Araştırma Makalesi
Adıyaman Üniversitesi
Mühendislik Bilimleri Dergisi
20 (2023) 113-124

APPLICATION OF A NEW DYNAMIC ANALYSIS METHOD TO PID CONTROLLED SPEED GOVERNORS IN HYDROELECTRIC POWER PLANTS

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Geliş Tarihi/Received Date: 26.09.2022 Kabul Tarihi/Accepted Date: 08.08.2023 DOI: 10.54365/adyumbd.1180531

ABSTRACT

Hydroelectric power plants are environmental-friendly as they are renewable energy sources and make great contributions to the economies. In addition, they are in the state of insurance of the electricity network, since they can respond to the power demands of the electricity network in a very short time. In this study, the effect of the speed governors dispense valve has been added to the classical control mechanism in order to accurately predict the reaction of hydroelectric power plants to the load requirement of the electrical network. As a result of the dynamic analysis, the data obtained in the simulation environment have been compared with the responses of the real hydraulic turbine. The results show that when the dynamic analysis of the speed governor dispense valve is added to the classical control mechanism, the power response behavior in the simulation environment becomes closer to the real hydraulic turbine behavior.

Keywords: *Renewable Energy, Hydroelectric Power, Power control, Dynamic Analysis, Transfer Function*

HİDROELEKTRİK SANTRALLERDE PID KONTROLLÜ HIZ REGÜLATÖRLERİNE YENİ BİR DİNAMİK ANALİZ YÖNTEMİNİN UYGULANMASI

ÖZET

Hidroelektrik santraller yenilenebilir enerji kaynakları olmaları nedeniyle çevre dostudur ve ülkelerin ekonomilerine büyük katkılar sunarlar. Ayrıca elektrik şebekesinin güç isteklerine çok kısa sürelerde cevap verebildikleri için elektrik şebekesinin sigortası durumundadırlar. Bu çalışmada, hidroelektrik santrallerin elektrik şebekesinin yük ihtiyacına verdiği tepkiyi gerçeğe yakın bir şekilde tahmin edebilmek için klasik kontrol mekanizmasına hız regülatörlerinin dağıtma valfinin etkisi eklenmiştir. Yapılan dinamik analiz sonucu simülasyon ortamında elde edilen veriler ile gerçek hidrolik türbinin tepkileri kıyaslanmıştır. Sonuçlar göstermiştir ki hız regülatörü dağıtma valfinin dinamik analizi klasik kontrol mekanizmasına eklendiğinde simülasyon ortamındaki güç tepkisi davranışları gerçek hidrolik türbin davranışına daha fazla yakınlaşmıştır. Bu sonuçlar hidroelektrik santrallerde hız regülatörü tasarımcılarının gerçeğe yakın davranışları tahmin etmelerinde yardımcı olacaktır.

Anahtar Kelimeler: *Yenilenebilir enerji, Hidroelektrik santral, Hız regülatörü, Dinamik analiz, Transfer fonksiyon*

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1.Introduction

Speed governors in hydroelectric power plants regulate the amount of water entering the turbine wheel, initially they adjust the turbine speed for network synchronization. These governors then adjust the active power output while energizing the electrical grid [1-2-3]. In addition, speed governors play an important role in ensuring safe connection with the electricity grid in hydroelectric power plants [4-5]. In order for the speed governor to do these important tasks properly, it must have a quality control system. Speed governor designers have to design different speed governors for hydraulic turbines operating in different operating conditions. So, they must be able to predict the behavior of the hydraulic turbine in a realistic way beforehand. At this stage, applying a dynamic analysis is important, by considering the mechanism of all equipment mechanism. Most of the studies in literature are based on predicting the behavior of the hydraulic turbine in real conditions in the simulation environment [6-7-8]. Mesnage et al. [9] have developed a non-linear control model to minimize the time when hydraulic turbines start producing energy. Vereide et al. [10] have investigated the effect of the surge tank on the stability of speed governor, power control and hydraulic transitions in hydroelectric power plants. Gonzalez et al. [11-12] have conducted a passive control and stability analysis by creating a block diagram and a mathematical model for hydro-turbine systems and counter-stack multi-hydro systems. Xu et al. [13] have analyzed internal energy losses by creating a mathematical model for stability analysis in the hydraulic turbine speed governor. Khodabakhshian and Hooshmand [14] have designed a new PID control for hydroelectric power plants and analyzed the system's responses to power changes. Li et al. [15] have made dynamic analysis of hydraulic turbines by modeling the speed governor control during the sudden load increase process. Yang et al. [16] have analyzed the hydraulic damping mechanism of low frequency oscillations in power systems by using a nonlinear hydroelectric power plant model. Perng et al. [17] have made an optimization by applying an algorithm to PID controlled speed governors used for hydraulic turbines. They state that as a result of the optimization, the damping of frequency oscillations improved under different operating conditions. Guo and Yang [18] have performed a modeling for frequency control in hydraulic turbines with a balance shaft and examined the response of the turbine to frequency changes. Hušek [19] has proposed PID optimal proportional gain (kp) and integral gain (ki) values within the stability of the frequency domain sensitivity in the governor modules of hydraulic turbine units. Adhikari and Wood [20] have analyzed partial load and flow control in cross-flow hydraulic turbines by making comparative analysis. Doolla et al. [21] have proposed a charge frequency control technique for an isolated small hydroelectric power plant based on a multiple flow control system. Sharma et al. [22] have designed the automated power generation control (AGC) scheme based on the artificial neuro-fuzzy inference system (ANFIS) for the hydro-turbine power system. Liu et al. [23] have developed an estimated fuzzy control method by developing a model for load frequency control with dynamic valve position modeling for the Hydro-thermal power system. Altay et al. [24] have increased their performance by reducing the feedback effect of the speed governor with the help of the mathematical model they created in an old hydroelectric power plant according to the data of the power plant.

