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Earthmoving Vehicle Powertrain Controller Design and Evaluation

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***Abstract*— Previous work [13,14] examined the control of a nonlinear Multi-Input Multi-Output electrohydraulic system representative of an earthmoving vehicle powertrain. The system layout incorporates a prime mover, a variable displacement pump, several flow valves and motors representing different loads on the powertrain. A gain-scheduled H_∞ controller [14] was successfully developed and tested on a Hardware-in-the-Loop (HIL) system. However, previous results were for relatively simple reference tracking tasks. In this paper, the previously developed systems and load emulation environments are implemented and used to provide a test environment for the human-centered evaluation of powertrain controller designs. Measures for performance and efficiency are first discussed and defined for the earthmoving powertrain. The H_∞ control problem is then presented along with a brief summary of the Earthmoving Vehicle Powertrain Simulator (EVPS) plant model developed previously. Various design plant models (DPMs), based on maximizing performance and/or efficiency, are described and controllers are synthesized using commonly available H_∞ control design software. Experimental comparisons of working cycle times, tracking performance, system efficiencies, and component efficiencies are made between the different controller designs. The task used for comparison is a 180-degree closed-loop working cycle for a medium sized wheel loader.**

I. INTRODUCTION

IN general, an earthmoving vehicle's powertrain can be thought of as a Multi-Input Multi-Output fluid power system in which the power is generated by a prime mover and then is distributed throughout the machine as shown in Fig. 1. Current generations of earthmoving machines rely mostly on human-in-the-loop coordination for completing different tasks. In contrast to the multiple-pump human-coordinated system in most current applications, an automatically coordinated control structure with a single-pump was proposed in [14]. A model-based multivariable controller is designed in two steps to coordinate the inter-load interaction and to compensate for the nonlinearities. Robust control methods such as H_2 and H_∞ algorithms were utilized in [14] to design linear local multivariable controllers for the linear local model evaluated in the neighborhood of some operating condition. These algorithms were then tested on the hardware-in-the-loop (HIL) system termed the EVPS.

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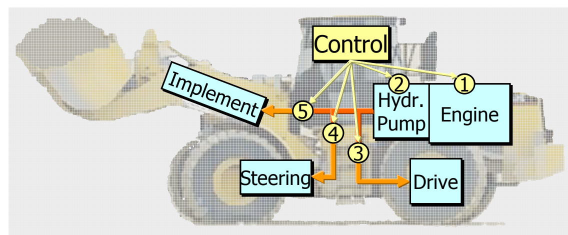

Fig. 1. Earthmoving vehicle powertrain layout.

In order to realistically evaluate the nonlinear MIMO algorithms developed in [13], the EVPS, shown in Fig. 2, had to incorporate detailed models of the loads that would be encountered by a typical machine. References [1] and [2] describe in detail the loading models that were developed to ensure the EVPS system faced loads similar to an actual machine.

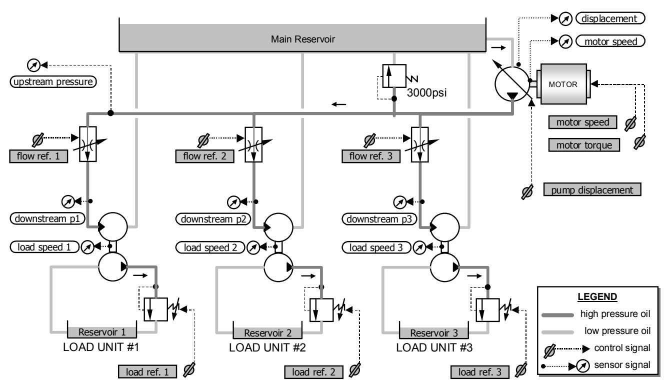

Fig. 2. EVPS schematic.

The loading environment and powertrain models were combined into a system that allowed for humans to interact with a realistic Human plus Hardware-in-the-Loop (H+HIL) experiment. Fig. 3 shows a schematic and information flow chart for the entire system.

