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INTERMITTENT MECHANISM OF SPEED GOVERNING SYSTEM'S PRESSURIZATION PUMPS ANALYSIS

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Abstract. *This article studies the dynamic behaviour of volumetric pumps, hydraulic flow and schemes of the speed governing system from Itaipu hydroelectric power plant machines. It is motivated for the need to update this system, optimization of maintenance times and reduction of wear and casualty in the pressurization subsystem components. The present text has the objective to propose improvement measures looking to reduce financial and energetic resources consumption within all the power plant. The methodology to develop the study consists in use the available data recorded in the supervisory system related to the work rate of the pressurization pumps. Based on the results obtained it is possible to specify improvement measures based on operational parameters to optimize the system.*

Keywords: *hydraulic systems, fluid mechanics, maintenance engineering, pressure vessels, power amplification*

1. INTRODUCTION

Hydraulic systems are extensively used in industries, whenever exists need for power amplification or precise position control. Fluid power systems have great power capability, movement inversion and versatility, which are characteristics demanded for several applications.

This work is directed to hydraulic systems that control hydraulic turbines for energy generation, acquaintance as speed governor. The speed governor is responsible to sustain the angular speed of the generating unit, which ranges due to electrical system's load deviation. Constant speed operation is necessary to maintain the frequency of the electrical system within acceptable limits.

Based on the pressurizations intermission time calculus, developed by an implemented algorithm, it is possible to develop the main analysis of the work. The computed results appointed the speed governor irregular operation, accordingly to the IEEE Std 1207, in its pressurization subsystem, even though the results are acceptable for the power plant operators. The intermission time in few units is below 10 minutes, which are a normative recommendation for larger turbines. This leads to high wear in the pump's components and energy consumption. To reduce this consequences, this study develops the specification of common industrial hydraulic components, looking to achieve the normative recommendations.

1.1 Motivation

Due to the long operation time of the speed governor (almost thirty years), the vast experience of the staff involved, high supply time for spare materials, high performance required by the maintenance methodology and the failures in the pressurization subsystem that occurred during the power plant operation, it is plausible to search for new solutions to improve the system's quality. The speed governor of Itaipu is an excellent equipment, fulfils its function safely and reliably, but there is several changes in the technology concepts in governor systems for hydraulic turbines since the operation start of Itaipu. These changes encourages the study to make the operation of the speed governor even better, looking to increase maintainability and reliability in its operation.

2. THEORETICAL ANALYSIS

2.1 Speed governor system

One common way to generate electric energy is the rotation of a synchronous machine, which is responsible to convert the mechanical energy from the driver and requires constant speed to produce an acceptable range of frequency for the consumers (de Souza *et al.*, 1999). The drivers for these generators are usually turbines, whether they uses steam, water or air as source of energy. Therefore, these machines are susceptible to deviations in the electric load, caused by the demand on the electric system and requires a speed governor to control the frequency generated.

This work studies the speed governor for Francis turbines, which actuates in the section of water entrance in order to promote its control. The wicket gate controls the water flow variation in hydro Francis turbines. The gate is a set of moving vanes that closes or opens, promoting the reduction or increase of the area, respectively, in the inlet of the turbine. A group of rods connected to the regulation ring drives the wicket gate, also called distributor. This ring is moved by the servomotor of the hydraulic system responsible for the speed control (Sanjuán, 1966).

The speed governor system operating in Itaipu has great proportions, with a large volume of hydraulic fluid, and requires safe and quick actuation. Due to the complexity of the system, it is possible to rearrange it in subsystems, so the circuit become more didactic to analyze. The major subsystems are the pressurization unity, accumulators, valves and control devices for servomotors, actuators (servomotors) and the air auxiliary system. Other essential mechanism for the safe operation of the speed governor is the overspeed sensor valve (actuated by the overspeed device mounted on the shaft), the control valve for the wicket gate closing time and the servomotor hydraulic lock.

