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Design of a highly efficient friction clutch apparatus for vehicle applications using a self-energizing mechanism

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Abstract

This study proposes a new design for a friction clutch actuator using the self-energizing principle for vehicle applications such that the power consumption for clutch control is significantly reduced. The self-energizing effect can be created by simply adding wedge structures to a conventional clutch system, and it assists in significantly reducing the actuation energy of the clutch with little additional cost. In this paper, a mathematical model of the clutch actuation system is derived on the basis of static force analyses with particular emphasis on the torque amplification factor due to the self-energizing effect. The slope angles of the wedges in the proposed clutch actuator are determined in order that the clutch system ensures appropriate torque amplification while considering various factors such as the variations in the friction coefficient and the return spring force. In addition, model-based analyses of the new clutch actuator system are performed in order to predict the dynamic effects of the self-energizing mechanism on the system, particularly for the clutch engagement process. The feasibility of the proposed clutch design and its high energy efficiency are verified experimentally using three prototypes with different slope angles.

Keywords

Automotive clutch, clutch actuator, friction clutch design, self-energizing effect, torque amplification

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Introduction

Improving the energy efficiency of ground vehicles has been a key issue in global environmental problems and the global energy crisis. For decades, there has been significant progress in increasing the efficiency of individual automotive systems that include internal-combustion engines, power transmission systems and other hydraulic and electrical components. Furthermore, the efficiency of delivering the mechanical power produced by an engine is an important part of the energy efficiency improvements. The engine power is predominantly transmitted to the wheels by the transmission systems, but it is also used to drive various components mechanically such as turbochargers and air conditioning (A/C) systems. In order to engage or disengage the power transmissions by using these subsystems, friction clutches (wet or dry) are commonly used. Because automatic control of the clutches performed by several types of actuator (e.g. hydraulics and motors) requires non-negligible power consumption, it is desirable to reduce the energy consumed for clutch actuation. Moreover, improving the efficiency of the clutch actuators is increasingly important because power transfer control and distribution control, such as those in four-wheel drive systems and hybrid electric drive systems, are becoming increasingly popular.[1, 2]

Previously, many studies have been carried out on new actuator designs which improve the performance of conventional systems in various fields, e.g. a rotary linear actuator using piezoelectric translators,[3] a serial dual actuator with a planetary gear train[4] and an electromechanical clutch actuator for automated manual

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transmissions.[5] For electromagnetic actuators, researchers have attempted to reduce the coil resistance and to optimize the magnetic flux in order to reduce the power consumption of the actuators.[6, 7] In addition to the previous research, a more fundamental change is needed as a solution to the problem while simultaneously considering the practical aspects. Hence, this paper proposes a highly efficient clutch actuator design using the self-energizing principle for a vehicle A/C system as an example of an automotive friction clutch.

Using the self-energizing principle, the actuation energy can be significantly reduced in order to create the same amount of clutch engagement force. The self-energizing mechanism recycles the frictional energy, which is normally dissipated in the clutch during its engagement, to boost the actuation force. This mechanism has already been used in several vehicle applications, including conventional drum brakes, synchronizers and electronic wedge brakes (EWBs).[8, 9] Fujii et al.[10] developed a mathematical model of a wet band brake to describe its dynamic properties and self-energizing mechanism. Jo et al.[11] and Park and Choi[12] investigated the self-energizing effect of newly designed EWBs and controlled them. Efficient clutch actuators for automotive transmission systems have also been developed on the basis of the self-energizing principle.[13, 14] In these studies, the torque amplification of the actuators was achieved using non-circular gears or racks arranged in a wedge shape. Yao et al.[15] examined the control problems of a self-energizing clutch for automatic transmissions by considering its dynamic behaviour.

In this paper, a new electromagnetic clutch for a vehicle A/C system is implemented by adding wedge blocks to the clutch disc of a conventional system. Because of mass production, this design must be cost effective; by adding only small inexpensive components to the system, its power consumption is significantly reduced in proportion to the amplification factor of the engagement force, and the actuation coil size can also be reduced accordingly.

