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Design and Control of a Hydraulic Based Tire Changer using H∞ and H_2 Optimal Synthesis Controllers

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Key words: Tire changer, synthesis, optimal synthesis

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Page No.: 14-19
Volume: 13, Issue 3, 2020
ISSN: 1995-4751
Botany Research Journal

Abstract: In this study, the design and control of a hydraulic system based tire changer machine have been analyzed and simulated using MATLAB/Simulink Toolbox successfully. The machine have a displacement input which is a leg pedal displacement in order to push the piston of the pump to fed the motor with a pressured hydraulic fluid to rotate the tire with an angular speed to mount and dismount it. Augmentation based H∞ and H_2 optimal synthesis controllers have been used to improve the performance of the machine. Comparison of the proposed controllers for tracking a reference input speed have been done using two reference inputs (step and random) signals. Finally, the comparative simulation results proved the effectiveness of the proposed tire changer with synthesis controller in improving the settling time and percentage overshoot.

INTRODUCTION

A tire changer is a machine used to assist tire technicians dismount and mount tires with vehicle wheels. After the wheel and tire meeting are removed from the car, the tire changer has all of the components necessary to cast off and replace the tire from the wheel. Different tire changers allow technicians to replace tires on motors, bikes and heavy-obligation trucks. New tire and wheel technology has improved positive tire changers to be able to alternate a low profile tire or a run-flat tire[1]. The mount/demount mechanism consists of the duckhead, swing arm and vertical slide. The duck head is on the backside give up of the vertical slide. The duck head is uniquely fashioned, like a tapered invoice, to healthy next and surround the rim of a wheel. It can both be made from metal or plastic. The duck head mounts and demounts the tire from the wheel. The swing arm moves left and right. The cause of the swing arm is to move the duck head near or far away from therim. The remaining factor of the mount/demount mechanismis the vertical slide. The vertical slide moves up and down in order that the duck head can match onto the edges of different length wheel widths. The vertical slide has a spring and locking take care of above the swing arm to set the duck head and preserve a solid position across the rim.

MATERIALS AND METHODS

Mathematical modelling of tire changer: Figure 1 shows the design of the hydraulic based tire changer. The tire changer is controlled by pedal which means an input displacement is entered using this pedal. This displacement is an input to the pump and the pump injects a hydraulic oil to the motor and the motor rotates the tire with an output angular speed[2].

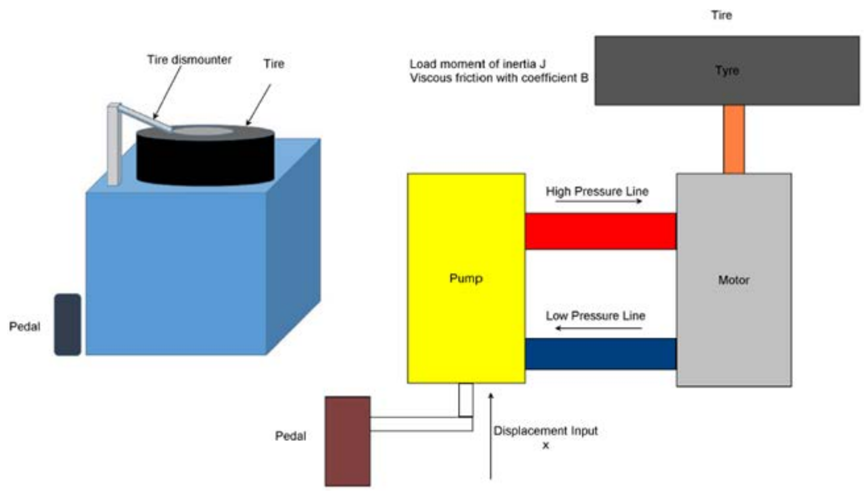

Fig. 1: Hydraulic based tire changer

The hydraulic motor is controlled by the amount of oil delivered by the pump. By mechanically changing the pump stroke, the oil delivered by the pump is controlled. Like in a DC generator and dc motor, there is no essential difference between hydraulic pump and motor. In a pump the input is mechanical power and output is hydraulic power and in a motor, it is vice versa. Let:

q_p = Rate at which the oil flows from the pump
q_m = Oil flow rate through the motor
q = Leakage flow rate
q_c = Compressibility flow rate
x = Input stroke length
θ = Output angular displacement of motor
P = Pressure drop across motor