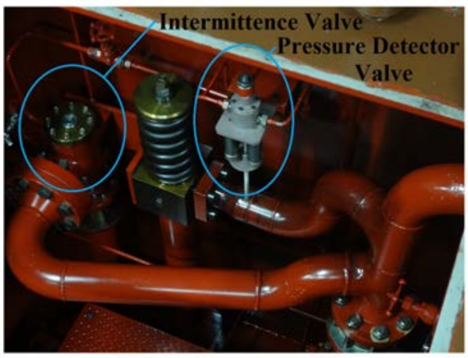
In Fig. 1 are shown some system components mentioned above. The servomotor in Fig. 1 (a) is the actuator that promotes wicket gate movement and is located at 92 m above sea level (elevation 92). The electric motors of the pressurization pumps of the speed governor are shown in Fig. 1 (b), the pumps are installed inside the sump tank Fig. 1 (d) as well as the other hydraulic components responsible for the speed governor's hydraulic logic. The accumulators (Fig. 1 (c)) and the sump tank are located at the elevation 98. The intermittence valve in Fig. 1 (e) is the hydraulic component responsible for directing the discharge of the pump, controlling the intermittent work of pressurization. In the same picture (Fig. 1 (e)) is shown the pressure detector valve, responsible to open or close the intermittence valve accordingly to the pressure in the accumulators.

(a) Servomotor

(b) Pressurization pumps drivers

(c) Accumulators

(d) Sump tank



(e) Intermittence valve and pressure detector valve

Figure 1. Components of the speed governor system

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2.2 Hydraulic accumulators

It is important to describe hydraulic accumulators, because the pressure transducer installed on this tank provides the measures used to estimate the pressurization intermission time in this article. In addition, the pumps have this intermittent work rate due to the existence of the air-oil pressure vessel, which makes possible the operation of the system.

In a simple hydraulic system, the designer specifies the pressurization pumps accordingly with the maximum capacity of the actuators. This causes an over-specification of these pumps, which will require a high power electric motor to drive the pump and, consequently, a high cost with motor starters and cabling (Parr, 1999). It becomes more critical when the system has a large size and works with actuation in short periods, which is the case of a speed governor system.

There are hydraulic systems that require a constant and close range of work pressure, that needs to be maintained for a long period of time. It's hard to keep a hydraulic circuit pressurized, due to the internal or external leakage of it's components that causes pressure drops (Mobley, 2000). That is the case of the speed governor and, besides the leakage, it requires a large amount of fluid for a short period of time during the actuation.

To make the system more economical and reliable it's inserted in its logic of work a hydraulic accumulator, that reduces the pump requirements (Parr, 1999) and supply the leakages of the system's components (Mobley, 2000). Accumulators have the function to absorb the shocks caused by the instantaneous pressure that may occur due to sudden flow stoppage. They can either compensate the effect of thermal expansion or contraction which depends on the system's operational conditions (Mobley, 2000).

In Itaipu, the speed governor system has a gas pressurized accumulator called air-oil tank, responsible for those functions described above. The pressure in this accumulator is constantly monitored by the supervisory system and controls the operational regime of the pumps. In other words, the device that senses the accumulator pressure is a spring adjusted to put the pump in full load when the pressure in the air-oil tank reaches $6.1\ MPa$. This device is called pressure detector valve, and it is a exclusively mechanical component, susceptible to deviations in the adjustment, variations in the system's pressure, mechanical wear and clearances. When this valve actuate it's discharge oil pressure in the intermittence valve. This valve controls the discharge of the pump, redirecting the flow to the main pressurized line or to the heat exchangers to promote recirculation of the fluid.

Due to safety there are alarms linked to the pressure inside the tank and alarms activated by the tank's level sensor. This sensor consists of a set of level switches that send a electrical signal to the supervisory system when the level of oil inside the tank reaches specific limits. Except in generating unit number 6 “UG06”, where a modernization prototype is being tested, the level measurement is made by a float element which is also susceptible to mechanical clearances and disagreements. On UG06 unit, the sensor that measures the oil level inside the tank is a differential pressure sensor. The measuring curves behaviour of this sensor is the same of the pressure inside the tank, acquired by the pressure transducer used in all units, except that its results are in meters of oil column.

2.3 Volumetric pumps

As the hydraulic accumulators it is important to describe the operation principle of volumetric pumps. Understanding the pumps operation, it is possible to infer the influence of changes in pressurization time interval and it is important to select another pump compatible with the fluid characteristics, which is capable to deliver the necessary flow at the system's working pressure.

