Improving High Efficiency and Reliability of Pump Systems using Optimal Fractional-order Integral Sliding-Mode Control Strategy

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Abstract— **In this paper, a robust optimal efficiency controller for a complete water pumping system is designed based on the Fractional order Integral Sliding Surface (FISMC) with Linear Quadratic Regulator (LQR) related to the Minimum Electric Loss (MEL) condition. The developed model's novelty is based on a strategy to improve the control's performance robustness with optimal costs, in which a compromise is made between minimizing electric motor power losses and accurate flow rate adjustment in order to operate the pump at its best efficiency point and increase discharge flow rate stability for more flexibility against frequently changing working conditions. The whole system is simulated in MATLAB SIMULINK workspace, and a comparative analysis based on control energy, chattering phenomena, stability and control robustness has been conducted between the conventional PI, LQR, Integral Super-Twisting Sliding Mode Surface (ISTSMC) and the proposed control strategy MEL-FISMC-LQR. Finally, we evaluated the performance of the designed controls, including the Integral Absolute Error (IAE). The simulation results show that the proposed control design significantly improves pumping system efficiency and stabilizes the discharge flow rate at each operation point of the pumping system, and also improves flexibility against variable-speed and throttling valve. Furthermore, the stability of the closed-loop control system is assured by Lyapunov approach, and dynamic performance regardless of external disturbances as well as unknown uncertainties and parameter variations.**

Index Terms— **Centrifugal pump, fractional-order integral sliding mode control, super-twisting algorithm, LQR, Minimum Electric Loss, Genetic Algorithm.**

I. INTRODUCTION

Pumps driven by electrical motors represent a significant part of water supply, wastewater treatment, and other industries' liquid transfer and delivery. One of the most popular types of pumping equipment is centrifugal pumps, which have consumed approximately 30% of electrical energy requirements in the industrial sector. The electrical energy of a motor-pump system is converted to the pressure energy of a liquid coming from the volute casing. During this transfer, some amount of this energy is lost in an unusable form, occurring in the motor windings, motor bearing frame, stuffing boxes, mechanical seals, friction, throttle valve, and other losses in the volute, pipeline, and impeller, [1],[2]. Therefore, many researchers focused on the energy efficiency savings of pump systems. Traditional methods include throttling valve regulation and variable speed drives (VSD), which are the two most common ways used in industry to regulate flow rate. According to research, VSD-fed IM reduces energy costs by modifying the hydraulic working point [3],[4],[5],[6]. The optimization of variable-speed pumps to achieve maximum efficiency of the pump system is a challenging due to the constraints of the system, evolutionary algorithms have been widely concerned by researchers to solve the optimization problem such as, Genetic algorithms, simulated annealing, particle swarm optimization, ant colony optimization; fuzzy optimization, golden section search (GSS), neural-network-based methods, Multi-objective optimization, convex problems, and multidisciplinary optimization. Other researchers have suggested alternative ways to improve energy efficiency. Optimal control is required to achieve electrical energy savings for hydraulic systems by reducing operating loss in an induction motor [8],[9],[10]. In recent years, different loss minimization techniques have been applied to induction motors for achieving minimum power losses for wide speed and torque operating conditions. The application of Loss Model Controllers (LMCs), electrical loss minimization techniques (ELMTs), fuzzy logic control, sliding mode observer (SMO), predictive control, and backstepping to adjust the magnetizing current by applying a nominal magnetization flux are commonly included in the loss model over the last few years [7],[11]. As reported in [3], the MEL of the induction generator is proposed to determine the optimum d-axis stator current in order to achieve the minimum electric loss of the induction generator. As described in Loss LQR, intelligent fuzzy logic control and the minimum-loss power factor propose ways to decrease the electric losses of the induction motor (IM) and to improve energy efficiency, the transient response, and dynamic stability through implementation of the optimal control strategy [12] [13]. In recent years, with the development of fractional controls, the most prominent method used is fractional order sliding mode control (FSMC), and the fractional fast terminal sliding mode (FOFTSMC) which are introduced for a class of uncertain dynamical systems. In this paper, and compared to [23],[24], the FISMC related to the cost function of the LQR with the MEL condition is proposed to improve the performance robustness of the control of the whole pumping system with optimal cost. The MEL strategy determine the optimal d-axis stator current. The LQR implemented with the linearized mathematical model of the dynamic model of IM coupled to a centrifugal pump, pipeline system, and flow control valve in order to minimise the input energy, track a desired reference, and cancel steady state errors. The fractional-order sliding mode control is employed to improve the dynamic stability and transient response of the motor-pump system in the presence of a perturbation caused by throttle valve, in a relatively short time and copes with uncertainty and parameter system variation. Finally, the Super-Twisting control is added to eliminate chattering effect and the stability analysis is studied by the Lyapunov theory. The main contributions of this paper are as follows:

- **Minimizing power losses and improving the pumping**

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systems efficiency contributes to energy conservation.

- **Offering a suitable operating mode by balancing efficiency and reliability.**
- **Improve the proposed method's effectiveness, and robustness through comparative numerical simulation analysis with previous methods.**
- **Improve the adaptability of centrifugal pumps to cope with various complicated working conditions.**
- **Applied Lyapunov theorem to study and establish the closed-loop system's stability under the presence of external disturbances, and uncertainties.**

II. Mathematical Modelling of System

1) Mathematical Modelling of Induction Motor Losses: The induction machine losses are classified as clearly, explained in the IEEE Standard into: stator joule losses; rotor Joule losses; friction, windage losses; core losses; stray-load losses; and brush contact losses (in wound rotor machines) [1],[2],[3],[4],[5]. In terms of d-and q-axis current components on the basis of [5], the total power losses of the induction machine can be described as follows

$$P_{Loss} = P_{cu(s)} + P_{iron} + P_{cu(r)} \tag{1}$$

where $P_{cu(s)}$: stator copper losses, P_{iron} : iron losses and $P_{cu(r)}$: rotor copper losses. The total motor losses have to be expressed in terms of i_{sq} and i_{sd}.

$$\begin{aligned} P_{Loss} &= (3R_s + c_{Fe}\omega_r^2 L_m^2) i_{ds}^2 + [3R_s + 3R_r(-\frac{L_m}{L_r})^2 \\ &+ c_{Fe}\omega_r^2 (L_m(-\frac{L_m^2}{L_r})^2 + c_{str}\omega_r^2(-\frac{L_m}{L_r})^2] i_{qs}^2. \end{aligned} \tag{2}$$

where ω_r, L_m,$(L_r; R_r)$,$(L_s; R_s)$, c_{Fe} and c_{str} are the rotor angular speed, mutual inductance, rotor self inductance-resistance, stator self inductance-resistance and the stray and iron losses coefficients respectively. Then the overall squirrel-cage induction motor electric power loss can be expressed as [4],[5]:

$$P_{Loss} = a i_{sd}^2 + b i_{sq}^2, \tag{3}$$

where $a = 3R_s + c_{Fe}\omega_e^2 L_m^2$, and $b = (3R_s + 3R_r \frac{L_m^2}{L_r^2}) + [c_{Fe}(L_m - \frac{L_m^2}{L_r})^2 + c_{str}\frac{L_m^2}{L_r^2}]\omega_e^2$.

2) Mathematical Modelling of Pump: The polynomial approximation of a centrifugal pump, able to predict the pressure differential H created by impeller rotating speed [1],[2],[4],[5],[6],[7],[8].

$$H = (A\omega_r^2 + BQ\omega_r + CQ^2) = k_1\omega_r^2, \tag{4}$$

where Q, $A = 27.53$, $B = -3042$, and $C = -1.018e{+}06$ are output flow rate and characteristic constants for each pump can be approximated with plotting and fitting (4) [12],[13]. The centrifugal pump resistive torque can be calculated by means

$$T_p = k_p Q \omega_r, \tag{5}$$

where k_p is the centrifugal pump constant, which depends on pump nominal data obtained by using the affinity laws.

