source: Lengsfeld *et al.*, 2005.

Additionally, we accounted for pipe reductions between the kicked diameter and the pump nozzle diameter, using **Equation 5**.

With all these values at hand, we calculated, for each respective flow rate, the available NPSH values, enabling the construction of curves according to the assumed diameter value.

For the discharge, the methodology for determining the head loss of the pipeline follows the same steps as the suction, however, as it is an interdependent calculation, it was necessary to set up a suction pipeline to analyze the pipe diameters.

Finally, by calculating the total manometric head using **Equation 11**, we consulted the pump performance curve provided by the manufacturer, enabling us to verify the power value using **Equation 12**.

Simulation: Scilab software

For greater calculation convenience in the sizing described above, we turned to the open-source software Scilab to develop a code that would provide suction and discharge pipe diameters as an alternative to using VBA tools via Excel.

Therefore, it was necessary to input the data, in txt format, from ANSI B.36.10 and B.36.19 standards, as previously shown in **Table 2**, as well as the information provided by the manufacturer, which in this case was extracted from the electronic catalog, exactly as mentioned earlier.

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RESULTS AND DISCUSSIONS

Pump selection

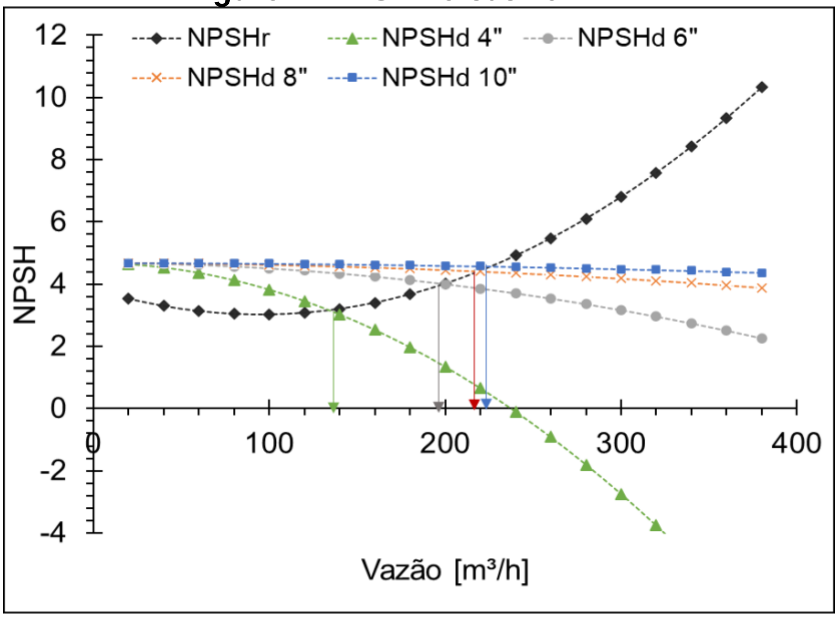
Based on the terrain conditions, we pre-sized a centrifugal pump operating with positive suction or "non-submerged" conditions, meaning the pump is positioned above the suction reservoir.

As a result, we selected a centrifugal pump model ITAP 125-260 manufactured by IMBIL with a rotor diameter of 228 mm and a rotation speed of 1750 rpm, whose line is commonly applied in power plants, distilleries, and irrigation systems.

Suction

After applying the described methodological procedure for suction, we obtained **Figure 4**, which relates NPSH to the limit flow rate for cavitation occurrence.

Figure 4. NPSH versus flow.



Source: By authors, 2024.

From the analysis of **Figure 4**, it is observed that for a suction diameter of 4", the limit flow rate to prevent cavitation is approximately 137 m³/h, which is well below the desired 200 m³/h.

Additionally, for flow rates above 240 m³/h, the NPSH is associated with negative values, which would be impractical.