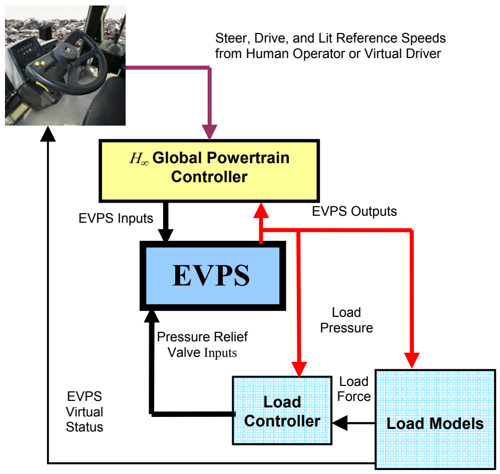

Fig. 3. Human + HIL layout.

Previous works by the authors outlined the controller design strategy and H+HIL environment, including the challenging task of realistic load emulation. However, the overall controller and H+HIL system had not yet been tested on realistic loading cycles. This paper gives a quantitative evaluation of these robust H_∞ powertrain controllers acting under human control in realistic working cycle scenarios. The controllers include an aggressive fuel-hungry controller that provides good performance at a steep cost as well as a fuel-thrifty controller which sacrifices performance for reduced fuel economy.

Emulated 180-degree working cycle experiments used for powertrain controller evaluations are described in detail in [1]. The working cycle, shown in Fig. 4, consists of (1) driving the loader to a loading site/pile, digging, (2) backing up and steering towards the dump site/truck, raising the implement, (3) driving to the truck, dumping, and then (4) backing away and lowering the implement, and steering towards the loading pile to complete the cycle.

Fig. 4. Earthmoving working cycle.

The remainder of the paper is presented as follows. Section II establishes the performance and efficiency measures that will be used to evaluate powertrain controller designs. The H_∞ control problem is presented in Section III along with a description of three design plant models (DPMs) used to emphasize particular design goals. Section IV gives experimental results and compares the efficiency and performance of the three controller designs for emulated working cycles. A conclusion then summarizes the paper and describes the direction of future research.

II. Performance and Efficiency Measures

To quantitatively evaluate the performance and efficiency characteristics of powertrain controllers, the proper metrics must first be defined. Efficiency may be defined here as the ratio of power output to power input. In terms of system efficiency, this equates to the ratio of the power output of the EVPS load motors to the power input to the prime mover. For an emulated working cycle, this is the ratio of the power output of the load motors to the power output of the emulated diesel engine. Overall efficiency is a product of the individual system component efficiencies as described by (1)-(4) [1].

$$\eta_{sys} = \eta_{engine} \cdot \eta_{pump} \cdot \eta_{valve} \tag{1}$$

$$\eta_{engine} = \frac{\omega_e - \omega_{optimal}}{\omega_{optimal}} \cdot 100\% \tag{2}$$

$$\eta_{pump} = \frac{P_u \cdot D_p \cdot \omega_e}{T_{el} \cdot \omega_e} \cdot 100\% \tag{3}$$

$$\eta_{valves} = \frac{\sum P_i Q_i}{P_u Q_p} \cdot 100\% \tag{4}$$

Maximizing the system efficiency involves maximizing the individual EVPS component efficiencies. The efficiency of valves can be controlled to a certain extent by careful coordination of the control inputs in order to operate at 'ideal' conditions. For the valve manifold, it is desired that upstream pressure be minimized while at the same time maximizing downstream pressure in order to minimize the pressure drop across the valve.

Fuel economy, emissions, and/or performance may dictate the 'ideal' operating conditions for the engine. The ideal brake specific fuel consumption curve (IBSFC) is optimal in terms of power produced per unit of fuel input to the engine [3]. In order to maximize efficiency, the engine should be controlled to operate on this curve. The EVPS emulated diesel engine model's [5,11] normalized IBSFC is shown in Fig. 5 [1]. At low power output, the engine is operating at low speeds relative to the maximum engine speed. This may cause undesired sluggish performance when a high load is encountered as the operator must wait for the engine to come up to the higher operating speed before the desired power is available for use. Therefore, it may be more beneficial to use the maximum torque curve or other pre-defined curve in the operating domain when performance is a key consideration.