3) Mathematical Modelling of Pipe System: The Darcy-Weisbach empirical equation gives an expression of the pressure loss generated due to the friction in pipelines [15],[16],,[17].

$$H_{losses} = k_2 Q^2, \tag{6}$$

where k_2 is the constant of losses which can be found by plotting and fitting the losses generated in the pipeline.

4) Mathematical Modelling of Flow Control Valve: The mathematical modeling (linear and the logarithmic characteristic) of the control valve is determined by using the simplifying hypothesis, which considers that the pipe drop pressure ΔP_p is not modified and the value of the energy balance of the hydraulic system is $P_0 = P_{out} + \Delta P_v + \Delta P_p$, where P_0 is the pump output pressure, ΔP_v is the drop pressure of the control valve, ΔP_p is the drop pressure of the pipe and P_{out} is the pressure of the exit of the pipe. In this case, the flow through the flow control valve can be written as [1],[2],[4],[5],[6],[7],[8]

$$Q = k_{v_0} A(x) \sqrt{\Delta p_v}, \tag{7}$$

where k_{v_0} is the constants of the valve at the maximum opening determined by the valve manufacturers, and

$A(x)$ is a function that expresses the change in the cross-section of the valve and that is determined on the basis of the constructive data for the valve. In this paper, it is assumed that $A(x)$ is nonlinear (the logarithmic characteristic) then $A(x) = e^{xk}$, with $k = \log(\frac{k_s}{k_{v_0}})$, and in this case $k_s = 25mh^{-1}, k_{v_0} = 1mh^{-1}$.

The pressure difference across the valve can be calculated by

$$\Delta p_v = \frac{H - H_{losses}}{10.2} = \frac{k_1\omega_r^2 - k_2 Q^2}{10.2}, \tag{8}$$

where the number 10.2 is a constant to convert bars to meters of water.

Using the simplifying hypothesis with replacing (8) into (7), the Q flow rate can be written as

$$Q = \sqrt{\frac{k_3 e^{2xk}}{[1 + k_4 e^{2xk}]}}\omega_r, \tag{9}$$

where $k_3 = \frac{k_1 k_{v_0}^2}{10.2}$, and $k_4 = \frac{k_2 k_{v_0}^2}{10.2}$.

III. Optimal Efficiency Controller design

The first part is based on the primary objective of this work, which is to minimize the value of the electric losses under the hydraulic constraints and determine the equilibrium operating point of the pump system.

1) MEL condition: It is possible to reduce the loss of machine, by minimizing the value of the electric losses, we consider that both the stator frequency ω_e and the electromagnetic torque T_e are constant with respect to i_{sd} current. The electric loss minimization condition is given by [3],[16]

$$\frac{\partial P_{Loss}}{\partial i_{sd}} |_{\omega_e = const.} = 0. \tag{10}$$

Using (3), condition (10) is satisfied when

$$a i_{sd} + b i_{sq}\frac{\partial i_{sq}}{\partial i_{sd}} = 0, \tag{11}$$

we consider that the electromagnetic torque is constant with respect to i_{sd} current

$$\frac{\partial T_e}{\partial i_{sd}} |_{\omega_e = const.} = 0. \tag{12}$$

using (12) with $T_e = \frac{3}{2}p\frac{L_m^2}{L_r} i_{sq} i_{sd}$ results $\frac{\partial i_{sq}}{\partial i_{sd}} = -\frac{i_{sq}}{i_{sd}}$, and substituting this result into (11), the MEL condition is obtained as follows

$$i_{sd_{opt}} = i_{sq} G_d \sqrt{\frac{1 + T_a^2 \omega_e^2}{1 + T_b^2 \omega_e^2}} \tag{13}$$

where $G_d = \sqrt{1 + \frac{R_r L_m^2}{R_s L_r^2}}$, $T_a = L_m \sqrt{\frac{c_{Fe}(L_r - L_m)^2 + c_{str}}{3\left(R_s L_r^2 + R_r L_m^2\right)}}$, and $T_b = L_m \sqrt{\frac{c_{Fe}}{3R_s}}$.

