# DEVELOPMENT OF A HYDRAULIC FLOW PUMP TEST BENCH

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***Abstract.*** *The paper presents a new design of a test bench for hydraulic flow pumps that was developed and assembled in the laboratory of Flow Machines "Prof. Dimer Benatti" at the Mauá Engineering School. The bench has as purpose of exploring the constructive details of pump impellers, such as blade form, blade inlet and outlet angles, superficial finishing and dimensional stability, to evaluate the impact of them in the global system efficiency. Basically the test equipment is formed by an electrical motor supported through its shaft by two carriers bearing that provide, to the motor body, a condition of free rotation around its shaft. Coupled to the motor body there is a reaction arm with his end connected to a load cell gage that makes possible the measure of the resultant force that maintains the motor body static. The pump and motor shafts are interlinked with a coupling that permits the correction of small misalignments between both and their speeds can be controlled by a variable frequency drive. The pump head capacity is measured by a pressure differential sensor. A similar sensor is also used to measure the flow by the applications of an orifice plate. The pump performance analysis is made by measurements of the power absorbed, the flow, speed and head, and their characteristics curves. From them will be possible to identify the influence of impeller modifications on the pump efficiency.*

***Keywords****: hydraulic pumps, test bench, flow machines, impeller design.*

## 1. INTRODUCTION

The energy is one of the most important problems of the world. We have to save energy because of the limited resources that, independent of its font, cause significant environmental pollution. On the other hand, the meaning of energy saving is lower energy consumption. This implies that the same quantity of goods and services must be produced with lower energy or more goods and services must be produced with the same quantity of energy. By doing so, competition is increased in national and international areas. Impeller blade number, impeller blade inlet and outlet angle, impeller outlet diameter etc., are significant design parameters affecting pump performance and energy consumption in pumps. The evaluation of these parameters and therefore the pump performance, in many cases, dashes in the need of tests in special benches.

This paper have as purpose to present a new design of a test bench for hydraulic flow pumps that was developed and assembled in the Mauá Engineering School.

The test was building to aid the laboratory of Flow Machines "Prof. Dimer Benatti" in the development of new technologies and supports the pump makers company's needs.

The main parts (carrier bearing, supports, etc) were designed and manufactured at the Mauá Engineering School work shops. The standard equipments (cell gage, motor, pressure sensor, etc) were bought or donated from partners suppliers.

## 2. THE BENCH DESIGN

The boundary operation condition for the test bench and also used in the design were:

- Maximum power for the electric motor: 10 kW. This specification was considered due to the electrical facilities of the laboratory;
- Speed range between 1000 to 4000 rpm. This range cover the conventional work speed for hydraulic pumps;

Figure 2.1 illustrates the main conception of the test bench.

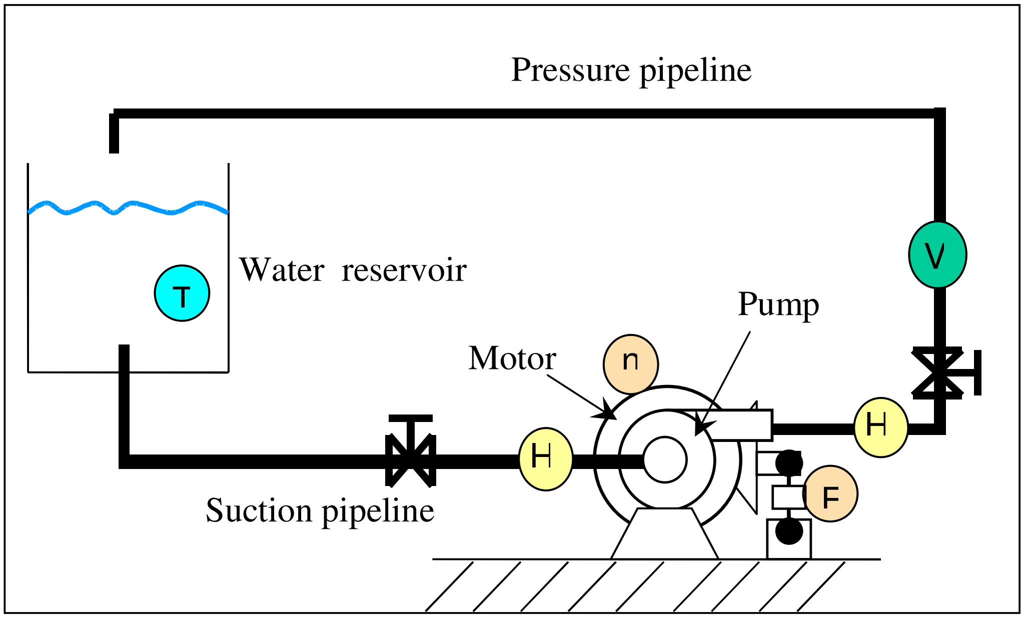


Figure 2.1 – Test bench main design. The balloon represents: V – flow rate, H – Pressure, F – Force, N – Motor speed and T – Fluid temperature

Another design consideration was that the water reservoir should be moved in the up/down direction. It makes possible to operate the pump in the suction mode condition accord to figure 2.2.