To obtain the necessary pressure in hydraulic systems, positive displacement pumps are used, which differently of the centrifugal pumps supply energy to the fluid due to the volume reduction (Mattos and Falco, 1992). The resultant fluid pressure depends on the restrictions applied to the flow on the downstream piping to the pump (Karassik *et al.*, 2000). Due to the pressure rise caused by the shutoff of these pumps it is necessary to install safety devices and construct a robust pipeline (White, 1998).

It is important to know the operating conditions to select a pump. Some of these conditions are the discharge pressure, speed expected range, temperature, noise level, characteristics of the motor drive, flow rate and viscosity of the discharged fluid (Exner *et al.*, 2005). Screw pumps have some advantages, like a wide range of flows and pressures, also a high range of liquids and viscosities, low internal velocities, some designs are tolerable for entrained air and other gases, cause low churning, low foaming, operate with low mechanical vibration, noise and the flow is pulsation-free (Karassik *et al.*, 2000).

These characteristics are enhanced accordingly to the design of the rotors. The pumps can be constructed with single or multiple rotors, where the multiple rotors type can be subdivided in timed or untimed design. The timed pumps generally refers to a pump with two rotors synchronized by timing gears, so the drive is connected to one screw and the other one has its movement engaged by the pair of gears, that can be assembled inside the pump housing or in a special casing that avoid contact with the pumped fluid. Untimed type of screw has rotors with mating-thread forms, so the driving force is transmitted continuously between the rotors. Generally it refers to pumps constructed with three rotors, one is the driven rotor that meshes with the others two screws called idlers. The housing provides support for the axial loads (Karassik *et al.*, 2000).

The pressurization pump operating in the speed governor of Itaipu consists of a multiple screw untimed rotor, which

ISSN 2176-5480

is similar to the description above, a power rotor and two lateral rotors. The screws have helical threads that promotes close fitting and engagement of the movement. The mating between the rotors is necessary to promote the flow through the pump. This flow is axial through the pumping elements unlike the other rotary pumps (Karassik *et al.*, 2000).

For further results it is necessary to evaluate the necessary theoretical power to drive the pump rotor (W_t) and the pump displacement (V_D). These operating parameters can be calculated with the Eq. 2 and Eq. 1 below. Q_t represents the theoretical flow through pump, N_b is the pump rotor speed, g is the gravitational acceleration, ρ is the specific mass of the pumped fluid and H_p stands for the manometric elevation of the pump. Equation 2 can be simplified to the product of ΔP, which is the provided differential pressure by the pump in Pa, with the theoretical flow Q_t.

$$Q_t = V_D N \ \therefore \ V_D = \frac{Q_t}{N_b} \tag{1}$$

$$\dot{W}_t = \rho Q_t g H_p = Q_t \Delta P \tag{2}$$

These equations can be applied to different positive fixed displacement equipment, either it is a vane, gear, lobe or screw pump. The differences between these types of rotor is the geometry used to calculate theirs displacement per revolution.

2.4 Induction motors

In this section, the emphasis given to the study of motor drives will be needed to explain the behaviour of currents and starting torque for induction motors. This is important because the proposed improvement, which this work suggests, causes a significant fall in energy consumption, because the number of pressurization cycles will be reduced, as well as the total time working in full load of the electric motor for the pressurization of the oil. For the use in the next results, it is important to estimate the output power of a induction motor, in the same way as it is necessary to know the angular speed of the driver motor. Equation 3 and Eq. 4 returns the electric power ($\dot{W}_{el}$), and the motor speed (N_m) respectively, where V_l is the line voltage, I_l is the line current and $\cos\varphi$ represents the induction motor power factor. In Eq. 4 f represents the electric tension frequency and P is the motor poles number.

$$\dot{W}_{el} = \sqrt{3}\cdot V_l \cdot I_l \cdot \cos\varphi \ \ [W] \tag{3}$$

$$N_m = \frac{120\cdot f}{P} \ \ [RPM] \tag{4}$$

Most of pump drives in industrial environment are three-phased induction motors. This equipment presents low cost when compared to other motors, it is easy to control, robust and efficient. Induction motors are composed essentially of a rotor and a stator. The stator is connected to the electric grid and electromagnetically induces currents flux in the rotor (WEG, 2013).