2) Equilibrium Operating Point: The second part is based on determining the second condition, which matches the equilibrium condition of the pump. The electromagnetic torque of the induction motor is a sum of torques about the common shaft of the motor and the water pump. [7],[8]

$$J \frac{d\omega_r}{dt} = \frac{3}{2} p \frac{L_m^2}{L_r} i_{sq} i_{sd} - (k_p Q + b)\, \omega_r. \tag{14}$$

where J and b are the rotor inertia, and the frictional damping coefficient. The equilibrium operating point is obtained when $\frac{d\omega_r}{dt} = 0$, from (14), it is concluded that the optimal MEL condition become

$$i_{sd_{opt}} = \sqrt{G_q G_d} \left[\frac{1 + T_a^2 \omega_e^2}{1 + T_b^2 \omega_e^2}\right]^{1/4} \sqrt{\omega_r}, \tag{15}$$

and

$$i_{sq_{opt}} = \sqrt{\frac{G_q}{G_d}} \left[\frac{1 + T_b^2 \omega_e^2}{1 + T_a^2 \omega_e^2}\right]^{1/4} \sqrt{\omega_r}. \tag{16}$$

where $G_q = \frac{2}{3p} \frac{L_r}{L_m^2} \left[k_p \sqrt{\frac{k_3 e^{2xk}}{\left[1 + k_4 e^{2xk}\right]}} \omega_r + b\right]$.

The next step is to design a suitable controller optimally to have the ability to ensure the dynamic stability and low tracking error, chattering, and energy consumption of the whole system in the presence of a perturbation, in a relatively short time.

3) Design of Optimal Fractional Sliding Mode Surface Strategies: The fractional-integral sliding mode control technique is proposed in this paper for that the system could be robust to uncertainties and external disturbances. [23],[24]. Using Caputo definition, an integral sliding mode surface based on fractional-order is presented as:

$$S = F\left[e(t) - e(0) - D^{\alpha}\left(Ae(t) + Bu_{eq}\right)\right], \tag{17}$$

where $e(0)$ is the initial condition of the system, $F = \left(B^T B\right)^{-1} B^T$, and A and B represent the state matrix and control matrix of the dynamic model of pumping system around the equilibrium operating point in state space $x = Ax + Bu + E$, where $x = \left[i_{sq}\ i_{sd}\ \psi_{rd}\ \omega_r\right]^T$, $u = \left[u_{sd}\ u_{sq}\right]^T$, with $e(t) = x(t) - x(t)_{ref}$ the tracking error between the four actual states of the system and the desired references and $E = \Delta A + \Delta B u + d$ is the unknown lumped uncertainties possesses un upper bound [20],[21],[22].

4) Linear-Quadratic Regulator: The optimal gain matrix, K, is determined by using LQR which can be considered a powerful optimal control among various linear controllers to design the fractional-integral sliding mode control with a minimum cost [18],[19]. Then the combination of equivalent control part u_{eq} and discontinuous part u_{smc} can be rewritten as:

$$u = -Ke(t) + u_{smc}. \tag{18}$$

The quadratic cost function is expressed by [16],[17],[18],[20],[21]

$$J = \int_0^{\infty} \left(e^T Q e + u_{eq}^T R u_{eq}\right) dt; \tag{19}$$

TABLE I

SPECIFICATIONS OF THE COMPONENTS.