The rate at which the oil flow from the pump is proportional to stroke displacement, i.e.. $Q_p \propto x$ Oil flow rate from-the pum:

$$q_p = K_p x \tag{1}$$

where, K_p is ratio of rate of oil flow to unit stroke displacement. The rate of oil flow through the motor is proportional to motor speed, i.e., $q_m \propto d\theta/dt$ oil flow rate through motor:

$$q_m = K_m \frac{d\theta}{dt} \tag{2}$$

where, K_m is motor displacement constant. All the oil from the pump does not flow through the motor in the proper channels. Due to back pressure in the motor, a slices of the shape flow from the pump leaks back past the pistons of motor and pump. The back pressure is the importance that is built up by the hydraulic flow to overcome the hostility of free movement offered by load on motor shaft. It is usually assumed that the leakage flow is proportional to motor pressure, i.e., $q_i \propto P$. Leakage flow rate:

$$q_i = K_i P \tag{3}$$

where, K_i is leakage flow rate constant. The back pressure built up by the motor not only causes leakage flow in the motor and pump but oil in the lines to compress. Volume compressibility flow is essentially proportional to pressure and therefore the tariff of flow is proportional to the rate of innovations of pressure, i.e., $qc \propto dP/dt$ Compressibility flow rate:

$$q_c = K_c \frac{dp}{dt} \tag{4}$$

where, K_c is coefficient of compressibility. The rate at which the oil flows from the pump is given by sum of oil flow through the motor, leakage flow rate and compressibility flow rate:

$$q_p = q_m + q_i + q_c$$

Substituting (Eq. 1-4) from above Equations, we get:

$$K_p x = K_m \frac{d\theta}{dt} + K_i P + K_c \frac{dP}{dt} \tag{5}$$

The torque T_m developed by the motor is proportional to pressure drop and balances load torque. Hydraulic motor torque:

$$T_m = K_t P \tag{6}$$

where, K_t is motor torque constant. The load has a moment of inertia J and viscous friction coefficient B, Then Load Torque:

$$T_l = J\frac{d^2\theta}{dt^2} + B\frac{d\theta}{dt} \quad (7)$$

Hydraulic power input:

$$= q_m P \quad (8)$$

Substituting (Eq. 2) into (Eq. 8), we get:

$$\text{Hydraulic power input} = K_m \frac{d\theta}{dt} P \quad (9)$$

$$\text{Mechanical power output} = T_m \frac{d\theta}{dt} \quad (10)$$

Substituting (Eq,. 6) into (Eq. 10), we get:

$$\text{Mechanical power output} = K_t P \frac{d\theta}{dt} \quad (11)$$

If the losses of the hydraulic motor are neglected, then the mechanical power output is equal to hydraulic motor input:

$$K_m \frac{d\theta}{dt} P = K_t P \frac{d\theta}{dt} \quad (12)$$

From (Eq. 12), it is clear that $K_m = K_t$. Hence, we can write:

$$T_m = K_t P = K_m P$$

Since, the load torque equals motor torque, $T_m = T\ l$:

$$K_m P = J\frac{d^2\theta}{dt^2} + B\frac{d\theta}{dt} \quad (13)$$

$$P = \frac{J}{K_m}\frac{d^2\theta}{dt^2} + \frac{B}{K_m}\frac{d\theta}{dt} \quad (14)$$

Differentiating (Eq. 14) w.r.t time, we get:

$$\frac{dP}{dt} = \frac{J}{K_m}\frac{d^3\theta}{dt^3} + \frac{B}{K_m}\frac{d^2\theta}{dt^2} \quad (15)$$

Substituting for P and dP/dt to (Eq. 5), we get:

$$Kp\frac{dx}{dt} = K_m\frac{d\theta}{dt} + K_i\left[\frac{J}{K_m}\frac{d^2\theta}{dt^2} + \frac{B}{K_m}\frac{d\theta}{dt}\right] + K_c\left[\frac{J}{K_m}\frac{d^3\theta}{dt^3} + \frac{B}{K_m}\frac{d^2\theta}{dt^2}\right] \quad (16)$$