Continuing with the analysis, for a 6" diameter, the maximum flow rate becomes exactly the desired flow rate, which also turns out to be not very feasible, given that projects, for the most part, do not correspond 100% to reality. Consequently, any fluctuation in this case would be sufficient to compromise it.

For the 8" and 10" diameters, there is already an assurance of a working flow rate close to 200 m³/h, even with a slight margin. However, if the client wishes to increase this flow rate in the future, another feasibility study would be necessary.

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To overcome this situation, a good alternative would be the association of pumps in parallel, as it is not always possible to find a pump on the market that operates at the desired operating point or close to the point of maximum efficiency, making these associations lead to better operational performance of the system (Gaio; Monteiro, 2005).

For a parallel association, there is promotion of fluid discharge to a single pipeline, so that for each total pumping head, the flow rate will result from the sum of the individual flow rates of each one (Denículi, 2005; Gaio; Monteiro, 2005; Azevedo Netto *et al.*, 1998).

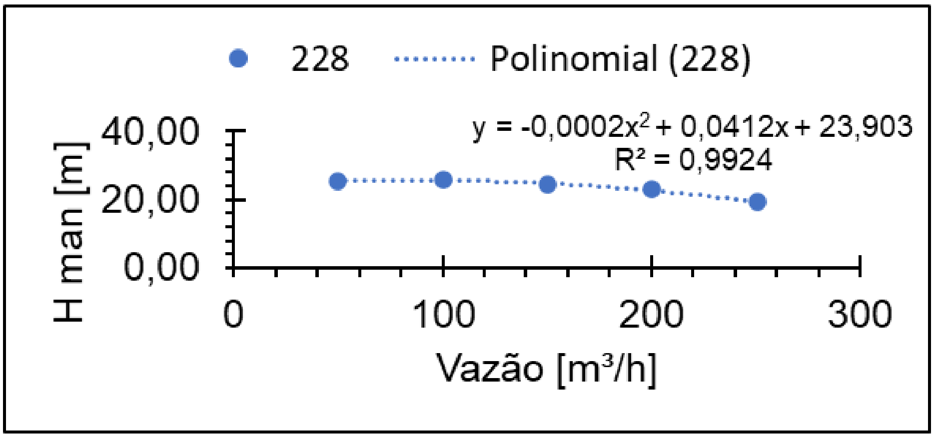
Therefore, considering that the desired flow rate remains at 200 m³/h and the diameters of 8" and 10" meet this requirement, and taking into account the cost of piping, it is more feasible to work with 8" in the suction line.

Discharge

In order to obtain the respective operating points for each of the diameters under analysis, it was necessary to define the pump curve for the 228 mm rotor using the data provided by the manufacturer through the electronic catalog available at https://ce.imbil.com.br/open.do?sys=IMB&action=openform&formID=464570603.

For this purpose, since there was no option to export the data from the curve provided by the manufacturer, we defined 5 points spaced at approximately 50 m³/h of flow and applied a second-order polynomial fit, the results of which are depicted in **Figure 5**.

Figure 5. Polynomial adjustment to obtain the pump curve**.**



Source: By authors, 2024.

With this accomplished, since the resulting coefficient of determination was very close to 1, it was possible to proceed, without the need for numerical methods, with the calculation procedures as described for the discharge for each of the fixed suction diameters.

As a result, we obtained different operating points for the chosen discharge diameters, which were 6", 8", and 10", according to their respective CCI, installation characteristic curves, as depicted in **Figure 6**.