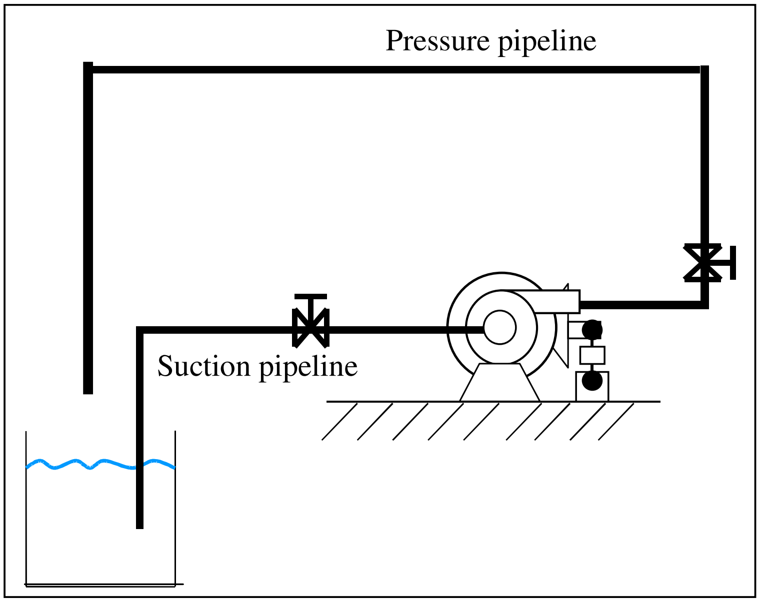


Figure 2.2 – Test bench in the pump suction mode condition

### 2.1. BENCH MEASURE EQUIPMENTS

The differential pressure in suction and pressure pipelines is measured by pressure transducers showed in the figure 2.1 by "H". In this way it is possible to measure the head pump capacity.

Coupled to the motor body there is a reaction arm with its end connected to a load cell gage that makes possible the measure of the resultant force that maintains the motor body static. It is showed in figure 2.1 by "F".

The power absorbed by the pump is calculated by the force "F", measured by the cell load, the angular speed "N", adjusted by a variable frequency drive and reaction arm length "B" (figure 2.1.1) through the equation 2.1.

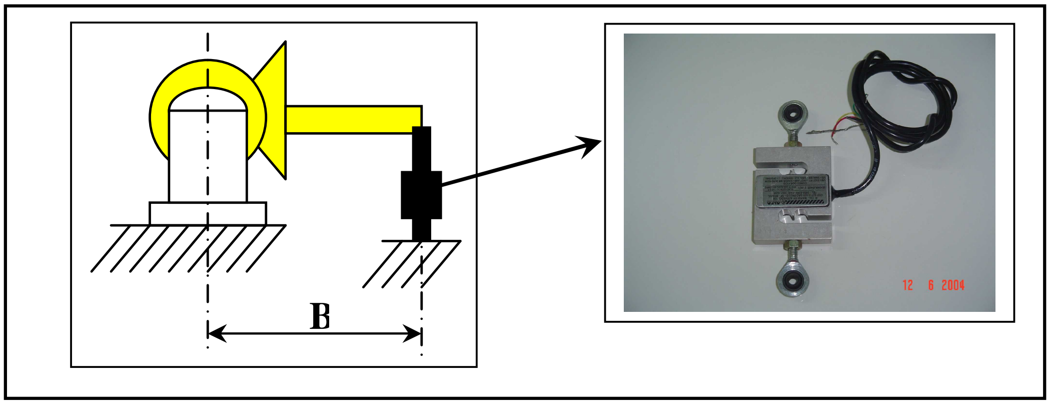


**Figure 2.1.1 – Cell gage**

$$P = F\,\frac{n.2.\pi.b}{60} \tag{2.1.1}$$

The volumetric flow rate, represented in the figure 1 as "V" is measured by an orifice plate connected to a differential pressure gage.

### 2.2. CARRIER BEARING

Two carrier bearings were designed to resist to the static and dynamic efforts of the system, allowing the electric motor a condition of free rotation around its shaft.

The applied load was calculated according to the model represented in the figure 2.2.1, where the force C is the sum of the motor and the reaction arm weight.