Taking Laplace transform we get:

$$\frac{\theta(s)}{X(s)} = \frac{K_p}{s\left[\frac{K_c J}{K_m}s^2 + \left[\frac{K_i J + K_c B}{K_m}\right]s + \frac{K_m^2 + K_i B}{K_m}\right]} \quad (17)$$

The speed become:

$$\omega(s) = s\theta(s) \quad (18)$$

Substitute (Eq. 20) into (Eq. 19) yields:

$$\frac{\omega(s)}{X(s)} = \frac{K_p}{\left[\frac{K_c J}{K_m}s^2 + \left[\frac{K_i J + K_c B}{K_m}\right]s + \frac{K_m^2 + K_i B}{K_m}\right]} \quad (19)$$

Rearranging (Eq. 19) the final transfer function becomes:

$$\frac{\omega(s)}{X(s)} = \frac{K}{s^2 + \alpha s + \beta} \quad (20)$$

Where:

$$K = \left(\frac{K_p K_m}{K_c J}\right)$$

$$\alpha = \left(\frac{K_i J + K_c B}{K_c J}\right)$$

$$\beta = \left(\frac{{K_m}^2 + K_i B}{K_c J}\right)$$

Table 1 shows the parameters of the system. The transfer function of the tire remover system is:

Table 1: System parameters

Parameters	Symbols	Values
Ratio of rate of oil flow to unit stroke displacement	K_p	0.91 m^2/sec
Motor displacement constant	K_m	13.73 m^3/rev
Leakage flow rate constant	K_i	12.25 m^3/pa.s
Coefficient of compressibility	K_c	0.5
Load moment of inertia	J	10 N. m sec^2/rev
Load viscous friction	B	5 N. m sec/rev

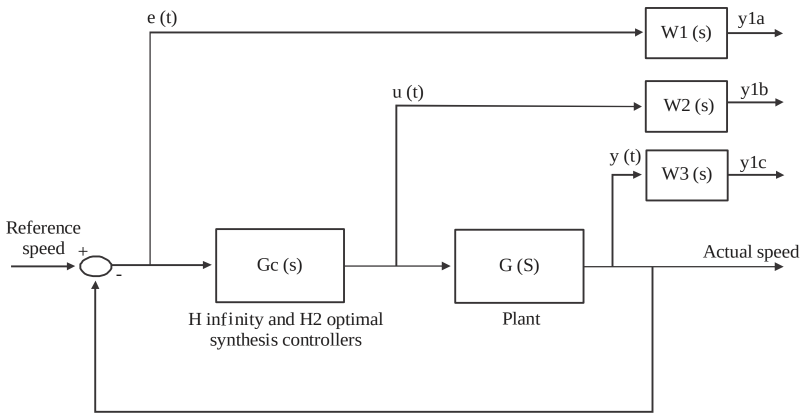

Fig. 2: Tire changer with H∞ and H_2 optimal synthesis controllers

$$\frac{\omega(s)}{X(s)} = \frac{2.5}{s^2 + 25s + 50}$$

The state space representation becomes:

$$\dot{x} = \begin{pmatrix} -25 & -50 \\ 1 & 0 \end{pmatrix} x + \begin{pmatrix} 1 \\ 0 \end{pmatrix} u$$
$$y = \begin{pmatrix} 0 & 2.5 \end{pmatrix} x$$

Proposed controllers design

Augmentation based H∞ and H_2 optimal synthesis controllers design: In this study, we evidence center on the weighted control structure design for H∞ and H_2 optimal synthesis controllers shown in Fig. 2 where $W_1(s)$, $W_2(s)$, and $W_3(s)$ are weighting functions or weighting filters. The assumption that G(s), W1(s) and W3(s) G(s) are all proper; i.e., they are bounded when $s\rightarrow\infty$. It can beseen that the weighting function $W_3(s)$ is not required to be proper.

One may marvel why we need to utility three weighting functions in Fig. 2. First, we memo that the weighting functions are respectively, for the three signals, namely, the error, the input and the output. In the two-port state space structure, the output vector y1 = [y1a, y1b, y1c] T is not used directly to construct the control signal vector u2.