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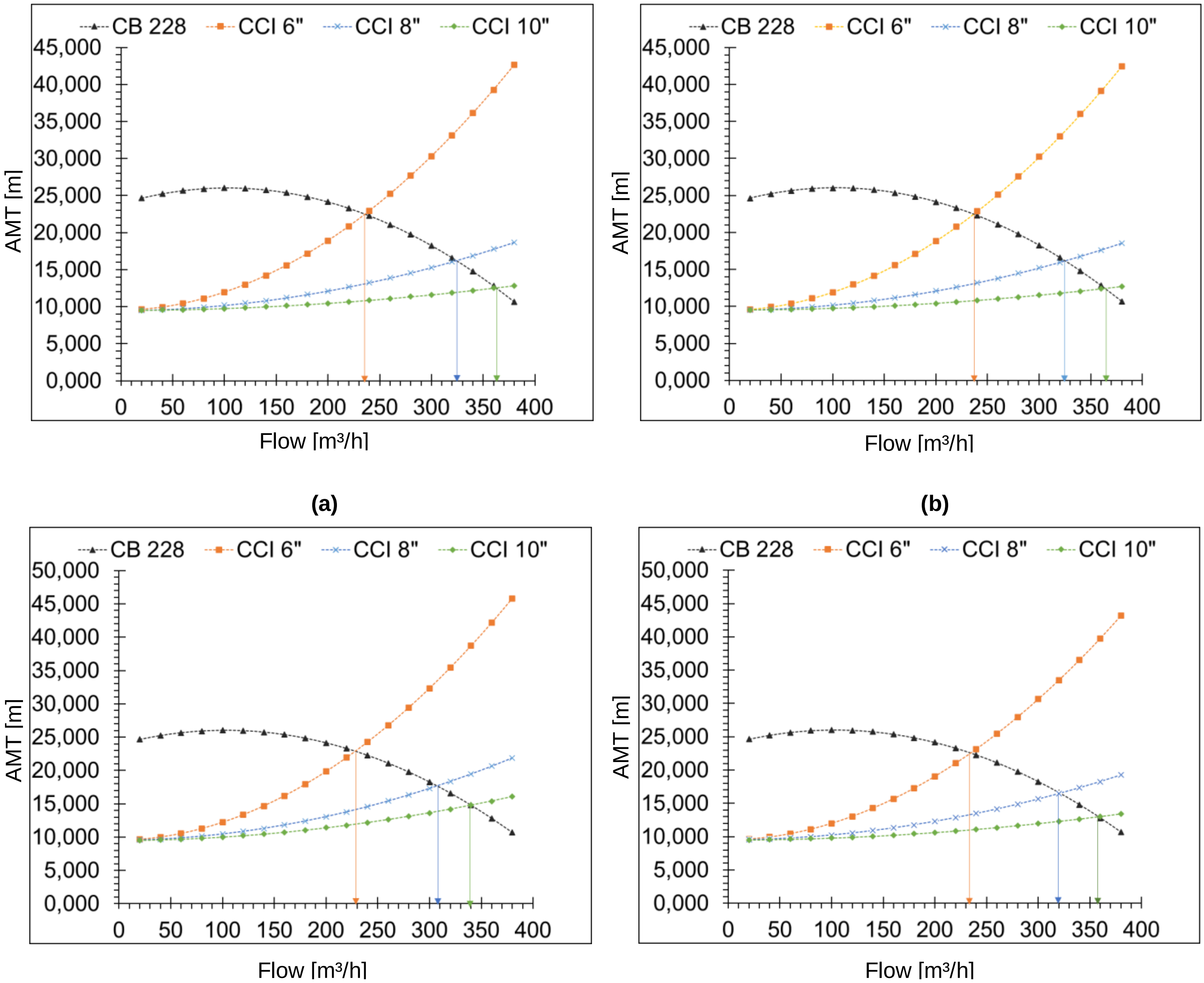
Figure 6. Operating points obtained by fixing a 4" suction **(a),** 6" **(b),** 8" **(c),** and 10" **(d).**



(a) (b) (c) (d)

Source: By authors, 2024.

From the analysis of **Figure 6(a)**, the operating flow rates in the discharge determined for 6”, 8”, and 10” were, respectively, approximately equivalent to values of 230 m³/h, 310 m³/h, and 340 m³/h.

Through analysis of **Figure 6(b)**, the operational flow rates in the discharge determined for 6", 8", and 10" were, respectively, close to 231 m³/h, 320 m³/h, and 359 m³/h.