The remainder of this paper is organized as follows. In the second section, the concept of the self-energizing mechanism is introduced, and the mathematical model for the proposed clutch actuator is derived. The design considerations for the wedge clutch system and detailed discussions are presented in the third section. In the fourth section, model-based analyses of the clutch system are made in order to predict the influences of the self-energizing mechanism on the clutch engagement process. Finally, in the fifth section, the validity and effectiveness of the proposed clutch actuator are verified experimentally using three different prototypes on an A/C test bench.

Mathematical formulation of the self-energizing mechanism

System overview

The structure of the proposed wedge clutch system is depicted in Figure 1. A few components including wedge structures are added to the conventional system in order to induce the self-energizing effect. Here, the friction clutch functions as a compressor controller which activates or deactivates the vehicle A/C system. The clutch system is composed of three main parts: the disc assembly, the pulley assembly and the field coil assembly. The disc assembly transfers the drive power from the pulley to the compressor, and the field coil assembly generates the electromagnetic force to engage the disc to the pulley. Regardless of the clutch engagement, the pulley rotates continuously, and it is connected to the engine via a belt. When the clutch is disengaged, the disc and the hubs are not rotating and

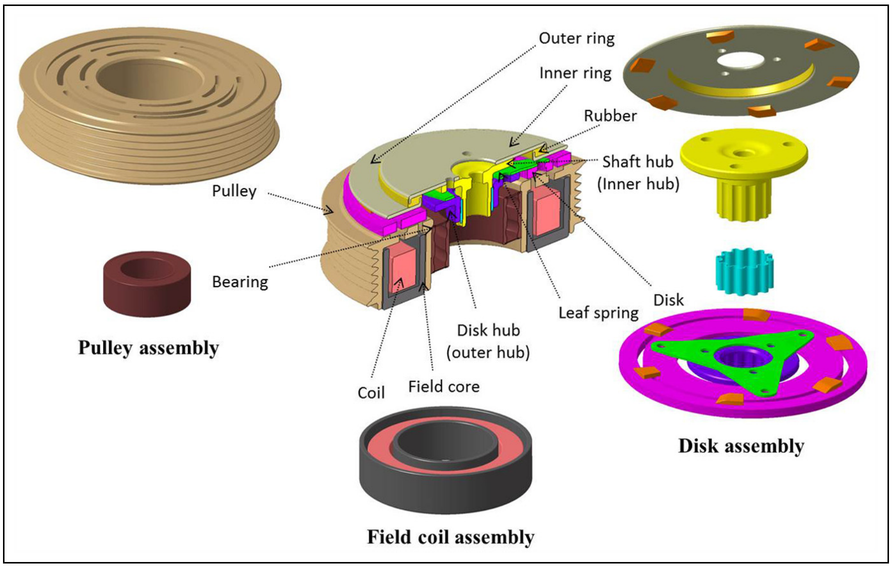

Figure 1. Wedge clutch structure.

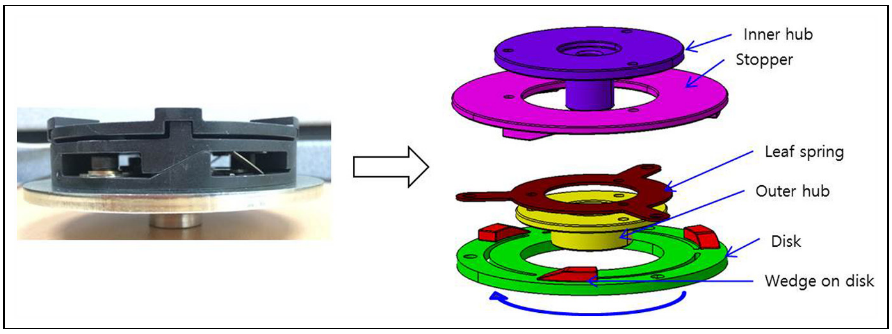

Figure 2. Clutch disc assembly.

the compressor is deactivated. If the compressor is required, the current is controlled to flow through the coil, and the created magnetic force acts on the disc to engage the clutch to the pulley. The disc moves towards the pulley and begins to rotate by the braking force (the frictional force) of the compressor owing to the contact force with the pulley. Once the current is switched off, the disc moves back to its original position. This procedure is performed by the restoring force of a leaf spring. A torsional spring can also be installed in the system to restore the disc more easily.