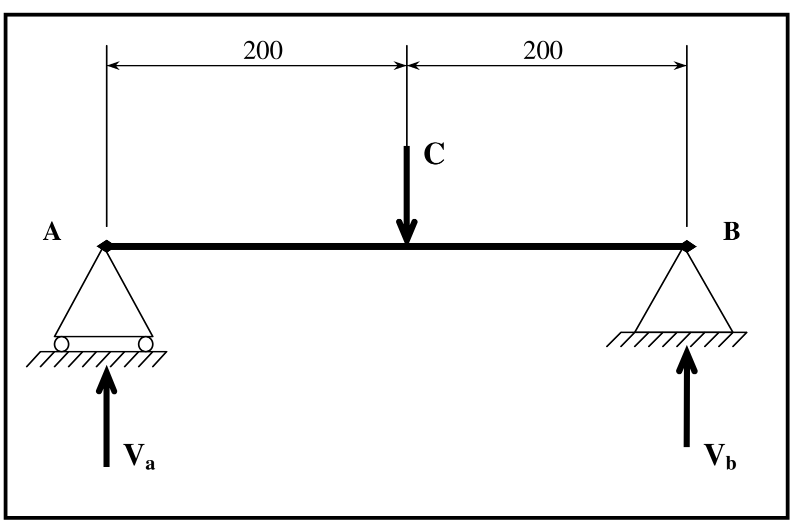


**Figure 2.2.1 – Carrier bearing loading model**

Considering the weight of motor 520N and the weight of reaction arm 100N, we get:

$$V_A = 310N \qquad V_B = 310N$$

The safety factor Cs considered in the design of the carrier bearing was:

$$Cs = 2{,}0$$

Then,

$$F_{aplicada} = 620N$$

The efforts applied in the carrier bearings were checked with a model generated in the software Mechanical Desktop, as it can be seen in the figures 2.2.2 and 2.2.3. The results indicated an appropriate design.

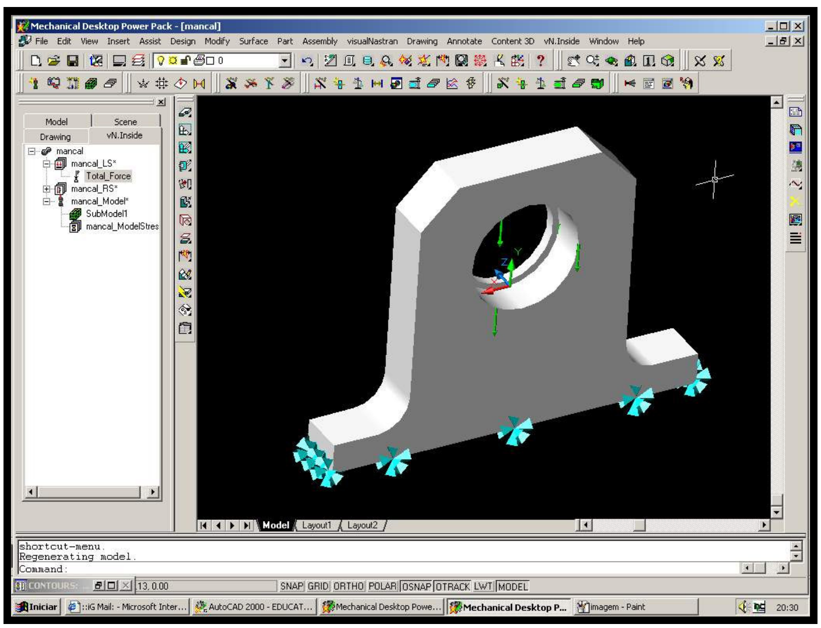


Figure 2.2.2 – Carrier bearings model

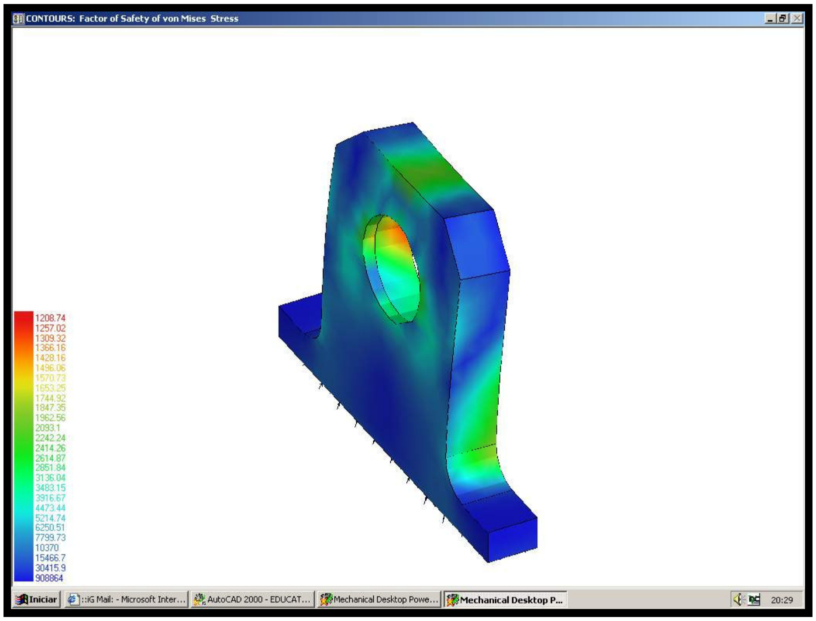


Figure 2.2.3 – Carrier bearing stress analysis.