We should understand that y1 is actually for the control design characteristic measurement. So, it is not strange to include the filtered "input signal" u(t) in the "output signal" y1 because one may requirement to measure the sovereignty energy to assess whether the designed controller is good or not. Clearly, Fig. 2 represents a more general picture of H∞ and H_2 optimal synthesis control systems. The weighting functions can also be regarded as filters. This type of frequency-dependent weighting is more practical. We will bazaar next that given the weighting transfer functions, we can design an H∞ and H_2 optimal synthesis controllers by using the impression of the augmented state space model. The weighting function $W_1(s)$, $W_2(s)$ and $W_3(s)$ are chosen as:

$$W_1(s) = 0.1\frac{s+100}{100s+1} \quad W_2(s) = 0.1 \quad W_3(s) = [\ ]$$

The H∞ synthesis controller become:

$$G_{cH} = \frac{6.24s^2 + 29.39s + 48.43}{s^3 + 25.22s^2 + 55.46s + 0.5521}$$

The H_2 optimal synthesis controller become:

$$G_{cH_2} = \frac{0.8158s^2 + 20.39s + 40.79}{s^3 + 25.05s^2 + 51.26s + 0.5101}$$

RESULTS AND DISCUSSION

Comparison of the tire changer with H∞ and H_2 optimal synthesis's controllers for a desired step reference input: The Simulink Model for the tire charger with H∞ and H_2 optimal Synthesis controllers for a desired step input speed is shown in Fig. 3. For a desired speed change from 0 to 1 rev/m input, the tire changer speed response simulation is shown in Fig. 4.

For operating the system with 1 rev/min, the simulation result shows that the tire changer with H∞ synthesis controller have a better rise time, smaller percentage overshoot and improved settling time than the tire changer with H_2 optimal synthesis controller[3, 4].

Comparison of the tire changer with H∞ and H_2 optimal synthesis's controllers for a desired random

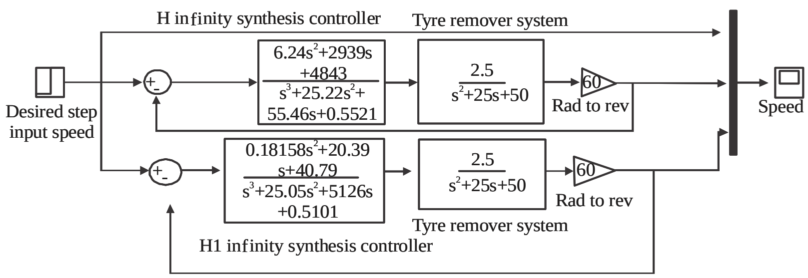

Fig. 3: Simulink model for the tire changer with H∞ and H_2 optimal Synthesis controllers for a desired step input speed