Component	Specifications
Electrical Motor	Max. power of the motor: 1.1 KW. Nominal speed 2800 rpm; $L_r = 0.029$; $R_r = 5.313$; $L_s = 0.029$; $R_s = 6.959$; $L_m = 0.6786$; $J = 0.6786$; $b = 0.00114$.
Centrifugal Pump	Nominal speed 1770 rpm, pump's parameters all defined in MATLAB by approximating polynomial
Pipeline	Nominal diameter: 0.381 m (1.5 in). Length: 10 m
Control Valve	Proportional Valve Nominal diameter: 0.381 m

After using the linearization prototype model tool of Matlab to obtain linearized dynamic around the equilibrium operating point in state space, and applying an adaptive Genetic Algorithm optimization tool (GA), the optimal matrix gain K of the closed loop optimal control law defined by proper selection of weighting matrix Q and R for the state $x(t)$ and the control input $u(t)$ respectively. Replacing the tracking error dynamics $\dot{e}(t)$ with its expression without taking into account the disturbances $\dot{x} - \dot{x}_{ref} = Ax + Bu_{eq} - \dot{x}_{ref}$, and using the conditions $S(t) = 0$ with $\dot{S}(t) = 0$ at which the state of the pump system reaches the sliding surface,

the equivalent control part is given by the following expression:

$$u_{eq} = (FB)^{-1}\left[F\dot{x}_{ref} - FAx + D^{\alpha}(FA - FBK)\, e(t)\right]. \tag{20}$$

5) Integral Supertwisting SMC: The discontinuous part is proposed based on the super-twisting integral sliding mode control as [19],[20],[21],[22].

$$u_{smc} = -(FB)^{-1}\left[\lambda D^{\alpha-1}\left(\sqrt{|S|} sgn(S)\right) + \int \beta sgn(S)\, dt\right]. \tag{21}$$

where λ and β are positive constant and $\beta > \vartheta > |\dot{E}(t)|$;$\vartheta$ denotes the derivative of the uncertainty upper bound as well $\vartheta > 0$.

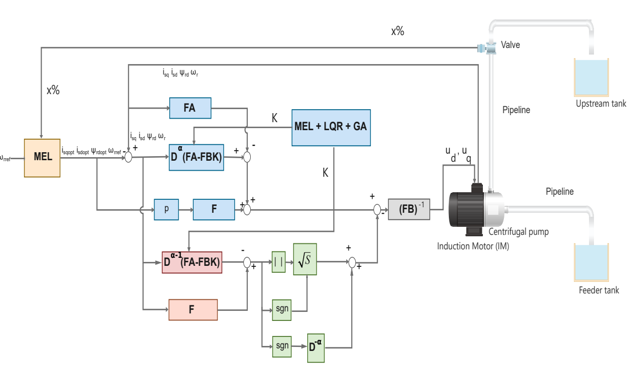

Fig. 1. Schematic diagram of the system with Control structure scheme.

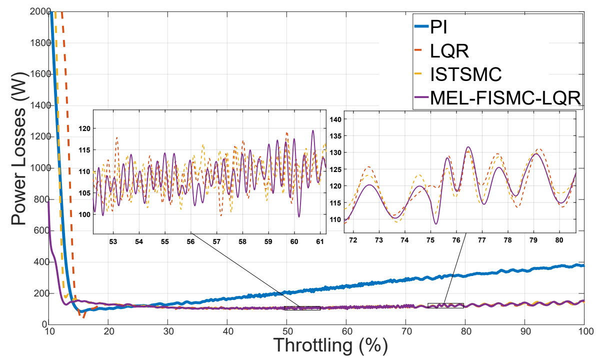

Fig. 2. Electric loss of the motor with adjusting the stroke of the throttle valve.

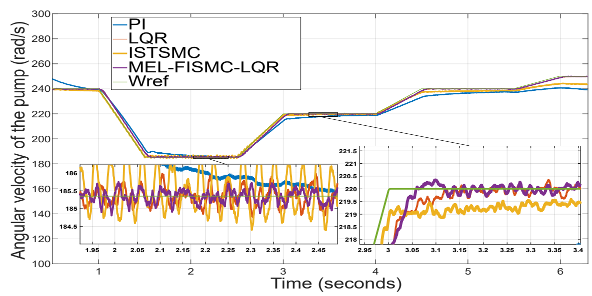

Fig. 3. Time response of speed response with adjusting the stroke of the throttle valve and rotational speed.