Figure 1 shows the control mechanism in a hydroelectric power plant. The system in Figure 1 works according to the principle of controlling the blades with pressure oil. Oil in unpressurized tank is supplying the pressurized oil tank and system by increasing its pressure with two screw pumps operating as each other's backup. When the pressurized oil tank reaches a sufficient level, a switch cuts off the energy of the solenoid valve and closes the oil path. In this process, the oil to control the adjustment blades is supplied from the pressurized oil tank. The pressure of the oil in pressurized oil tank is kept constant with pressured air according to the incompressible fluid principle. Servomotors, which are connected to the adjustment ring according to the load governor's command, open or close the adjustment wings. The opening position of the adjusting blades is transmitted to the speed governor by a feedback mechanism and the speed governor restricts the movement of the main dispense valve.

While examining the load-frequency control in hydroelectric power plants, the response given by the speed regulator to the load or frequency change and the effect of this response on the gauges are analyzed by mathematically modeling. In this study, the effect of the speed regulator distribution valve has been added to the classical control mechanism in order to predict the response of the electrical network to the load requirement more precisely. Because friction and inertia forces will be effective during the movement of the distribution valve. By adding such effects to the mathematical model with the help of dynamic analysis, the system can be modeled more closely to its real behavior.

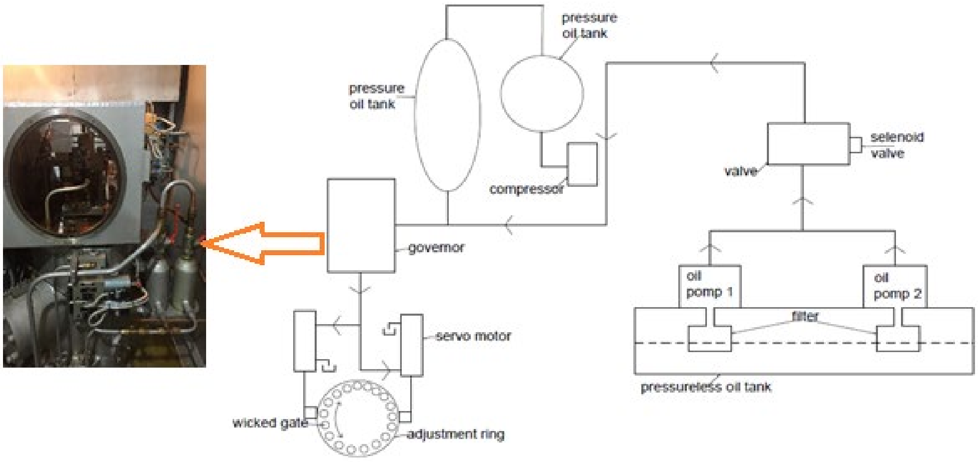

Figure 1: Adjustment blades control system in hydroelectric power plants

2.Materyal ve Metod

2.1. *Calculation of transfer function of speed governors by classical modeling method*

PID (proportional-integral-derivative) control is the most widely used control mechanism in hydroelectric power plants. This control can be applied mechanically, pneumatically and electrically. It controls the system precisely. Equation 1 shows PID control equation [3].

$$u(t)=K_p\, e(t)+K_i\int_0^t e(t)dt+K_d\frac{d}{dt}e(t) \qquad (1)$$

K_p is proportional gain, K_i is integral gain and K_d is derivative gain. The formula in Equation 2 shows the transfer function (TF) of the speed governor in "s" domain in a hydroelectric power plant connected to the interconnected system [3].

$$TF_{Governor}=K_p+(K_i/s)+K_d\, s/(T_d\, s+1) \qquad (2)$$

$$K_p=0{,}97\frac{Tm}{Tw}\ ,\ K_i=0{,}39\frac{Tm}{Tw^2}\ ,\ K_d=0{,}4T_m \qquad (3)$$

Equation 3 shows the calculation of PID control gain factors.

2.2. *Dynamic Analysis of Speed Governors Dispense Valve (New Model)*

In modern hydroelectric power plants, the control mechanism of the speed governors is based on the pressured oil control of the dispense valve to the servomotors. The main logic is the pressured oil control of the speed governor dispense valve to the servomotors, no matter how different the electronic boards and mechanical equipment in the auxiliary systems of the speed regulators are. By applying laplace transform to the mathematical model obtained as a result of a comprehensive dynamic analysis on how the dispense valve motion control works, the transfer function of the speed governor can be obtained. Figure 2 shows the location of the distribution valve inside the speed regulator.