The interest in this study are the curves obtained for the motor torque and current, from the start of the motor to the steady operation. This is critical for the speed governor's motors, because of the high electric current that appears due to the direct on line start used. In the electrical motors of the speed governor, the start current can reach 7 to 9 times the nominal operation current. Also, when two pumps need to be driven simultaneously, the electrical panel has an overcurrent that shuts down the source of the motors and causes trip on the generating unit.

3. MATERIAL AND METHODS

To develop this article it was necessary to identify the variables involved with the pressurization pumps operation. The study started with the interpretation of the hydraulic schemes available on the technical library of Itaipu. These diagrams lead the author to comprehend the redundant assemble used to pressurize the oil. It was verified that these pumps cannot stay too long out of operation and the housing of the electric motor should always be heated. The heating is necessary due to bearings lubrication, if the housing has its heating off, the grease gets too thick and there is no possibility of starting the engine.

After this previous study and the orientation of professionals responsible for the maintenance engineering, the author identified the pressure in the hydraulic accumulator as the variable of interest, associating the intermittent behaviour of the pressurization pumps with the oil level lowering in the tank. As mentioned above the hydraulic accumulator is responsible to supply the system internal leakage and this leakage is different in all generating units, that implies different intermission times between pressurizations. This occurs due to the difference of the clearances in the hydraulic system's inherent components, that may appear with different operational conditions for the equipment.

The data collected on the supervisory system represents the level in the air-oil accumulator of the speed governor system. These data behaves as shown on Fig. 2 (a) and (b). This diagram refers to the pressure inside the tank, acquired in all generating units as a safety alarm trigger. The Fig. 2 shows the difference that can exist between the intermission time in the generating units, that means different wears and lifetime of the main pressurization pumps.

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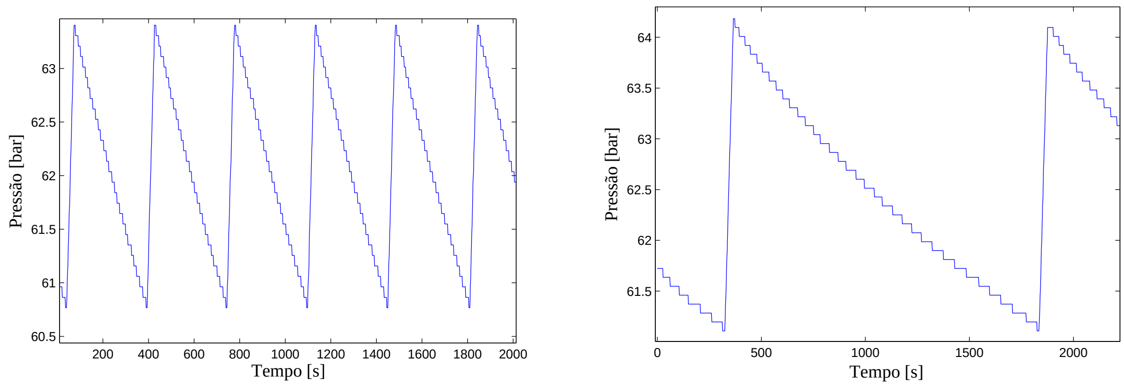

(a) Pressure diagram measured in UG02 (b) Pressure diagram measured in UG18

Figure 2. Behaviour of the oil level in two different generating units. In (a) is represented UG02 and in (b) is represented UG18

To obtain the leakage flow of each generating unit, it was necessary to measure the oil level on the generating unit 6 (UG06), where a prototype of level measurement is under test to be approved and installed in other units. With the resultant measures of this sensor, the author was able to correlate the pressure collected by the supervisory system with the oil level variation inside the accumulator tank. This leads to the possibility of estimate the leakage flow that leaves the tank in steady state operation.

To apply statistical treatment in the data collected on the supervisory system, it was implemented an algorithm in Matlab®. The algorithm consists in comparing near points in the data vector to determine the position of the maximum and minimum points. This way the pressurization intermission time can be estimated by subtracting the minimum and the maximum points, although this is not valid for the pressurization time. The pressurization time varies within 30 to 40 seconds and the minimum sampling time of the supervisory system is 2 seconds. Due to the pressure behaviour inside the tank or the sensor dynamic, the measuring error is not acceptable with this sampling time. So, the pressurization time can be evaluated by field measurement with timer device.