Finally, through the analyses of **Figure 6(c)** and **Figure 6(d)**, it was noticeable that the operational flows in the discharge determined for 6”, 8”, and 10” assumed values very close to those described for **Figure 6(b)**, fluctuating around 3 m³/h to 5 m³/h more from one to another.

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In practical design terms, as recommended by the most experienced engineers at the Plant, an error of approximately 30 to 50 m³/h should be accounted for in this flow rate, both in excess and deficiency. Therefore, considering overall performance, practically all 12 working points obtained meet the desired flow rate of 200 m³/h requested by the client, disregarding what has been noted for suction diameters.

Therefore, in order to determine the optimal diameter for this sizing, we relied on the concept of economic diameter to make the final decision.

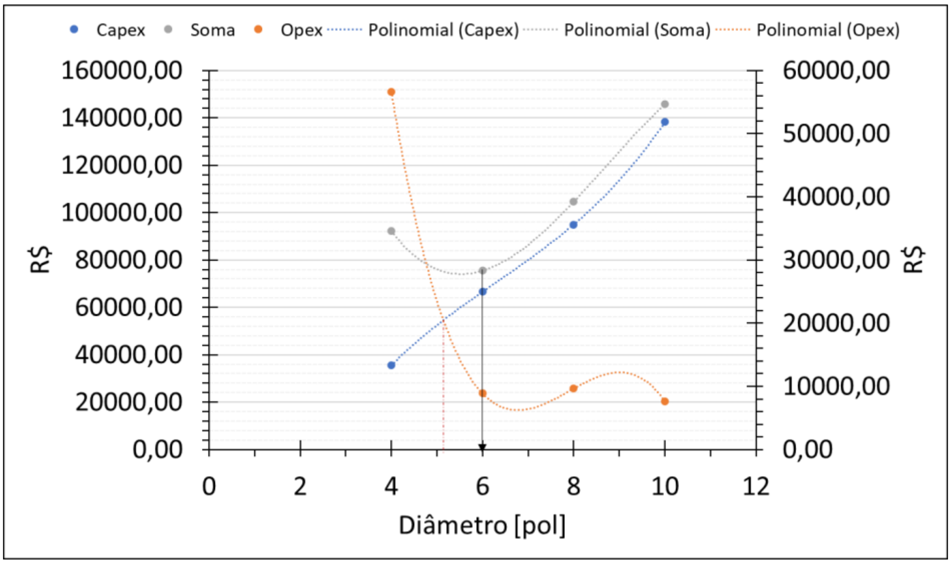
Economic diameter

To calculate the optimal diameter, it was necessary to determine the capital expenditure (capex), operational expenditure (opex), and the sum of both for constructing the analysis graph according to the chosen methodology.

For such determination, we consulted the values registered in the Plant's supplier system, which will also not be disclosed in this paper due to industrial confidentiality reasons, of piping, accessories, and valves surveyed in the sizing, all according to the norms pre-established by them for the defined material and thickness, as well as the average expenditure on labor and energy in projects similar to this one.

In the end, all results were compiled graphically as shown in **Figure 7**.

Figure 7. Economic diameter analysis of the project.



Source: By authors, 2024.

From the analysis of **Figure 7**, following the application of the described method for obtaining the economic diameter, as a result, the lowest point of the sum line, that is, the most economical value for the project, returned a discharge diameter equivalent to 6".

Thus, combining the analysis conducted for suction, discharge, and economic diameter, we opted for the value of 8" for suction due to its lower cost for the project, since both 8" and 10" satisfied the imposed condition for NPSH in order to avoid cavitation.

As for the discharge, we defined 6" as the optimal project diameter, with a working point equivalent to approximately 235 m³/h, satisfying the requirement of 200 m³/h and still yielding a proven cost savings after economic diameter analysis.