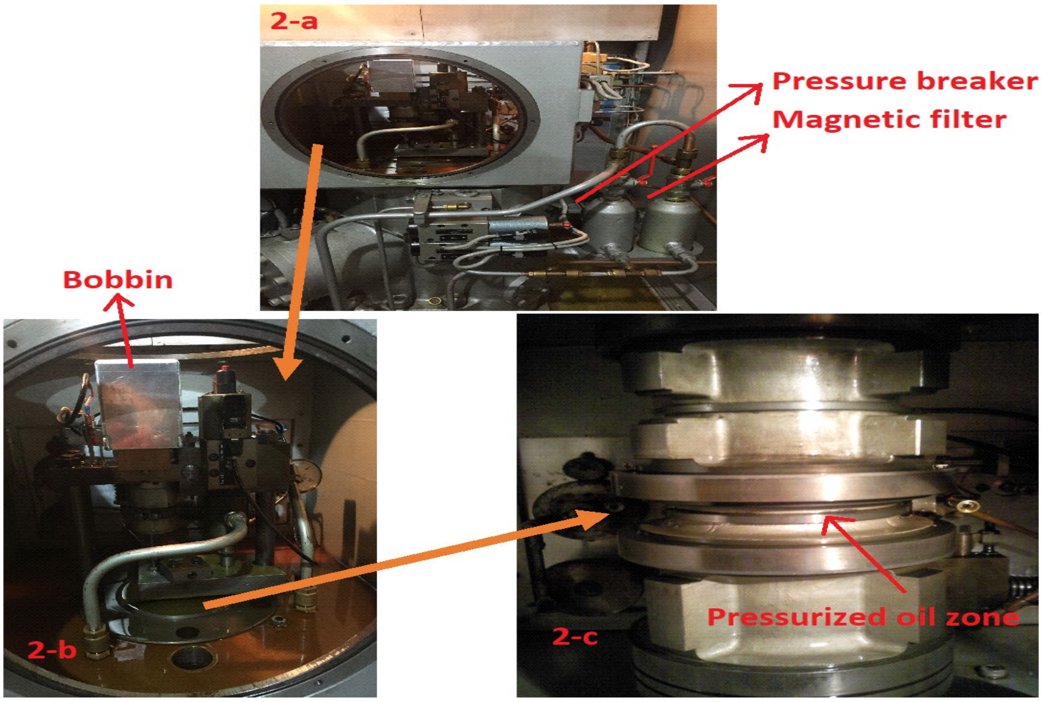

Figure 2.a) General view of the speed governor, b) View of dispense valve on speed governor, c) View of the dispense valve after mounted on the speed governor

Figure 2-a shows general view of the speed governor. The speed governor has a precise working principle to achieve frequency and power generation goals as quickly as possible. The signals coming from the primary or secondary frequency control are transmitted to the bobbin in Figure 2-b in the range of -5 to +5 volts. Some of the pressurized oil used to control the adjusting blades in servomotors passes through a set pressure breaker and creates pressure in the pressured oil zone of the dispense valve in Figure 2-c and balances the dispense valve. The balanced dispense valve sends pressurized oil to the servomotors in the direction of opening or closing the servo motors with the up and down movement of the bobbin shaft according to the incoming volt value.

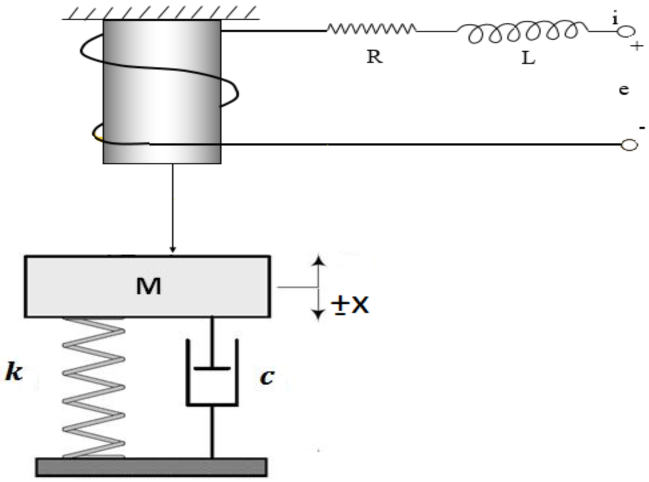

Figure 3. Schematic representation for dynamic analysis of the speed governor dispense valve

Figure 3 shows modeling of the speed governor dispense valve for dynamic analysis, working principle of which can be found in Part 2.2.

The symbols "k" and "c" in Figure 3 refer to the stiffness and damping coefficients of the oil in the dispense valve pressured oil region respectively. "M" refers to the mass of the dispense valve, "R" refers to the bobbin resistance, "L" refers to the bobbin impedance, "i" refers to the current value to the bobbin and "e" refers to bobbin voltage. The mathematical modeling of the model in Figure 3 is expressed by the following equations.

$$e(t) = Ri(t) + L\frac{di(t)}{dt} \tag{4}$$

$$\frac{di(t)}{dt} = -\frac{R}{L}i(t) + \frac{1}{L}e(t) \tag{5}$$

$$e(t) = kx(t) + c\frac{dx(t)}{dt} + \left(\frac{PA}{g} - M\right)\frac{d^2x(t)}{dt^2} \tag{6}$$

The term "$\left(\frac{PA}{g} - M\right)$" in Equation 6 is taken into account as the difference between the mass of the speed regulator main distribution valve and the effect of the oil pressure applied in the opposite direction. The value of "P" in Equation 6 refers to the oil pressure coming to the pressured oil area of the dispense valve after the pressure breaker passes, "A" refers to the area of the part where the pressurized oil affects on the dispense valve and "g" refers to gravitational acceleration.

$$Ri(t) + L\frac{di(t)}{dt} = kx(t) + c\frac{dx(t)}{dt} + \left(\frac{PA}{g} - M\right)\frac{d^2x(t)}{dt^2} \tag{7}$$

If Laplace transform is applied to Equation 7, 8 is obtained.

$$Ri(s) + Lsi(s) = kx(s) + csx(s) + \left(\frac{PA}{g} - M\right)s^2x(s) \tag{8}$$

The transfer function showing the up and down movement (x) of the dispense valve in response to the current value entering into the speed governor dispense valve bobbin as a result of PID evaluation is obtained in Equation 9.

$$\frac{x(s)}{i(s)} = \frac{R+Ls}{k+cs+\left(\frac{PA}{g}-M\right)} \tag{9}$$

3. Results and Discussion

The expression "(P*A/g)-M" in the denominator of Equation 9 determines the limit of actuation of the pressured oil coming from the pressure breaker to the dispense valve. If $P*A/g<M$, the dispense valve will not start and the turbine cannot start energy production since the oil pressure passing through the pressure breaker is not sufficient.

Table 1 shows the values used in Equation 9.

Table 1: The values used in Equation 9

R	Resistance	30 ohm
L	Inductance	30 H
k	Stiffness[25]	$5*10^7$ N/m
c	Damping[25]	$5*10^4$ Ns/m
P	Pressure	0/32 bar

A	Area affected by oil pressure	0,00149 m^2
M	Dispense valve mass	100 kg
i	Current	-0,15/+0,15 ampere
x	Movement of the dispense valve shaft	Max. 0,02 m

Table 1. Continue

The values shown in Table 1 include the characteristics of the speed regulator of the vertical Francis turbine with a power of 155 MW.

Figure 4 shows "(PA/g)-M" value according to the oil pressure coming from the pressure breaker.