To accept the results for the intermission times, the author has established a maximum standard deviation of 6%. The first results was computed with 10 seconds between measuring points, during two days of operating time. The calculus obtained in some generating units was heavily dispersed and a finer study was necessary. After tests with 2 and 5 seconds sampling time, during constant output power of the generating unit, the standard deviation was acceptable in all results, as shown on the next section.

The main results of the algorithm consists in the intermission times for all generating units in minutes, the standard deviation, the oil volume reduction between pressurizations, and the leakage flow of the hydraulic accumulator. As explained above, to determine this flow is necessary to use the measuring level prototype installed in UG06. This is possible due to hydraulic system's similarity between all generating units.

This evaluated result is necessary to develop the proper equipment selection. The main purpose of this article is to determine a steady state pump, accordingly to the criteria established in the standards IEEE Std 125 and IEEE Std 1207 . This pump is also called by makeup pump or pony pump, and has the function to supply the hydraulic system's oil leakages during regular operation of larger generating units.

4. RESULTS AND DISCUSSION

The methodology explained on the previous section leads to the results in Tab. 1. The computed results was evaluated with 5 seconds sampling time, on July 16th, from 15h10min to 23h59min, except generating unit 10, that was on maintenance downtime. The data used to estimate the UG10's intermission time refers to August 2nd, from 15h10min to 23h59min. The units output power in this period was above 80%.

The deviations for the measures was acceptable in all generating units. The servomotor displacement during normal operation of the generating unit has high influence in the time between the pressurizations, which can lead to inconsistent analysis. The global analysis presented in the Tab. 1, makes possible to determine a equipment to operate in all units. Accordingly to IEEE Std 1207, the makeup pump helps reduce the number of start/stop cycles of the lead pump when the servomotors are at constant operating position. Generally, the makeup pump should be able to provide two times the oil consumption during steady-state conditions. The excess can be diverted to a kidney loop filtering system, which also helps to maintain the oil cleanliness level.

The whole idea of this design is to provide common equipment, capable to supply the demanded flow, which can either improve the oil quality and reduce the number of pressurizations of the lead pump. The major direct impact of this

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Table 1. Results obtained for the intermission times in all generating units

UG	**Intermittent time** [min]	**Standard deviation** [min]	**Percent deviation** [%]	**Volume leaked** [m^3]	**Leakage flow** [l/min]
UG01	8.84	0.36	4.10	0.62	70.15
UG02	5.53	0.29	5.22	0.54	96.96
UG03	17.31	0.75	4.32	0.59	34.37
UG04	16.51	0.54	3.26	0.58	35.04
UG05	16.11	0.75	4.65	0.57	35.18
UG06	20.97	1.35	6.45	0.60	28.54
UG07	21.19	1.19	5.63	0.72	34.10
UG08	13.07	0.63	4.80	0.63	48.24
UG09	10.74	0.32	3.00	0.58	54.19
UG09A	6.51	0.13	1.98	0.63	96.84
UG10	15.62	0.75	4.78	0.64	40.94
UG11	13.35	0.66	4.94	0.58	43.20
UG12	11.66	0.29	2.53	0.55	47.04
UG13	13.16	0.32	2.45	0.59	45.01
UG14	19.79	0.37	1.88	0.55	27.58
UG15	5.79	0.14	2.38	0.54	93.53
UG16	16.46	0.48	2.92	0.58	34.92
UG17	10.62	0.17	1.63	0.65	61.40
UG18	23.62	0.62	2.61	0.67	28.37
UG18A	11.33	0.19	1.63	0.66	58.62

installation is the energy economy it can result for the system. The main pump drivers are 160 kW motors. This motor has a high energy demand and elevated start currents. From data collected on the maintenance registers and using the Eq. 2 it is possible to estimate the energy savings potential as shown on the Tab. 2.