Self-energizing mechanism

The disc assembly of the wedge clutch system is presented in Figure 2. As the wedge and stopper structures are installed in addition to the standard clutch, the self-energizing effect occurs such that the engagement force is amplified. Both the stopper and the pulley are physically fixed in the vertical direction. The magnetic force brings the wedges of the disc into contact with the wedges of the stopper during engagement. The normal force acting on the surface of the disc wedges presses the disc against the pulley, which forces it to engage more deeply and also to press against the surface of the wedges on the stopper. This process occurs repeatedly and causes the vertical normal force on the disc to become significantly larger than the actual magnetic force, particularly with the increased friction coefficient between the pulley and the disc.

Amplification factor of the clutch engagement force

The self-energizing effect can be interpreted as the amplification factor of the clutch engagement force. The amplification of the engagement force implies that a smaller magnetic force and a lower current are required to actuate the clutch system than in the conventional system. The amplification factor can be derived on the basis of the static force balance analysis by drawing a free-body diagram of the system.

Figure 3 presents the free-body diagram for the clutch disc assembly when it is engaged. The equation for the moment balance at the disc in the rotating direction is derived as

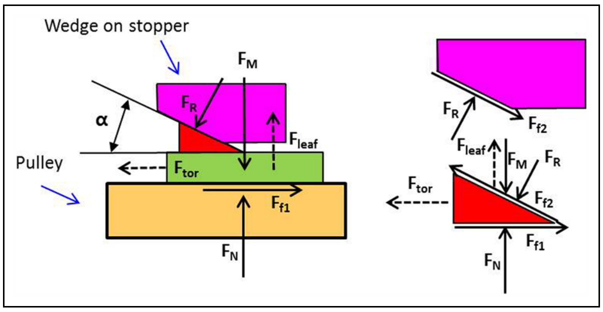

Figure 3. Free-body diagram of the wedge clutch system.

$$R_d F_{f1} - R_w F_R \sin\alpha - R_w F_{f2} \cos\alpha - R_{tor} F_{tor} = J_d \dot{\omega}_d \tag{1}$$

where α is the slope angle of the wedge blocks, F_{f1} is the clutch frictional force, F_R is the normal force on the wedge surface, F_{f2} is the frictional force on the wedge surface, F_{tor} is the torsion spring force, R_d is the effective radius of the friction lining on the disc, R_w is the effective radius of the wedge locations and R_{tor} is the effective radius of the torsion spring.

In order to derive the static amplification factor for analyses of the proposed system, the operating phase of the clutch is assumed to be in the steady state where the clutch is fully engaged, i.e. $\dot{\omega}_d = 0$, and the force amplification effect induced during engagement still remains. Thus, equation (1) is reduced to

$$R_d F_{f1} - R_w F_R \sin\alpha - R_w F_{f2} \cos\alpha - R_{tor} F_{tor} = 0 \tag{2}$$

When the force balance in a vertical direction is considered and if it is assumed that there is no displacement in that direction, then it is found that

$$F_N - F_R \cos\alpha - F_M + F_{f2} \sin\alpha + F_{leaf} = 0 \tag{3}$$

where F_M is the magnetic force, F_N is the normal force from the pulley and F_{leaf} is the leaf spring force.

When the dynamic friction coefficient of the pulley surface is denoted as μ_1, the compressor braking force can be expressed as

$$F_B = \mu_1 F_N \tag{4}$$

Similarly, the frictional force acting on the surface of the wedges is described as

$$F_f = \mu_2 F_R \tag{5}$$

where μ_2 is the friction coefficient of the wedge surface.