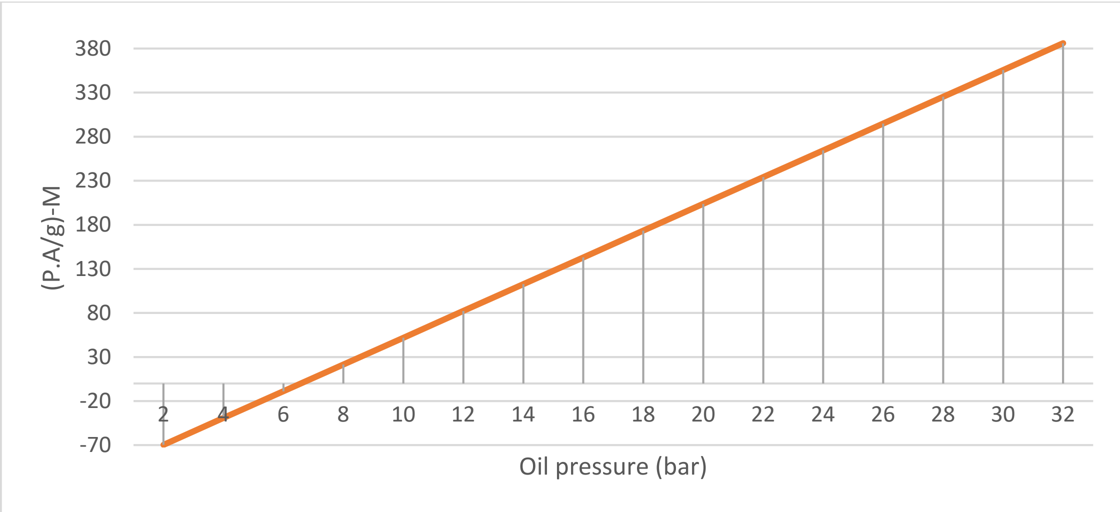

Figure 4. "(PA/g)-M" change according to the oil pressure value coming to the dispense valve

As in Figure 4, after 6 bar pressure, the dispense valve starts to act and the turbine can start producing energy after this value. In Figure 5, it shows how the 2 bar oil pressure (any pressure value less than 6 bar can be selected) coming to the dispense valve has an effect on the movement of the dispense valve.

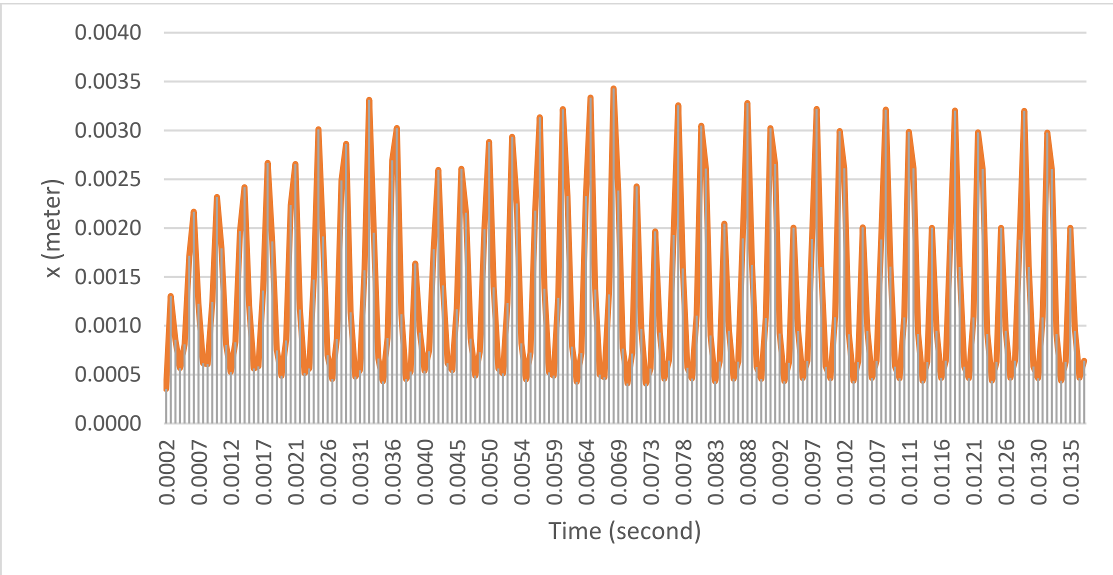

Figure 5. The situation of the speed governor dispense valve at 2 bar oil pressure

The graphic in Figure 5 shows the movement of the dispense valve when 2 bar pressure comes from the speed governor pressure breaker. Dispense valve moves irregularly and it is subject to excessive vibration. In this case, it is impossible for the turbine-generator unit to start producing energy. This movement of the dispense valve cannot send pressurized oil to servomotors in the direction of opening the adjustment blades. Since no water will come to the turbine wheel because the adjustment blades are not open, the turbine wheel cannot rotate, so the unit cannot start producing energy. In Figures 6, 7, 8, the movement of the dispense valve can be seen when the oil comes to the dispense valve at the pressure of 10, 18 and 26 bar from the pressure breaker and when different flow values are applied to the dispense valve bobbin.