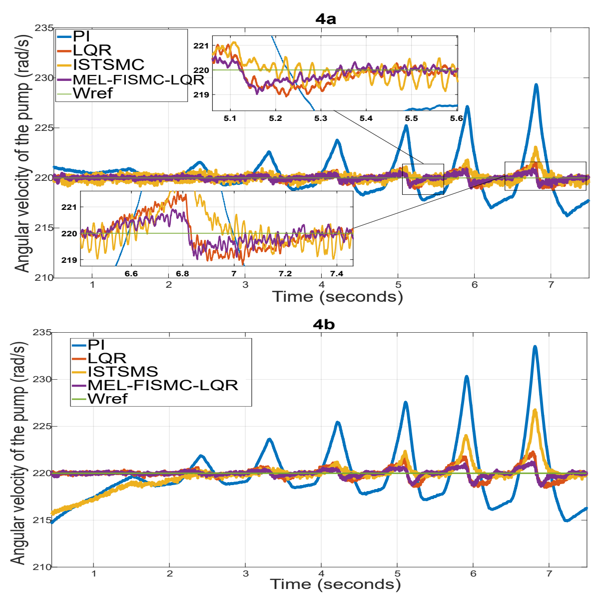

Fig. 4. Time response of the speed response under disturbance using sudden closing of the throttle valve.

Theorem : The pump system model with designed controller in (20) and (21), we can conclude that the system is asymptotically stable.

Proof. The Lyapunov function of the pump system is

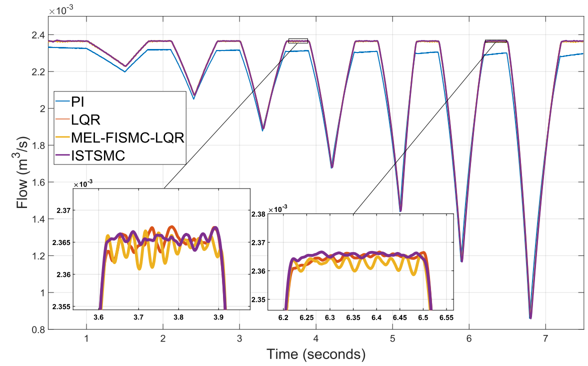

Fig. 5. Time response of flow rate in the presence of disturbance using sudden closing of the throttle valve.

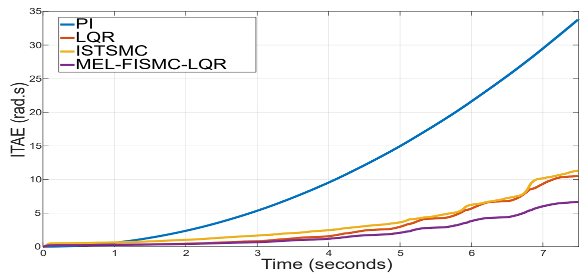

Fig. 6. Integral of time multiplied by absolute error in the presence of disturbance using sudden closing of the throttle valve.

chosen as $L = \frac{1}{2}S^2$, with $\dot{L} = S\dot{S}$.

Using (17), and after replacing the time derivative of the tracking error dynamics $\dot{e}(t)$ by its expression with taking into account the disturbances $\dot{x} - \dot{x}_{ref} = Ax + Bu + E(t) - \dot{x}_{ref}$, the time derivative of sliding surface becomes:

$$\dot{S}(t) = F\left[Ax + Bu + E(t) - \dot{x}_{ref} - D^{\alpha}\left(A - BK\right)e(t)\right]. \tag{22}$$

After substituting (20), and (21) into (22), we have

$$\dot{S}(t) = FAx + FE(t) - F\dot{x}_{ref} - D^{\alpha}\left(FA - FBK\right)e(t)$$
$$+\ F_{ref} - FAx + D^{\alpha}\left(FA - FBK\right)e(t)$$
$$-\lambda D^{\alpha-1}\left(\sqrt{|S|}sgn\left(S\right)\right) - \int \beta sgn\left(S\right)dt. \quad (23)$$