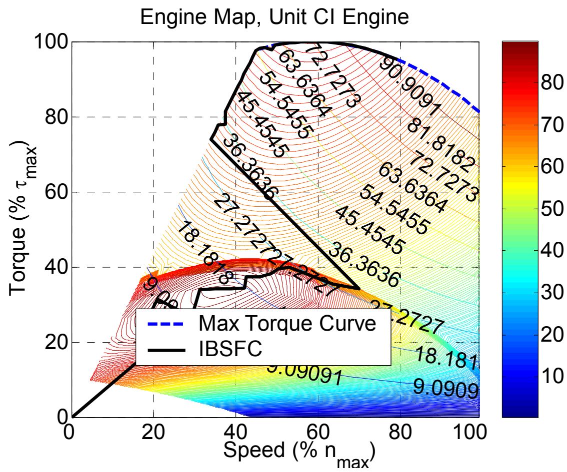

Fig. 5. Ideal brake specific fuel consumption [1].

An alternative definition of total system efficiency, is working cycle fuel consumption. Fuel consumption is directly proportional to the integral of the product of the fuel index, γ, and engine speed, ω_e, as given by (5) [11]. C is a constant engine parameter.

$$m_f = C\int \gamma \cdot \omega_e dt \tag{5}$$

The performance of a working cycle will be evaluated based on the time required to successfully complete all working cycle tasks shown in Fig. 4. While cycle time is an intuitive measure of controller performance, the *quality* of the cycle should also be considered. The quality of the cycle represents the ability of the powertrain controller to coordinate the inputs to the system to track the speed references provided by the operator with minimum tracking error in the face of load disturbances. This measure, coupled with cycle time, will provide a more complete picture of controller performance. Tracking error is an average value of the three individual average tracking errors over the entire working cycle as given by the following:

$$e_{track} = \frac{\sum_{i=1}^{3} e_{i\_avg}}{3} \tag{6}$$

III. H_∞ Controller Design

An introduction to the generalized H_∞ control problem will be presented below. This framework will provide the foundation for the controllers that will be designed. Fig. 6 shows the general control configuration described by the following state-space realization of the generalized plant G [9]:

$$\begin{bmatrix} z \\ v \end{bmatrix} = G(s)\begin{bmatrix} w \\ u \end{bmatrix} = \begin{bmatrix} G_{11}(s) & G_{12}(s) \\ G_{21}(s) & G_{22}(s) \end{bmatrix}\begin{bmatrix} w \\ u \end{bmatrix} \tag{7}$$

$$u = K(s)v \tag{8}$$

$$G = \left[\begin{array}{c|cc} A & B_1 & B_2 \\ \hline C_1 & D_{11} & D_{12} \\ C_2 & D_{21} & D_{22} \end{array}\right] \tag{9}$$

where u is the vector of control input signals, v is the measured outputs, z is the error signal vector, and w is the exogenous input vector of disturbances and reference commands.

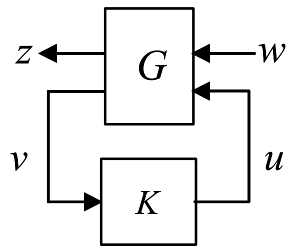

Fig. 6. General control configuration [9].

The closed-loop transfer function from the disturbance inputs w to the error signals z is of interest. It can be represented by the linear fractional transformation (LFT) given by (10) and (11).

$$z = F_l(G,K)w \tag{10}$$

$$F_l(G,K) = G_{11} + G_{12}K(I - G_{22}K)^{-1}G_{21} \tag{11}$$

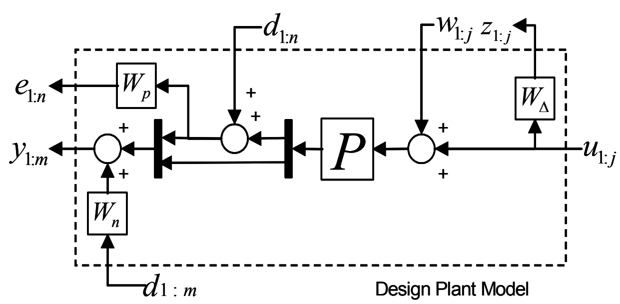

Fig. 7. Example of a design plant model (DPM).

The goal of the H_∞ control problem involves the selection of a controller K that minimizes the $\|F_l(G,K)\|_\infty$ norm associated with the matrix given by Equation 12. The controller that is designed will be of the same dimension as the generalized plant G, and in general, the controller is a sub-optimal controller that satisfies Equation for a given value of γ [9].

$$\|F_l(G,K)\|_\infty < \gamma \tag{12}$$

In order to design an H_∞ powertrain controller for the EVPS, a plant model, G, of the system is required. The

EVPS consists of an AC motor, a variable displacement pump, three proportional flow valves, three gear motors, and four main high-pressure hoses. Previous research involved modeling the individual components of the system, interconnecting them to form a system model, and validating the models through system identification techniques [7,8,14].