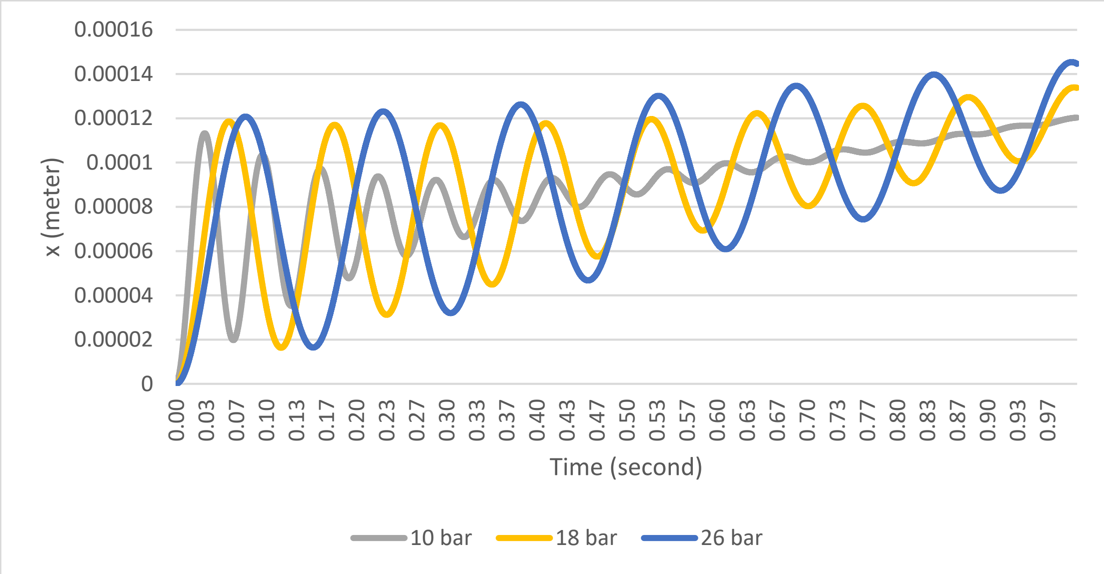

Figure 6. Movement of the dispense valve at different oil pressures when linearly increasing current is applied to the dispense valve bobbin

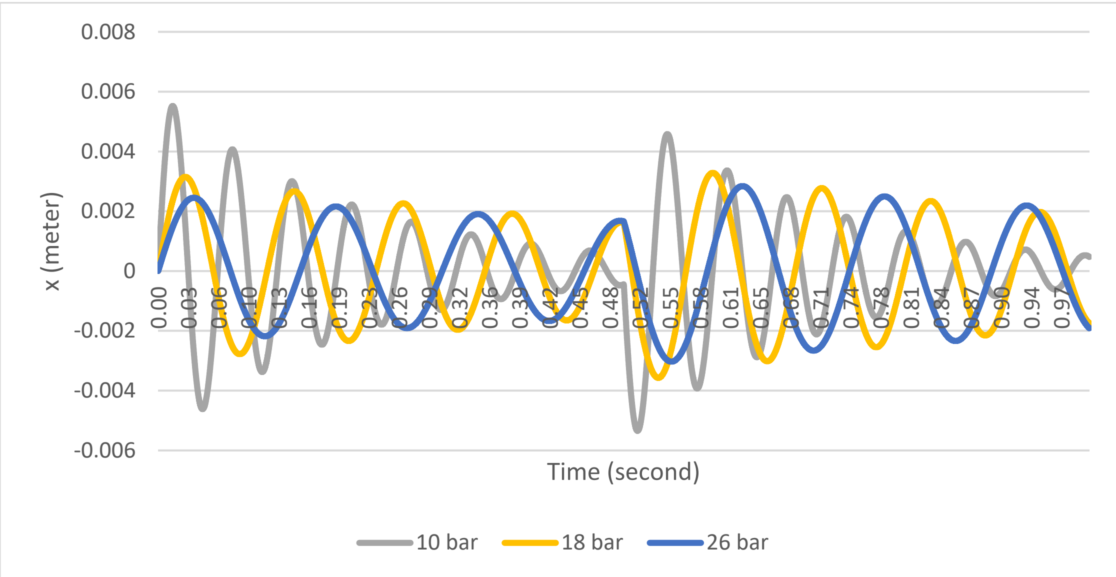

Figure 7. Movement of the dispense valve at different oil pressures when high and low flow is applied to the dispense valve bobbin in certain periods

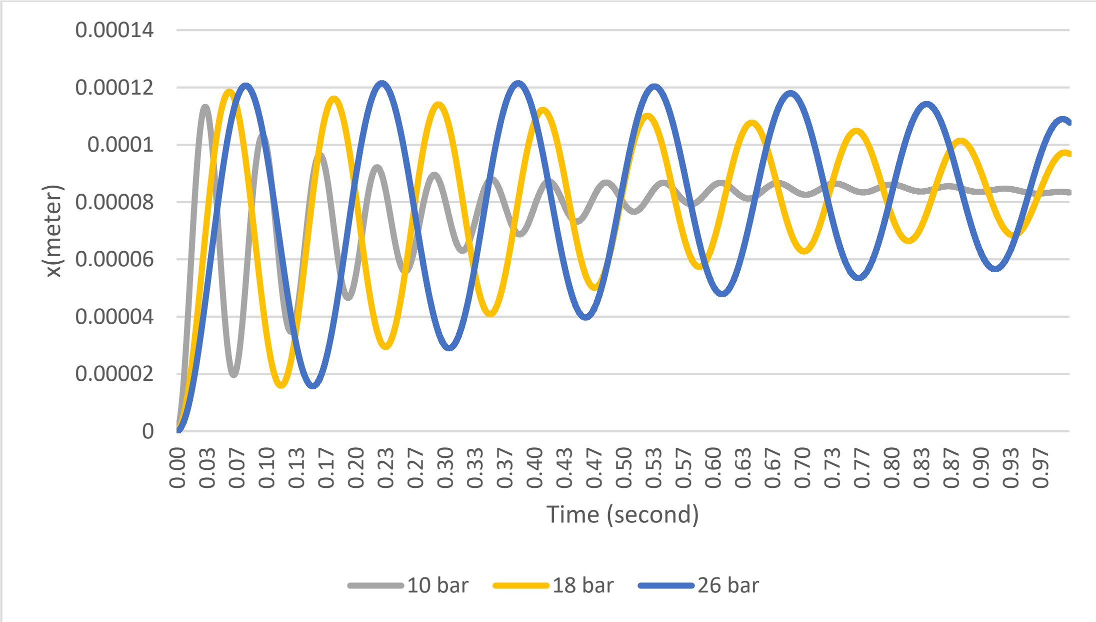

Figure 8. Movement of the dispense valve at different oil pressures when current in the form of sine wave is applied to the dispense valve bobbin

In Figures 6, 7 and 8, the situation of the dispense valve is shown in the case of 10, 18 and 26 bar pressure coming from the pressure breaker to the dispense valve pressure area and in the case of sending the current as a linear increasing, increasing and decreasing sine wave current respectively to the dispense valve bobbin. From the moment the first current is applied to the bobbin, it can absorb the 10 bar oil pressure movement faster compared to other pressure values and become stable. This indicates that 10 bar oil pressure coming from the pressure breaker to the dispense valve will provide a more precise control in the speed governor control. In addition, the limit of the time term on the x-axis of the graphs is determined as the system approaches stability.