Table 2. Energy savings potential

UG	**Energy consumed without load** [kWh]	**Energy consumed with load** [kWh]	**Energy savings potential** [%]
UG01	1381		17.2
UG02	1491		26.6
UG03	1292		9.7
UG04	1300		10.4
UG05	1302	1178	10.6
UG06	1269		7.75
UG07	1281		8.79
UG08	1323		12.4
UG09	1354		14.9
UG09A	1451		23.2
UG10	1254		8.9
UG11	1277		10.9
UG12	1290		11.9
UG13	1274		10.5
UG14	1238	1152	7.5
UG15	1414		22.7
UG16	1256		8.9
UG17	1302		13.1
UG18	1230		6.8
UG18A	1290		12

The division on the Tab. 2 appears due to differences in the operational parameters of 50 Hz and 60 Hz drivers. Computing the global energy savings potential, it leads to 2.97 MWh daily economy and to 1085 MWh annual economy.

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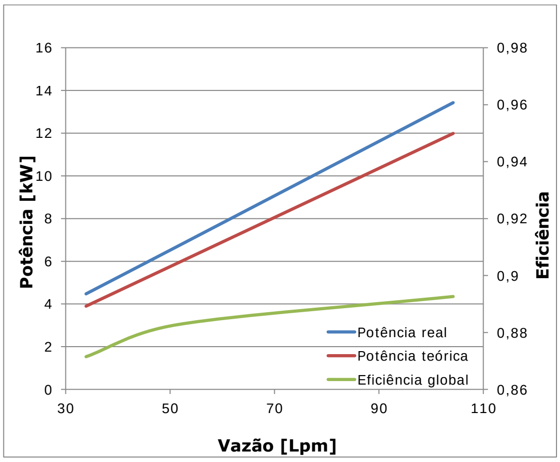
This value is theoretical, it may reduce with the makeup pump installation, for instance, but it is a very high potential for every common industrial environment.

The energy saving is only the direct impact of this modernization. The major gain for the maintenance staff is the cost reduction, low amount of spare lead pumps required and improvement of the oil's quality. Consequently, this leads to great reliability and uptime on the generating units. Therefore, based on the results evaluated and in catalogues from manufacturers it is possible to choose one equipment capable to operate in the described conditions.

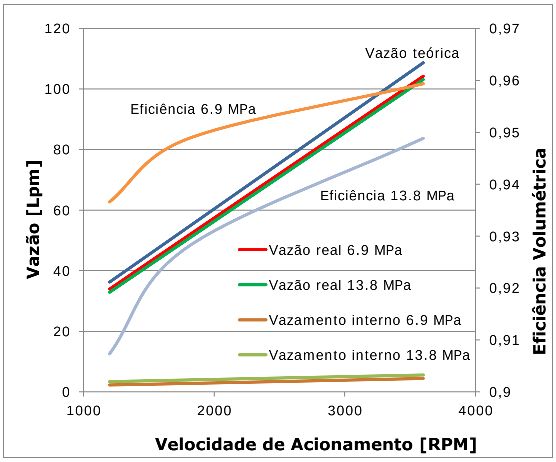
4.1 Hydraulic pump selection

Table 1 gives the maximum leakage flow of 96.96 Lpm, which will be assumed as the theoretical flow Q_t. Evaluating the maximum driver speed, given by Eq. 4, results in 3600 RPM, which leads to a minimum pump displacement of 26.93 cm^3. Equation 2 returns a minimum driver power of 10342 W, with operating pressure of 6.4 MPa and the same theoretical flow. The reference flow accordingly to the standard is the twice of the demanded by the hydraulic system. However, as the flow of generating unit 2 is not quite representative, the maximum driver power should be fixed in 15 kW. This is necessary to attend the purpose to design a small and economic system. This way, the maximum theoretical flow given by Eq. 2, using the same pressure as before, is 140 Lpm.

Based on the previous criteria, the author was able to search in manufacturers catalogues, looking for the equipment which best suits those functions. First, the rotor pump that will be used is the gear type. Gear pump is a viable option due to its simplicity of use and maintenance, it is capable to supply the demanded flow and pressure, it is efficient, resistant to cavitation and compact. Second, reliable technical data was found on Parker Hannifin's catalogue, which leads to a pump with a fixed displacement of 30.18 cm^3, that can provides maximum theoretical output flow of 104.23 Lpm. The information obtained in the catalogue is essential to plot the characteristics curves of the gear-pump, shown on Fig. 3. The design selected to operate in this project is the H77 gear pump, which can supply oil up to 13.9 MPa pressure.