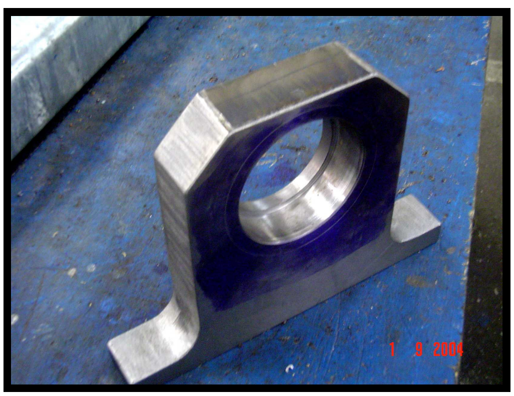

Figure 2.2.4 – Carrier bearings manufactured.

## 2.3. BEARINGS

The bearings used in the carrier bearings were designed to a life of 20.000 hours of work. The axial efforts were ignored.

Through the efforts calculated from the equations 2.2.7 could be checked the bearing life according to reference 4 (equation 2.3.1).

$$L_{10} = (\frac{C}{P})^{\rho} \quad (2.3.1)$$

$$L_{10} = 289890.h$$

## 2.4. COUPLING

The pump and motor shafts are interlinked by a coupling that permits the correction of small misalignments between both.

The coupling was selected to support the maximum torque generated by the electrical motor (equation 2.4.1).

$$Mt = 7162.\frac{P}{n} \tag{2.4.1}$$

$$Mt = 71{,}6.N.m$$

## 2.5. ELECTRIC MOTOR

For the test bench it was chosen an electric motor 220V tension and power of 10 kW due to the laboratory facilities. The main motor characteristic is a double shaft where the carriers bearing are fitted (figure 2.5.1). This configuration permits a condition of free rotation around of its shaft.

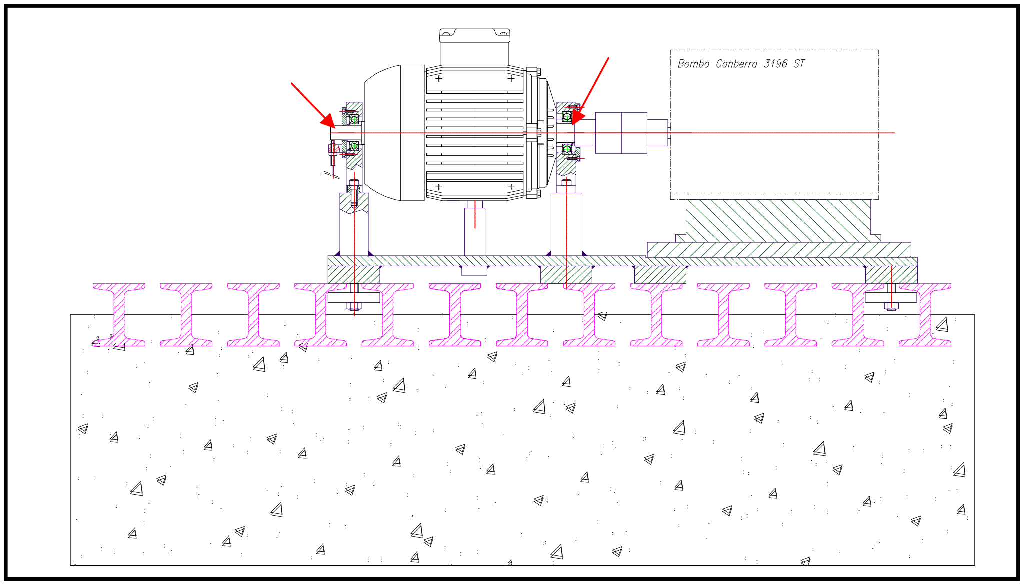


Figure 2.5.1 – Electric motor assembled. The arrows indicate the double shaft.

## VARIABLE FREQUENCY DRIVE

The bench speed and therefore the pump speed is controlled and adjusted between 1000 to 4000 rpm by a variable frequency driver.


Figure 2.6.1 – Variable frequency driver

## 3. BENCH APPLICATION

Nowadays the test bench is used for the experimental tests for the master degree works development from the two first authors of this paper. Those master programs deals with the development of pump rotors with its focus on pump efficiency.

## 4. ACKNOWLEDGEMENTS

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