3.1 *Application of new dynamic analysis to PID controlled general system in hydraulic turbines*

In order to examine the control and response to the aimed power in hydraulic turbines, the general response of the water flow in the penstock, the adjustment blades, the servomotors, the turbine-generator unit should be examined.

The adjustment blades control the water flow coming to the turbine wheel. The position of the adjusting blades depends on the control signal of the speed governor. Equation 10 shows the adjustment blades transfer function [26].

$$TF_{\text{guide vane}} = \frac{Y(s)}{U(s)} = \frac{1}{(T1.s+1).(T2.s+1)} \tag{10}$$

T1 and T2 are time constant and determined by the pressure and flow characteristics of the adjustment blades and servomotors. Y(s) refers to speed governor output signal and U(s) refers to the feedback signal coming from adjustment blades to the speed governor.

There are two servomotors in hydraulic turbines. Servomotors control the adjustment blades with the pressurized oil coming from the speed governor dispense valve. Equation 11 shows the servomotor transfer function.

$$TF_{\text{servo}} = \frac{1}{s.Tp+1} \tag{11}$$

Mathematical modeling of the penstock of a hydroelectric power plant is determined by three basic equations: the velocity of water in the penstock, the effect of gravity and power generation in the turbine. Equation 12 shows the transfer function for penstock.

$$TF_{penstock} = \frac{1 - s.Tw}{1 + 0,5.s.Tw} \tag{12}$$

$$T_w = \frac{L1.q}{h.g.Ap} \tag{13}$$

Generator dynamics refers to the oscillation equation that associates rotating machine inertia (Tm) with acceleration torque. The load depends on the D value of the load damping. Equation 14 shows the transfer function for the turbine-generator [3].

$$TF_{türbine\text{-}generator} = \frac{1}{s.Tm + D} \tag{14}$$

Table 2 shows the explanation of the terms in Equations 10, 11, 12, 13 and 14 and the values of the 155 MW power Francis turbine used in the calculations

Table 2. Values used in calculations

L1	Turbine penstock length	548,684 m.
q	The water flow passing from the penstock	115 m^3/s
h	Net head	135 m.
g	Gravity acceleration	9,81 m/s^2
A_p	Sectional area of the penstock	21,2 m^2
T_w	Water starting time of a single penstock	Calculated
T_p	Pilot valve servomotor time constant	0,05 s.
T_m	Machine starting time	7,99 s.
D	Load damping factor	0,5

Figure 9 shows the block diagram with the PID controlled control system in hydroelectric power plants and the speed governor dispense valve transfer function added to this control system.

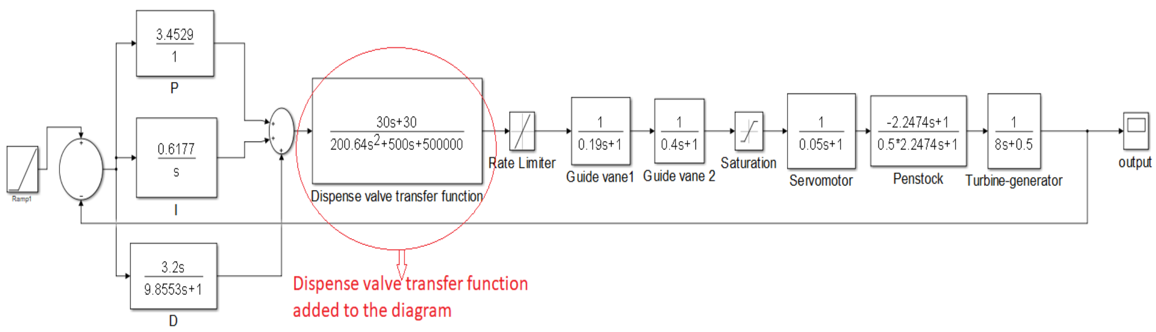

Figure 9. Version of speed governor dispense valve block added to PID controlled block diagram in hydroelectric power plants

The graphic in Figure 10 shows the response of hydraulic turbine to aimed power after the dispense valve transfer function is added and not added depending on block diagram in Figure 9.