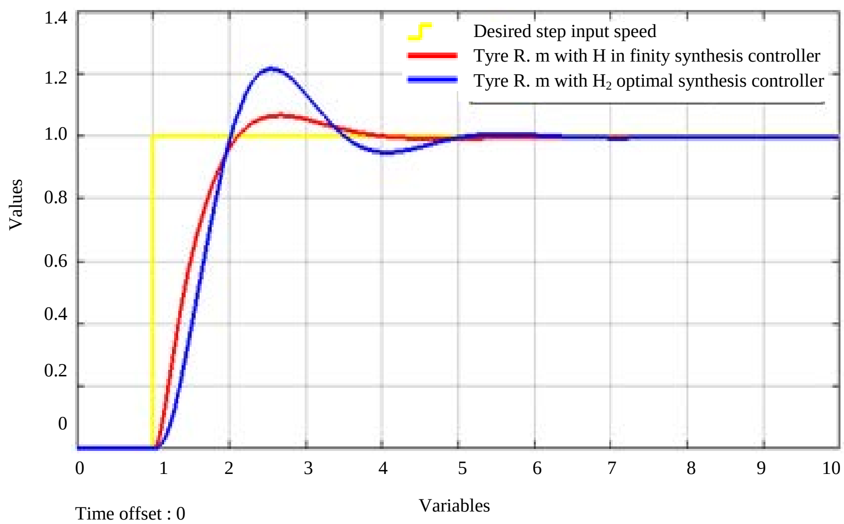

Fig. 4: Tire changer speed response simulation for a step input

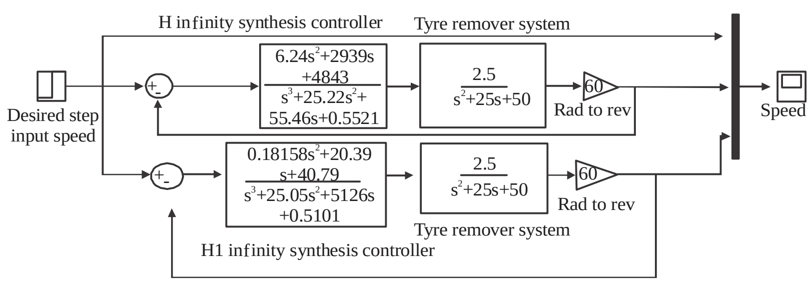

Fig. 5: Simulink model for the tire changer with H∞ and H_2 optimal synthesis controllers for a desired random input speed

Reference input: The Simulink Model for the tire changer with and optimal Synthesis controllers for a desired random input speed is shown in Fig. 5. For a desired random speed change from 0-1 rev/m input, the tire changer speed response simulation is shown in Fig. 6.

For operating the system with a random speed, the simulation result shows that the tire changer with H∞ synthesis controller have track the random input speed with small overshoot than the tire changer with optimal H_2 synthesis controller[5].

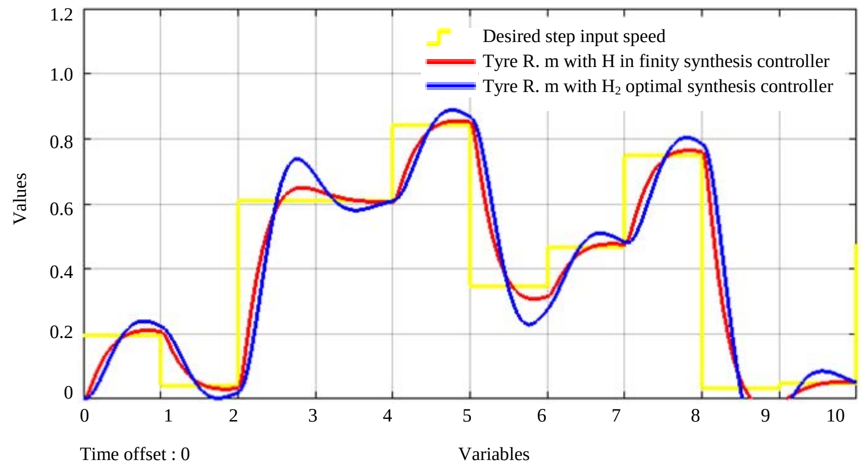

Fig. 6: Tire changer speed response simulation for a random input

CONCLUSION

In this study, a hydraulic system based vehicle tire changer is modelled and designed using a hydraulic pump and motor. The machine is controlled using a leg pedal displacement input to bush the pump piston in order to pressurize the hydraulic fluid to the motor. The motor rotates the tire with an angular speed to dismount and mount the tire. For improving the performance of the machine a robust controller have been used. And optimal synthesis controllers have been used to improve the tire changer speed of rotation. Comparison of the proposed controllers have been analyzed and simulated using two reference inputs (step and random). Finally, the comparative simulation results proved the effectiveness of the proposed tire changer with synthesis controller in improving the settling time and percentage overshoot.

REFERENCES

01. Abebe, M., K. Akele, M. Solomon, M. Bancha and T.D. Megiso, 2018. Design of tire changer. Int. J. Eng. Trends Technol., 64: 8-23.
02. Pornima, T., 2016. Machine for tire removing and fitting on rim of wheel: A review. Int. J. Sci. Res. Dev., 4: 52-54.
03. Gururaj, S., M.S. Bhairavnath and M. Madan, 2017. Study and analysis of tire changing machine components. Int. Res. J. Eng. Technol. (IRJET.), Vol. 4, No. 6.
04. Felix, D.G., K. Sridhar and M. Puviyarasa, 2014. Quick tyre changer. Int. J. Mech. Prod. Eng., 2: 57-59.
05. Patel, H.V., B.D. Kayasth, M.E. Scholar, M. Design and A. Sal, 2015. A literature review on design and analysis of tyre envelope expander for tyre retreading process. Int. J. Eng. Innovation Sci. Res., 1: 10-13.