We conclude that the time derivative of the Lyapunov function becomes:

$$\dot{L} \leq -|S|\ \left[\lambda D^{\alpha-1}\left(\sqrt{|S|}sgn\left(S\right)\right) + \beta D^{\alpha-1}sgn\left(S\right)\right.$$
$$-\ D^{\alpha-1}\dot{E}(t). \ (24)$$

$$\dot{L} \leq -|S|\ D^{\alpha-1}\left[\lambda\sqrt{|S|}sgn\left(S\right) + \beta sgn\left(S\right) - \dot{E}(t)\right]. \tag{25}$$

Because $\beta > \vartheta > |\dot{E}(t)|$, then (21) becomes:

$$\dot{L} \leq -|S|\ D^{\alpha-1}\left[\lambda\sqrt{|S|} + (\beta - \vartheta)\right] \leq 0. \tag{26}$$

This means that by using Lyapunov theory, the proposed controller ensures the stability of pump system model.

TABLE II

ISTSMC

ω (rad/s)	240	230	220
System Efficiency η (%)	69.55	68.51	66.84
Power losses (W)	105.3	106.6	106.8
Q (10^{-4}m^3/s)	18.14	18.86	18.03
H (m)	34.74	31.93	28.3
Valve Opening (%)	57.38	64.13	64.05

TABLE III

MEL-FISMC-LQR

ω (rad/s)	240	230	220
System Efficiency η (%)	69.84	68.71	67.07
Power losses (W)	103.1	100.3	100.5
Q (10^{-4}m^3/s)	18.12	17.61	19.26
H (m)	34.78	31.8	27.41
Valve Opening (%)	57.26	58.31	70.58

IV. Simulation results and discussion

The whole control system and the performance of the proposed controller MEL-FISMC-LQR is evaluated through Matlab/simulink software. The parameters of the asynchronous electric motors, centrifugal pump, pipeline, and valve system are mentioned in Table I, and as shown in Fig1. Schematic diagram of the system with Control structure scheme. The numerical simulations with a comparative analysis with the PI, LQR, ISTSM controllers are conducted to better illustrate the performance and the effectiveness of the proposed controller was tested at different range of speeds and openings of throttling valve and also tested in the presence of external disturbance, uncertainties and parameter system variation. The Figs. 2 show the electrical losses curves as a function of the stroke percentage of the valve at a reference speed of $\omega_0 = 240rad/s$ with the initial 0% opening of the valve. As can be seen, the comparison of results between the PI, LQR, ISTSM and proposed controller shows that the proposed technique gives one an advantage over others controllers during adjusting the stroke of the throttle valve. The simulation results of water pumping system in tables II-III-IV-V can be used to better understand the effects shown previously, as it recapitulate simulation results of the water pumping system for variable rotational speed with different stroke of the throttle valve in terms of the overall efficiency, power losses, flow rate, and head of proposed controller and the controllers PI, LQR, ISTSM without sacrificing too much the precision in the desired the pressure head of the pump and flow rate. As can be seen, when the controls are set at $\omega_0 = 240rad/s$, the proposed controller had a considerable reduction in power losses, reaching a $36,515\%$, $2,46\%$ and $2,089\%$ reduction compared to PI, LQR and ISTSM respectively. This superiority can be justified by the effect of the MEL which determines the optimal d-axis stator current according to torque conditions. Consequently, the efficiency of the motor is increased by $11,583\%$, $1,4\%$ and $0,406\%$ compared to PI, LQR and ISTSM respectively. This demonstrated that the optimized controller finds new optimal operating point at which the minimum electric losses of the IM is attained. It is noticed that the table III indicate, when the stroke of the throttle valve at 57.26% with speed $\omega_0 = 240rad/s$, the maximum possible efficiency of the whole motor-pump system is approximately 69.84%. According to the findings, adjusting both the speed and the valve has a significant impact on power losses and the operating region, which can be expanded. Figure. 3 illustrates the resulting performance of the proposed controller under the effect of variable rotational velocities and throttle valve stroke. It can be seen that the proposed control method outperforms the other controllers in terms of flexibility against variable-speed and throttling valves when a variable reference speed is changed from 240 to 185.35 rad/s and then increased from 185.35 to 250 rad/s.Therefore, we conclude that the proposed control scheme improves flexibility against variable-speed and throttling valves compared to the other controllers. The performance of the proposed controller is also tested with other controllers in the presence of disturbance induced in the pipeline by suddenly setting the throttle valve. Figure. 4a, shows the comparison of the speed response of different control techniques under the influence of the perturbation by suddenly setting the throttle valve at a stroke from 89.3% to 25% at speed $\omega_0 = 220rad/s$. It can be observed that the proposed control strategy regulates the speed of the motor after the disturbance with a settling time of 0.83s, compared with 1.3s for other controllers, as shown in Fig.4a. Figure. 4b, shows the comparison of the speed response of different control techniques under the influence of uncertainties and parameter system variation. In this case, the speed response of induction motor a 50% increase in (Rr), magnetizing inductance (Lm) with 30% decrease and pipe roughness uncertainty is considered. It can be observed that the proposed control strategy is immune to the uncertainties and parameter system variation even though the influence of disturbance compared with results of Figure.4a Figure. 4c, shows the comparison of the speed response of different control techniques under the influence of the perturbation by suddenly setting the throttle valve at a stroke from 89.3% to 25% at speeds $\omega_0 = 230$, and $240rad/s$. Based on these