(a) Power curves



(b) Volumetric efficiency and slip curves

Figure 3. Characteristic curves for the gear pump

Figure 3 shows that the input power increases with the flow, in this graphic it is possible to see the theoretical and the real power for pressurization at 6.9 MPa, which is an amount given by the catalogue close to speed governor's operating pressure. It is possible to estimate the pump efficiency evaluating the theoretical an the real power ratio. As shown on Fig. 3 a medium value obtained for the efficiency is about 88%.

It is possible to evaluate the internal leakage of this pump to the given pressure and speed, comparing the flow given by the catalogue and the theoretical flow. The internal leakage is empirically measured in bench tests applied to the equipment and behaves as a curve dependent both on pressure discharged and driver speed. The driver torque value depends on the working pressure, which is a limitation to map its behaviour. A constant operating pressure implies in a constant driver torque, that makes the power directly proportional to the pump's rotation.

Table 3 shows the mean value of the steady state leakage, the flow reference accordingly to the standard, the respective necessary power to drive the pump and the nominal rotation, applying the mean efficiency evaluated above. The values in red represents the units where is not possible to follow the normative recommendations to the hydraulic design. These occurrences do not discredit the design proposition, even though it is not possible to achieve the required flow, the number of pumping cycles will be decreased or even eliminates it in some cases. Another design restriction that must be respected accordingly to manufacturer is the minimum suction height, which is necessary to keep the flow pressure on the pump's entrance above -254 mmHg at 1800 RPM and -127 mmHg at 3600 RPM.

With the results described above, it is possible to note that these pumps cover the necessary flow at the system pressure,

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Table 3. Operating parameters of each generating unit to the selected pumps

UG	Mean flow [l/m]	Normative flow [l/m]	Required power [kW]	Driver speed [RPM]
UG01	68.04	136.08	15.44	3450
UG02	98.98	197.95	15.44	3450
UG03	33.10	66.19	8.65	2200
UG04	34.31	68.62	8.97	2270
UG05	34.11	68.21	8.91	2260
UG06	27.33	54.66	7.14	1810
UG07	33.04	66.07	8.63	2190
UG08	45.31	90.62	11.84	3000
UG09	52.30	104.61	15.44	3450
UG09A	93.16	186.33	15.44	3450
UG10	38.90	77.80	10.17	2580
UG11	42.33	84.66	11.06	2800
UG12	45.53	91.05	11.90	3020
UG13	44.86	89.73	11.73	2970
UG14	26.72	53.43	6.98	1770
UG15	93.30	186.60	15.44	3450
UG16	34.25	68.51	8.95	2270
UG17	60.41	120.82	15.44	3450
UG18	27.06	54.11	7.07	1790
UG18A	56.28	112.56	15.44	3450

with a low power driver. Furthermore, the efficiency is acceptable and it is possible to control the delivered flow rate by changing the speed of the pump driver. The benefits of these changes in the hydraulic circuit consist in reducing significantly the power consumption from the electrical motors used on the main pumps, extend the main pressurization pumps life, which implies in reducing the costs and decreasing maintenance downtime. Adapting this pump can be an alternative to achieve those goals by extending or eliminating the pressurization intermission time.

4.2 Hydraulic design and general description

To the proper operating of the hydraulic circuit, it is necessary to specify components as pressure line filter, recirculating filter, control vale to change the flow direction, manometers in the upstream and downstream line of the pump, flow-meter, check valves, relief valves, cabling and an electric panel to connect the induction motor and the directional valve. Also, it is necessary to control the actuation of the directional valve using the pressure inside the accumulator tank.

This control system should change the directional valve position whenever the pressure inside the accumulator reaches 6.1 MPa. To avoid the main pump pressurization, the pressure detector sensor should be adjusted to close the intermittence valves below 6 MPa. In this configuration, the main pump would be responsible to maintain the temperature of the speed governor's oil, while the makeup pump provides the system's leakage during steady state operation.

Figure 4 shows the hydraulic scheme of the makeup pump. When the pressure inside the accumulator reaches 6.1 MPa, the directional valve is on the position shown on the Fig. 4, discharging filtered oil to the accumulator tank. When the pressure reaches 6.4 MPa the directional valve changes the flow direction to the recirculating filter. The pressure line filter is a non-bypass filter to avoid entrance of unfiltered oil in the main hydraulic circuit. Both filters have switches that remotely indicates its obstruction or malfunction. All the components should have good manufacturing quality, long lifetime and market continuity. This design concept can substantively improve the oil's quality if implemented with the necessary adjustments to make the system better.