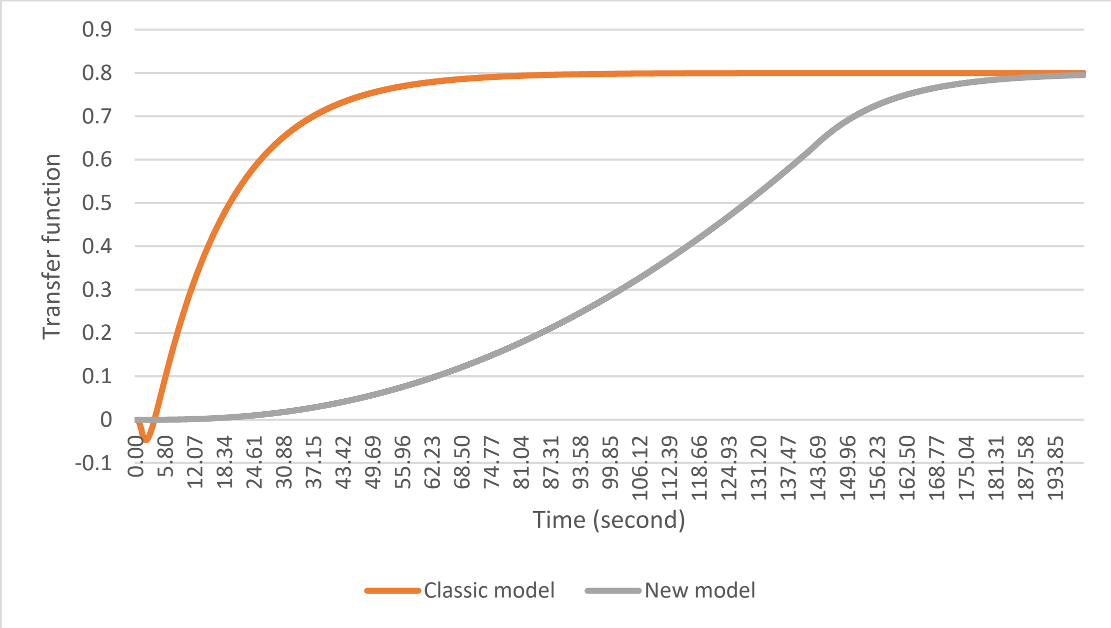

Figure 10. Responses of the classical model and the new model to the power demand

Figure 10 shows that the power demand stabilizes in approximately 80 seconds in the classical model, and the power demand stabilizes in approximately 180 seconds in the new model. The new model is closer to reality. In order to prove this fact, the response to the power demand in the real hydraulic turbine in Figure 11 is compared with the graphic in Figure 10.

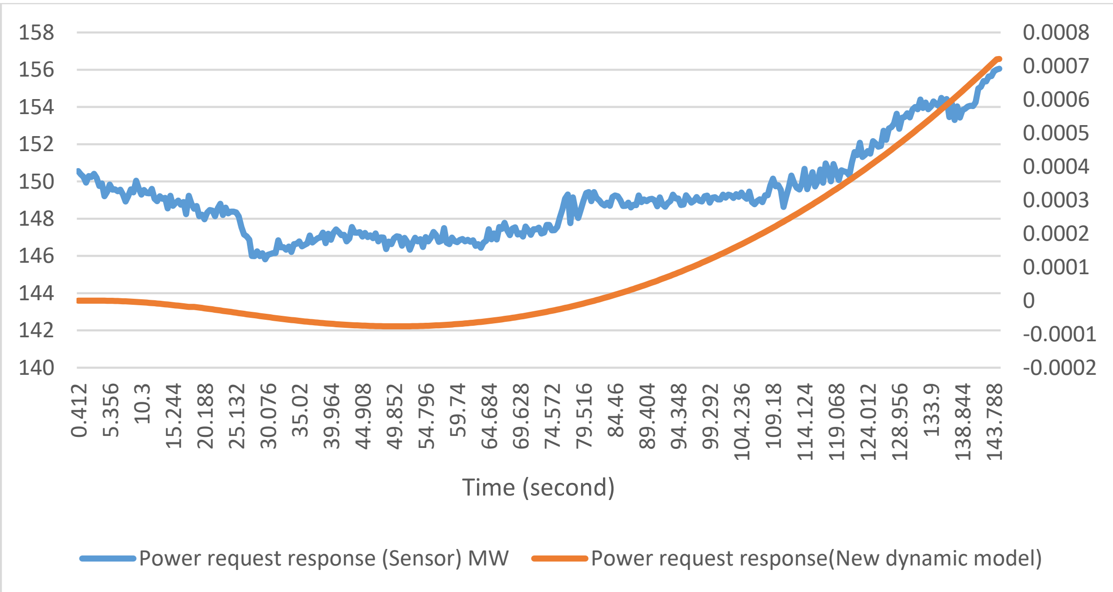

Figure 11. Comparison of the hydraulic turbine's response to the power demand and the response of the new dynamic model to the power demand

Figure 11 shows the comparison of the response of the hydraulic turbine to the power demand with the response of the dynamic model created by measuring under real operating conditions. While the hydraulic turbine produces 150 MW of energy under real operating conditions, 156 MW power demand is given. While the hydraulic turbine is performing this power demand, the graph is similar to the graph of the dynamic model. Figure 11 shows that the new model is closer to the real conditions.

4. Conclusion

This research investigates the effect of the dispense valves movement of the speed governors on the control mechanism, which is a neglected part when creating the control mechanisms of hydroelectric power plants in the simulation environment. The working principle of the dispense valve is explained, its dynamic analysis is made and the movement conditions are examined. The expression of (P.A)/g-M in the denominator of the formula obtained as a result of dynamic analysis is movement condition formula of the dispense valve and for the dispense valve to start (when dispense valve starts to act it means that the hydraulic turbine starts to rotate.) in the graphic drawn based on this formula, it is clear that pressure coming to the dispense valve should be above 6 bar. To support this idea, a simulation is applied as if 2 bar pressure oil is sent to the dispense valve, and the dispensing valve does not move but only vibrates. Then, different currents are sent to the dispense valve bobbin at different oil pressures above 6 bar, and charts are drawn. It is observed that the dispensing valve becomes stable in a shorter time at 10 bar oil pressure. The general control diagram of the hydroelectric power plants and the new model to which dispense valve transfer function is added, are compared according to the load response. The difference in this comparison is compared with the power response of the actual hydraulic turbine. As a result of this comparison, it is obvious that the new model gives a closer result to the reality. This situation is thought to be very useful for speed governor designers in terms of anticipating closer to real conditions.

Conflict of Interest Statement

The authors of the article declare that they do not have any personal or financial conflicts of interest with any institution, organization, or person.