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Furthermore, despite controversy in practice regarding this concept, as a good design practice, we achieved a higher design velocity for discharge compared to suction, usually recommended when feasible due to the reduction in kinetic energy at the pump inlet, in other words, the decrease in head loss at suction.

Power verification

Once the pipeline diameters for suction and discharge were determined, we simply needed to verify the power consumption limit for the project, which was set at 25 HP.

To do so, we again consulted the electronic catalog to obtain the pump efficiency for the flow rate and total manometric head values found, resulting in a value close to 80%, ultimately enabling the application of **Equation 12**.

As a result, we achieved approximately 17 horsepower for the conducted sizing, a value with a margin of about 23%, which is considered acceptable. Additionally, it's worth noting that in the motor's manufacturing process, the established power is approximately 1.5 times higher than the nominal value (25 horsepower), serving as a safety standard by the manufacturer.

Sizing using the open-source software Scilab.

In order to optimize the process implemented in Excel, we first simulated the suction methodological procedure, obtaining the values for NPSH and the corresponding flow rate for each suction diameter, while respecting the criterion of cavitation avoidance.

We applied the same mathematical procedure to calculate the discharge, thus finding the respective operating points for each diameter. The obtained data is presented in **Tables 4 to 7**.

Table 4. Values obtained for required NPSH and available NPSH in relation to flow rate.

Tube	Flow rate [m³/h]	NPSHr	NPSHd
4	100	3,06	3,41
6	180	3,78	4,06
8	200	4,15	4,46
10	200	4,15	4,60

Table 5. Operating points determined by fixing the suction diameter at 6".

Tube	Flow rate [m³/h]	Ham_f	Hman
6	240	20,23	18,51
8	240	20,23	18,51
10	360	8,39	7,90

Table 6. Operating points determined by fixing the suction diameter at 8".

Tube	Flow rate [m³/h]	Ham_f	Hman
6	220	21,57	19,38
8	300	15,13	13,75
10	340	10,82	10,61

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Table 7. Operating points determined by fixing the suction diameter at 10".

Tube	Flow rate [m³/h]	Ham_f	Hman
6	220	21,57	20,21
8	300	15,13	14,58
10	320	13,06	11,16

Source: By authors, 2024.

By analyzing **Tables 4** to **7**, we observe that the most viable choices for the suction and discharge pipe diameters were indeed, as observed via Excel, 8" and 6", respectively.

Therefore, we can assert that the results obtained in the sizing using Scilab were consistent with the data found using the Excel software. However, the numerical responses were much more practical and effective, as the reliance on graphical analysis was eliminated, and mainly because it applies a free and accessible tool to everyone.

CONCLUSION

We were able to dimension a water pumping system from a stream of the Sapucaí River to the supply tank of the drip irrigation system of a sugarcane ethanol plant, respecting the desired flow rate of 200 m³/h and a consumed power of 25 HP.

Based on the obtained results, to prevent liquid vaporization in the pipeline, or in other words, cavitation, we determined that the suction diameter should be 8 inches. With a flow rate of 200 m³/s, this provides an available NPSH greater than the required NPSH, thus preventing future bottlenecks.

Regarding the discharge, we found that the ideal diameter to meet the desired flow rate is 6 inches, respecting the operating point and the economic analysis.

In pursuit of practicality in the sizing process, the use of Scilab was essential for comparing the results obtained in Excel, facilitating the study and the selection of the piping to be used in the pumping project. Results returned in matrix form are much faster and visually clearer compared to graphical analysis methods.

It is worth noting that this study was of great importance for applying knowledge related to centrifugal pumps and conveyance systems for irrigation, and it significantly enhanced real-world insights into economic strategies in industrial projects.

Finally, as a suggestion for future research, increasing the flow rate while excluding the replacement of the current pump is proposed. To achieve this, it is recommended to start by paralleling the same model of the current pump, as mentioned in the results, and also considering the installation of a frequency inverter or a control valve, along with a comparative analysis between the two.

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