5. CONCLUSIONS

The speed governor is very important for energy generation. These equipment depends on hydraulic systems to develop the necessary actuating power and this article discusses one possible mean to improve the system's quality. From this study, it is possible to conclude that there are several possibilities of improvement in the actual speed governor's hydraulic system. This technical study of the pressurization intermission times is the beginning of the validation of several updates that can be applied, demanding more research on the components that restrain the reliability of the system. It happens, as explained before, due to the complexity of the system's components, changes in the technology concepts, causing the

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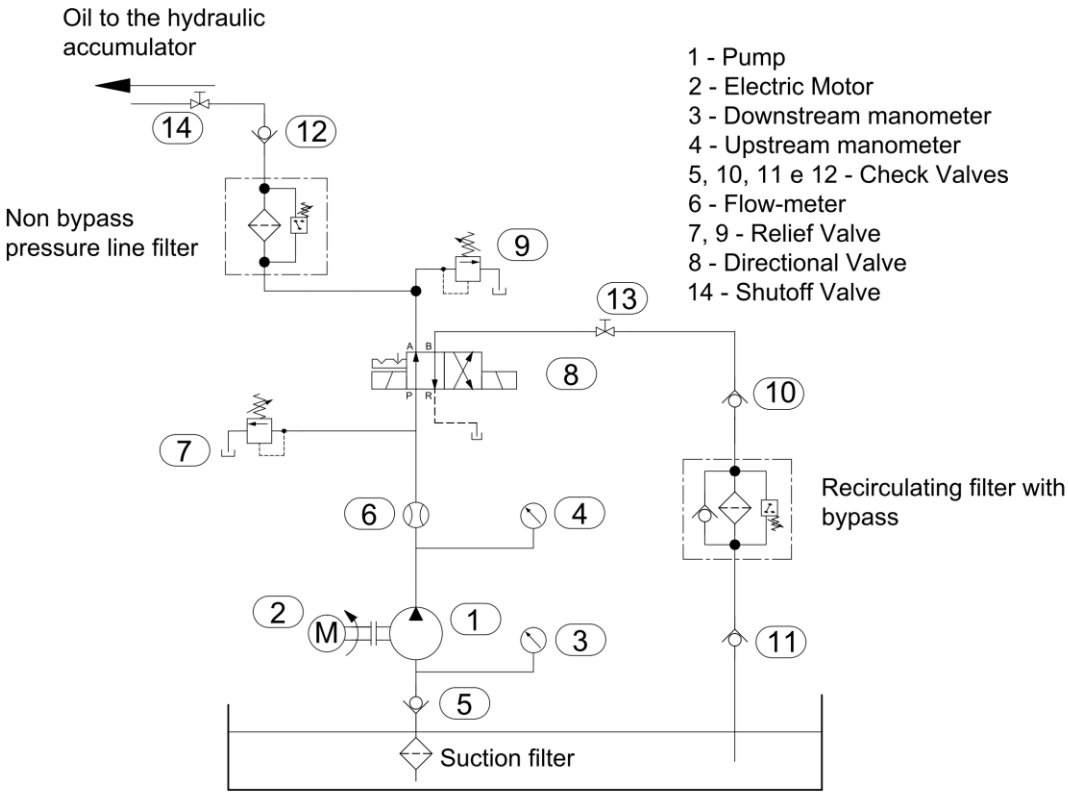
Figure 4. Hydraulic scheme for the makepu pump



discontinuity of spare parts or difficult logistics to obtain spare materials.

This proposal shows the possibility of reducing significantly the power consumption, leading to a more efficient system and raising the time of recirculation on the heat exchangers, ensuring that overheating do not occurs in the oil. This demand is crucial in some applications and more changes can be studied to be applied on the future after implementing the small pump. Also, the design attempts to improve the system's cleanliness level, with the pressure line and recirculating filtration.

The results revealed, on the intermission analysis, shows that some generating units are operating out of normative recommendations and it is a good practice to follow standards. Large systems, that generally are unique, must be operated carefully, respecting every safety recommendation and following the procedures.

6. REFERENCES

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