The design associated with the H_∞ control problem involves the selection and tuning of the DPM weighting matrices in Fig. 7. Performance, noise, model uncertainty, and input weightings (not shown) are design parameters than can be manipulated to achieve desired controller characteristics. In [1], approximately 20 DPM variations were used and the resulting controller performance and efficiency measured using emulated 180-degree working cycles on the EVPS. Herein, the three basic designs will be detailed; the performance design (PD), manifold efficiency design (ME), and engine efficiency design (EE).

A. *Performance Design (PD)*

A DPM that will be referred to as the performance design (PD) is constructed to provide a pure performance related controller (i.e. minimize cycle time). The PD DPM includes integral type engine and load speed error weightings [1], [12]. The powertrain controller will attempt to track the engine speed associated with the total maximum engine power. This ensures that the engine is operating on its maximum torque curve and will be able to provide, on demand, the required power to complete the desired actions demanded by the operator. This, however, is inefficient as the prime mover is often times producing more power than is being utilized by the load actuators. The excess power is simply dissipated through the upstream pressure relief valve as high pressure hydraulic fluid.

The desired engine speed will be tracked while a fuel index (equivalent to throttle for an SI engine) of 100% is provided to the diesel engine. This will ensure that the engine is always operating on its maximum torque/power curve. Since the fuel index command is not available to control engine speed, the pump displacement will be used to control the engine speed. This is equivalent to using a transmission to alter engine operating speed similar to a continuously variable transmission (CVT) [4,6,10].

B. *Manifold Efficiency Design (ME)*

One of the areas in which efficiency of the powertrain can be increased is at the valve manifold (see Eq. 1). The manifold has three individual pressure drops associated with it. The larger the pressure drop, the lower the efficiency of the manifold. In order to incorporate pressure drop minimization into a control objective, the output matrix of the linearized EVPS plant model is altered to include the pressure differential across the valves in the place of the three downstream load pressures. This approach results in the following modified state-space output matrix [1]:

$$\delta y = C\delta x$$
$$= [\delta n_e, \delta P_u \mid \delta Pd_1, \delta Pd_2, \delta Pd_3 \mid \delta n_{m1}, \delta n_{m2}, \delta n_{m3}]^T \quad (13)$$

where Pd_i represents the pressure drop across the i^{th} valve. By including the pressure differentials in the output equation, the controller will attempt to minimize them as a part of the minimization of the matrix norm $\|F_l(G,K)\|_\infty$. This design will be referred to as the manifold efficiency (ME) DPM. This design does not include the integral performance weighting on engine speed that is included in the PD.

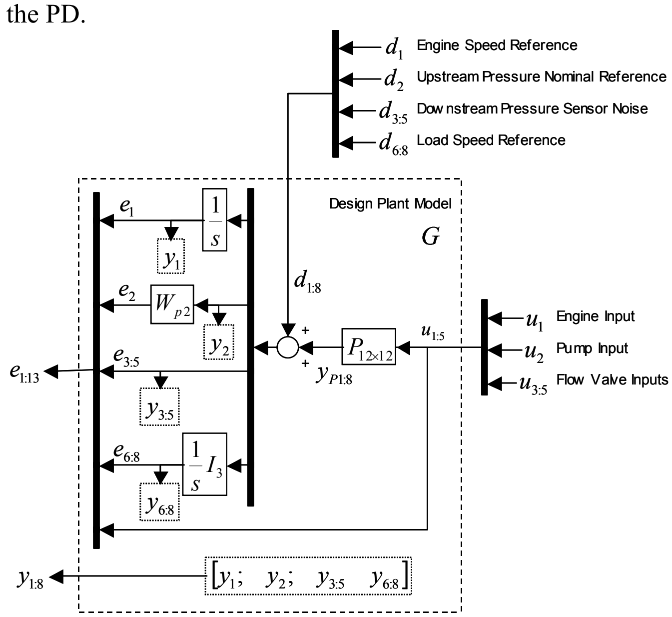

Fig. 8. PD DPM [12].