When equations (4) and (5) are substituted into equation (2), the equation can be arranged with respect to the normal force F_R on the wedge surface as

$$F_R = \frac{\mu_1 R_d F_N - R_{tor} F_{tor}}{R_w(\sin\alpha + \mu_2\cos\alpha)} \tag{6}$$

In the same manner, when equation (3) is combined with equation (5), the equation for the normal force F_N on the pulley surface can be obtained as

$$F_N = F_M - F_{leaf} + F_R(\cos\alpha - \mu_2\sin\alpha) \tag{7}$$

Then, by combining equations (6) and (7), the amplification factor Y_s of the engagement force due to the self-energizing effect is expressed as

$$\begin{aligned} Y_s &= \frac{F_N}{F_M} \\ &= G\left[1 - \frac{F_{tor}R_{tor}(1 - \mu_2\tan\alpha)}{F_M R_w(\tan\alpha + \mu_2)} - \frac{F_{leaf}}{F_M}\right] \end{aligned} \tag{8}$$

where the terms in the square brackets represent the factor reduction due to the torsion spring and leaf spring and where G is the original amplification factor determined using the characteristics of the clutch system and is given by

$$G = \frac{(R_w/R_B)(\tan\alpha + \mu_2)}{\tan\alpha\ (R_w/R_d) + \mu_2(R_w/R_d) - \mu_1 + \mu_1\mu_2\tan\alpha}$$

Amplification factor of the clutch torque capacity

The torque capacity, which is the maximum transmittable torque of the clutch and is one of the most representative quantities that demonstrates the performance of an energy-efficient clutch system, can be significantly increased using the self-energizing principle. If the static friction coefficient of the clutch is not changed during its operation, the torque amplification factor may be equal to the force amplification factor. In order to investigate the characteristics of static friction, experiments with the conventional clutch system were performed. The measured static friction coefficient values according to the applied clutch normal force are presented in Figure 4.

The static friction coefficient values were obtained on the basis of the measured torque capacity of the base clutch. When the normal force applied to the clutch was less than 500 N, the static friction coefficient was reduced significantly, which indicates that the corresponding transmittable torque was equally reduced. However, when the normal force was larger than approximately 1700 N, the friction coefficient was almost constant, which implies that the normal force was proportional to the clutch torque in that range. Based on these characteristics, it is inferred that the amplification factor of the normal force differs from the corresponding torque factor because the static friction coefficient can be changed when the normal force is amplified by the self-energizing mechanism. When this is considered, the amplification factor of the clutch torque capacity is described as

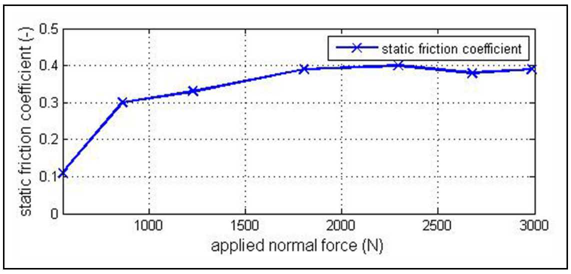

Figure 4. Static friction coefficient characteristics of the clutch.

$$\frac{T_{c,max\_w}}{T_{c,max\_o}} = Y_s\frac{\mu_{1s\_w}}{\mu_{1s\_o}} \tag{9}$$

where T_{c,max_o} is the torque capacity of the base clutch, T_{c,max_w} is the torque capacity of the wedge clutch, μ_{1s_o} is the static friction coefficient of the base clutch and μ_{1s_w} is the static friction coefficient of the wedge clutch.

Design considerations and analyses

Amplification factor of the engagement force

The amplification factor of the engagement force is the most important factor in the design of a wedge clutch system. The wedge slope angle α should be appropriately designed in order that an appropriate amplification factor is achieved by the wedge structure. With reference to equation (8), the amplification factor is affected by five factors: α, μ_1, μ_2, F_{leaf} and F_{tor}. Because there are too many variables, this increases the difficulty of the analysis; therefore, it is assumed that some variables have nominal values which can be determined empirically. The torsion spring, which is an optional component, was assumed to be not used in the system. For the clutch system without a torsion spring, equation (8) can be simplified to

$$\begin{aligned} Y_s &= \frac{F_N}{F_M} \\ &= G\left(1 - \frac{F_{leaf}}{F_M}\right) \end{aligned} \tag{10}$$