References

[1] Gezer D., Taşcıoğlu Y., Çelebioğlu K., Speed control of hydraulic turbines for grid synchronization using simple adaptive add-ons, Measurement and Control 2018;51: 276-284. 10.1177/0020294018786743.
[2] Strath B., Kuljaca O., Vukic Z., Speed and active power control of hydro turbine unit, IEEE Trans. Energy Convers. 2005;20: 424– 434. 10.1109/TEC.2004.837278
[3] Hernandez G.A.M., Mansoor S.P., Jones D.L.i., Modelling and Controlling Hydropower Plants. Springer, 2013 (pp. 169-172).
[4] Ahn S.H., Xiao Y.X., Wang Z.W., Zhou X.Z., Luo Y.Y., Performance prediction of a prototype tidal power turbine by using a suitable numerical model, Renew. Energy 2017;113:293-302. https://doi.org/10.1016/j.renene.2017.06.021
[5] Riasi A., Tazraei P., Numerical analysis of the hydraulic transient response in the presence of surge tanks and relief valves, Renew. Energy 2017;107:138-146. https://doi.org/10.1016/j.renene.2017.01.046
[6] Weixelbraun M, Renner H, Kirkeluten O, Lovlund S. Damping low frequency oscillations with hydro governors. IEEE PowerTech (POWERTECH). Grenoble 2013.
[7] Tanıdır Ö., Cebeci M.E., Gençoğlu C., Tör O.B., A strategy to enhance AGC performance of power systems that suffer inter-area oscillations and a case study for Turkish power system, International Journal of Electrical Power & Energy Systems 2012;43:941-953. https://doi.org/10.1016/j.ijepes.2012.06.047
[8] Guo W., Yang J., Stability performance for primary frequency regulation of hydro-turbine governing system with surge tank, Applied Mathematical Modelling 2018;54:446-466. https://doi.org/10.1016/j.apm.2017.09.056

[9] Mesnage H., Alamir M., Perrissin-Fabert N., Alloin Q., Nonlinear model-based control for minimum-time start of hydraulic turbines, European Journal of Control 2017;34:24–30. http://dx.doi.org/10.1016/j.ejcon.2016.12.002
[10] Vereide K., Svingen B., Nielsen T.K., Lia L., The Effect of Surge Tank Throttling on Governor Stability, Power Control, and Hydraulic Transients in Hydropower Plants, IEEE Transactıons On Energy Conversıon 2017;32. 10.1109/TEC.2016.261482
[11] Gonzalez W.G., Garces A., Escobar A., Passivity-based control and stability analysis for hydro-turbine governing systems, Applied Mathematical Modelling 2019;68:471-486. https://doi.org/10.1016/j.apm.2018.11.045
[12]Gonzalez W.G., Montoya O.D., Garces A., Standard passivity-based control for multi-hydro-turbine governing systems with surge tank, Applied Mathematical Modelling 2020;79:1-17. https://doi.org/10.1016/j.apm.2019.11.010
[13] Xu B., Jun H.B., Chen D., Li H., Zhang J., Blanco C.J.C., Shen H., Stability analysis of a hydro-turbine governing system considering inner energy losses, Renewable Energy 2019;134:258-266. https://doi.org/10.1016/j.renene.2018.11.026
[14] Khodabakhshian A., Hooshmand R., A new PID controller design for automatic generation control of hydro power systems, Electrical Power and Energy Systems 2010;32:375-382. 10.1016/j.ijepes.2009.11.006.
[15] Li H., Chen D., Zhang H., Wang F., Ba D., Nonlinear modeling and dynamic analysis of a hydro-turbine governing system in the process of sudden load increase transient, Mechanical Systems and Signal Processing 2016;80:414-428. https://doi.org/10.1016/j.ymssp.2016.04.006
[16] Yang W., Norrlund P., Bladh J., Yang J., Lundin U., Hydraulic damping mechanism of low frequency oscillations in power systems: Quantitative analysis using a nonlinear model of hydropower plants, Applied Energy 2018;212:1138-1152. https://doi.org/10.1016/j.apenergy.2018.01.002
[17] Perng J.W., Kuo Y.C., Lu K.C., Design of the PID Controller for Hydro-turbines Based on Optimization Algorithms, International Journal of Control, Automation and Systems 2020;18:1-13. https://doi.org/10.1007/s12555-019-0254-7
[18] Guo W., Yang J., Modelling and dynamic response control for frequency regulation of hydro turbine governing system with surge tank, Renewable Energy 2018;121:173-187. https://doi.org/10.1016/j.renene.2018.01.022
[19] Hušek P., PID controller design for hydraulic turbine based on sensitivity margin specifications, Int. J. Electr. Power Energy Syst. 2014;55:460-466. https://doi.org/10.1016/j.ijepes.2013.09.029
[20] Adhikari R.C., Wood D.H., Computational analysis of part-load flow control for crossflow hydro-turbines, Energy for Sustainable Development 2018;45:38-45. https://doi.org/10.1016/j.esd.2018.04.003
[21] Doolla S., Bhatti T.S., Bansal R.C., Load Frequency Control of an Isolated Small Hydro Power Plant Using Multi-pipe Scheme, Electric Power Components and Systems 2011;39 https://doi.org/10.1080/15325008.2010.513362
[22] Sharma G., Nasiruddin I., Niazi K.R., Bansal R.C., ANFIS Based Control Design for AGC of a Hydro-hydro Power System with UPFC and Hydrogen Electrolyze Units, Electric Power Components and Systems 2018;46. https://doi.org/10.1080/15325008.2018.1446197
[23] Liu X., Kong X., Lee K., Distributed model predictive control for load frequency control with dynamic fuzzy valve position modelling for hydro–thermal power system, IET Control Theory and Applications 2016;10 10.1049/iet-cta.2015.1021.
[24] Altay A., Şahin C., İskender İ., Gezer D., Çakır C., A compensator design for the aged hydroelectric power plant speed governors, Electric Power Systems Research 2016;133;257-268
[25] Li Q., Zhang S., Ma L., Xu W., Zheng S., Stiffness and damping coefficients for journal bearing using the 3D transient flow calculation, Journal of Mechanical Science and Technology 2017;31-5:2083-2091. DOI 10.1007/s12206-017-0405-9
[26] Munoz-Hernandez G.A., Mansoor S.P., Jones D.L., Modelling and Controlling Hydropower Plants, Springer 2013. Doi 10.1007/978.1.4471.2291.3