C. *Efficiency Design (EE)*

In an attempt to achieve an upper bound on cycle efficiency by tracking an engine efficiency map such as the IBSFC curve, there are two possible designs: the engine fuel index can be maintained as a part of the powertrain controller, or the fuel index and pump displacement inputs can be separated from the powertrain controller. In this design termed the engine efficiency (EE) DPM, the engine and pump controls are separated from the authority of the powertrain controller and are controlled via an engine/transmission controller as described by Fig. 9 and Fig. 10. This allows the control designer to exploit the continuously variable transmission (CVT) that is available with a variable displacement pump.

The powertrain controller will be designed on a plant model that has a flow input in place of the engine and pump inputs in the other designs. The desired flow is multiplied by the current upstream pressure to generate a desired power demanded by the powertrain. The engine controller is responsible for generating the torque that corresponds to the desired power level. A fuel index command is given to the engine based on efficiency maps from the IBSFC curve for the diesel engine. The pump is responsible for acting as a CVT for the powertrain. By adjusting the displacement, the engine can be loaded to track the desired speed. The pump controller shown in Fig. 10 is a PD type control algorithm.

Theoretically, this design will provide only the power that is required by the powertrain to track the desired load motor reference speeds. Power will be produced at the most efficient point on the engine operating map by following the IBSFC curve. This will provide the highest possible engine efficiency and, consequently, the lowest working cycle fuel consumption.

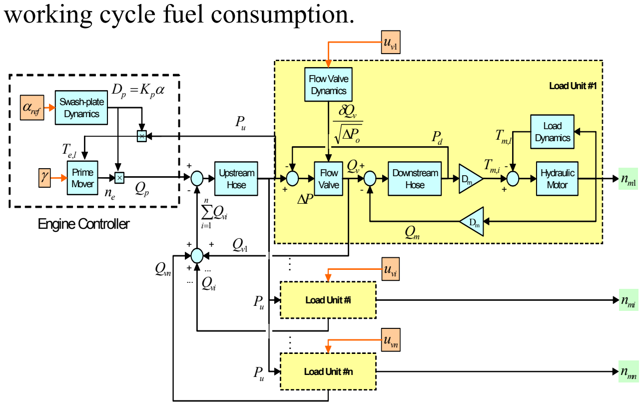

Fig. 9. Engine and pump controller subsystem relative to overall powertrain controlled system [1].

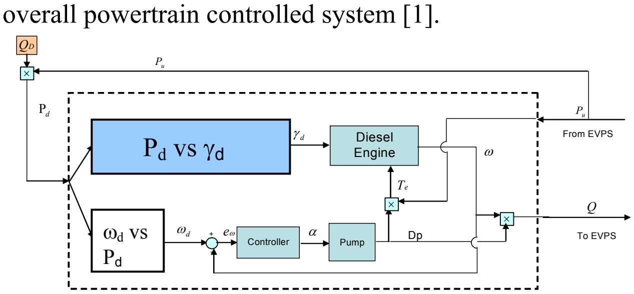

Fig. 10. Detailed engine and pump controller for use with EE DPM [1].

IV. Results

With the DPMs in the previous section now defined and summarized in Table I, the controllers were then synthesized utilizing the Matlab™ μ-Analysis Toolbox. Each controller design was evaluated three times and averaged utilizing emulated 180-degree working cycles with a computerized driver on the EVPS [1].

The results for performance and efficiency measures are given in Table I. It is clear that PD design has the best performance of the three designs in terms of cycle time. This is of no surprise as the design dictates that the maximum power of the emulated diesel engine is available for use by the powertrain on demand. As a direct result of this minimum cycle time, the fuel usage associated with this design is almost 2.5 times that of the other designs. The tracking error associated with this design is also the highest on average of the three designs. This can be attributed to high overshoots caused by excessive hydraulic flow being sent to the loads during transient operations.

The ME design can be considered a compromise between efficiency and performance. It has a slightly higher cycle time than that of the PD design, but has a much lower tracking error. This implies that with a decrease in cycle time, the quality of the cycle can be increased by use of this controller. The other advantage is clearly the increased fuel economy over the PD design. Fig. 11 shows the tracking results and efficiency profiles for the ME design.