TABLE IV

MEL-LQR-GA

ω (rad/s)	240	230	220
System Efficiency η (%)	69	68.50	66.43
Power losses (W)	105.7	105.9	106.20
Q (10^{-4}m^3/s)	17.68	17.57	18.81
H (m)	35	31.76	27.6
Valve Opening (%)	55.43	58.11	68.17

TABLE V

PI

ω (rad/s)	240	230	220
System Efficiency η (%)	62.52	62.35	62.27
Power losses (W)	162.4	165	150.1
Q (10^{-4}m^3/s)	13.81	15.14	16.14
H (m)	37.26	33.16	29.4
Valve Opening (%)	41.2	48.55	55.89

results and as shown in Fig.5, it is clear that the proposed control regulates with a faster settling time compared to the other controllers the speed of the pump to bring the flow rate back to the desired value after the disturbance induced in the pipeline by suddenly setting the throttle valve. The performance indices integral of absolute speed error (IAE) of the candidate controls under the influence of the perturbation induced in the pipeline by suddenly setting the throttle valve are presented in Figs 6. The comparison shows the proposed technique is the best candidate since it renders the flattest error profile and the lowest error and can effectively improve the rejection performance of the disturbance when compared with the other controllers, and consequently, the pump-system can be balanced robustly using the proposed technique.

V. Conclusion

This study proposed an optimal Fractional-order Integral Sliding-Mode Control for high efficiency and high dynamic performance of the whole pumping system. Firstly, the LQR with MEL condition is implemented as a nominal part of integral sliding mode control to design an optimal controller with low energy consumption. Then, fractional integral sliding mode surface is introduced for that the system could be robust to uncertainties and external disturbances, also to prove the dynamic stability and transient response of the motor-pump system in the presence of a perturbation in a relatively short time, and finally, the Super-Twisting control is added to reduce chattering effect. To validate the performance and effectiveness of the proposed control scheme, a comparative analysis based on the overall efficiency, the control robustness, and the performance indices has been made between the PI, LQR, ISTSM controllers and the proposed control strategy MEL-FISMC-LQR based on the simulated results. The numerical simulation results confirm that the proposed controller is more beneficial from energy efficiency, robustness achievement, and chattering reduction.

Future work will include experimental validation. Furthermore, the effectiveness of control paradigms might be further developed to attenuate the chattering phenomenon significantly and improved by other variants such as second-order integral sliding mode control.

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