Table I
Powertrain Controller Summary and Working Cycle Results

DPM	PD	ME	EE
Engine Speed Tracking	**X**		**X**
Valve Efficiency Considered		**X**	
Design Type	Performance	Peformance/ Efficiency	Efficiency
Description	Track Max Engine Power	Minimize Valve Pressure Differential	Separate Engine and Pump Control
Engine Efficiency (%)	***87.8***	**-**	***86.4***
Pump Efficiency (%)	***29.3***	***51.1***	***66.2***
Manifold Efficiency (%)	***50.6***	***56.9***	***40.19***
Fuel Consumption (kg)	***0.82***	***0.31***	***0.30***
Tracking Error (rad/s)	***21.5***	***14.2***	***21.0***
Cycle Time (sec)	***31.7***	***32.4***	***32.4***

The final design tested using emulated working cycles on the EVPS was the EE design. Structurally, this design is much different. The powertrain controller has no control authority over the diesel engine fuel index or the pump displacement. It simply provides a desired flow reference to the engine/transmission controller. The goal of the EE design is to minimize fuel consumption. It does accomplish this, but by only 0.01 kg/cycle. The cycle time while identical to that of the ME design, the tracking error is almost 150% higher.

The EE design also performs poorly when more than one load is encountered by the powertrain. This can be explained by the fact that during times of multiple loads coming online, the power demand increases dramatically. The diesel engine power production transients are then realized by the powertrain as it waits for the power to become available. This results in very sluggish behavior. One alternative to this approach that will not be described in detail here is to set a minimum power level for the engine to operate at. This approach has been shown to decrease the tracking error, but results in higher fuel consumption. Tracking the IBSFC curve, can also be difficult when the maximum pump displacement is not capable of providing enough load torque to the engine [1].

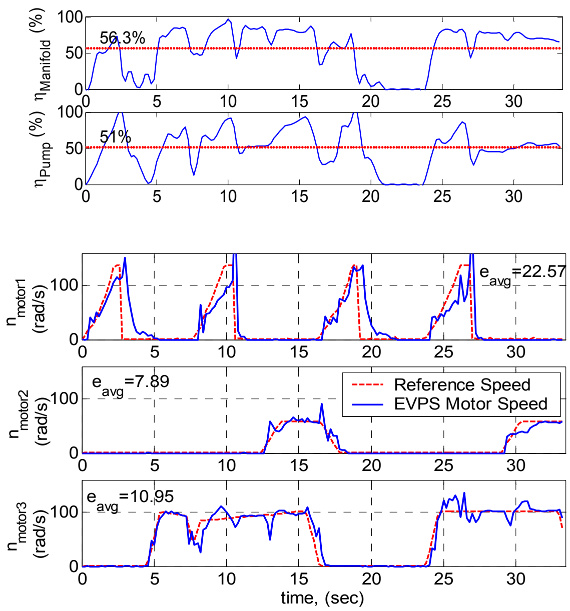

Fig. 11. Working cycle results for ME; component efficiencies (top), load tracking performance (bottom) [1].

V. Conclusion

Three robust H_∞ powertrain controller designs have been presented. The three designs were tested using the EVPS at UIUC in emulated 180-degree working cycles. The results show that cycle time, fuel economy, or a combination can be emphasized in the final controller design by manipulating the DPM weightings and structure. The PD design was shown to minimize cycle time at the cost of increased tracking error and decreased fuel economy. The EE design, while theoretically attractive, was not as impressive when implemented. While giving the lowest fuel consumption over a cycle, its sluggish behavior strongly outweighs its benefits. For a compromise between cycle time and fuel economy, the ME design is a good choice. It provides fuel economy comparable to the EE design with slightly higher cycle times than the PD design but with much better load tracking.

The controller evaluations provided here lay a foundation for further research into performance and efficiency of the multi-load hydraulic powertrain. It has been successfully shown that by selecting appropriate DPM designs, efficiency or performance can be emphasized in the H_∞ controller. This can result in decreased operation times or alternatively, decreased operating costs. Although the differences in Table 1 may be small, over the course of thousands of cycles, the savings in terms of time and fuel can be substantial. Recommendations on which controller is 'best' is strictly dependent on the economics of the situation. It becomes an opportunity cost optimization between cycle time, quality, and fuel economy.

Acknowledgment

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