It is assumed that $R_w = R_d = 0.05$ m, $F_{leaf} = 2200$ N, $F_M = 2200$ N, $\mu_1 = 0.25$–0.45 and $\mu_2 = 0.1$–0.4. When the friction coefficient μ_1, which rarely changes once the clutch system is being engaged, is fixed to 0.25, 0.3, 0.35, 0.4 and 0.45; the variations in the

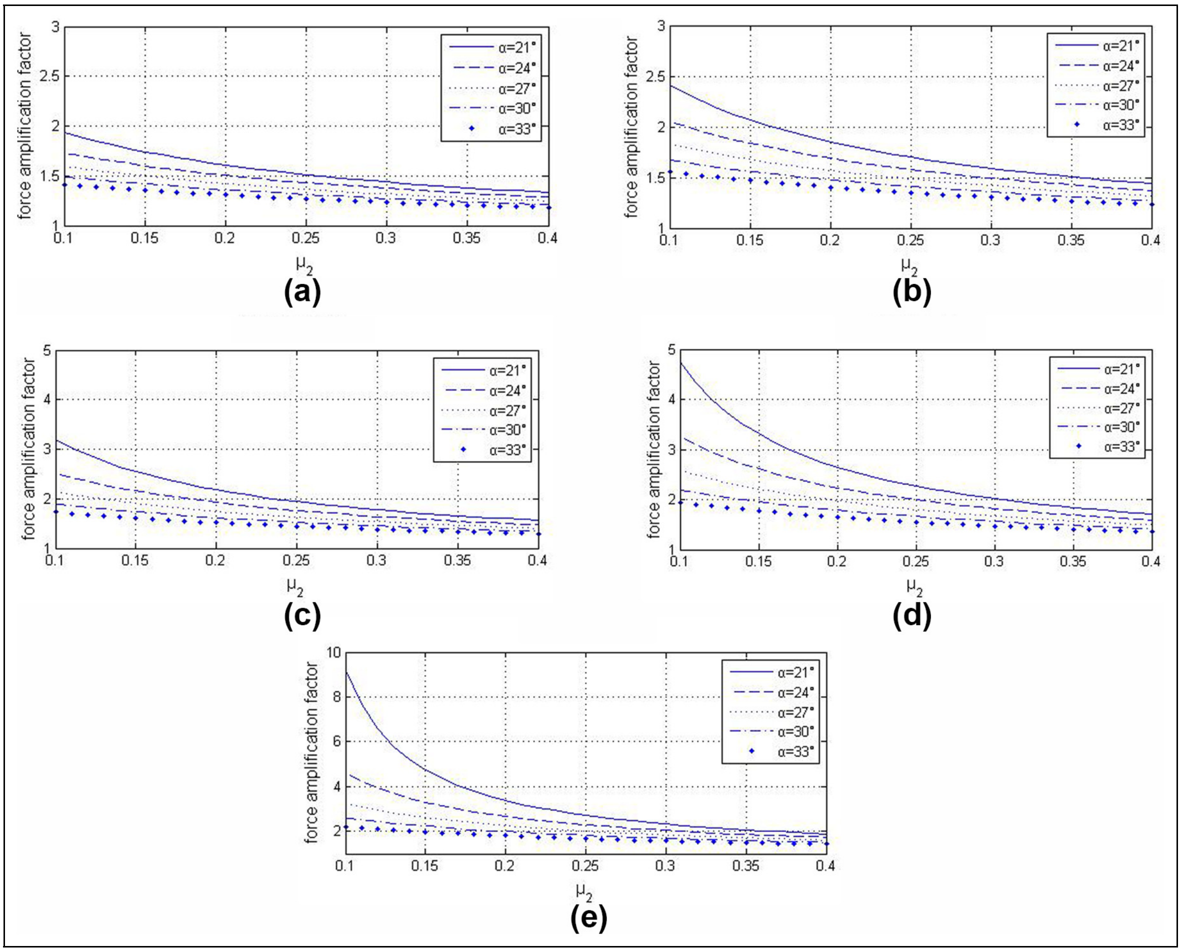

Figure 5. Variations in the force amplification factor according to μ_2 and α for (a) μ_1 = 0.25, (b) μ_1 = 0.3, (c) μ_1 = 0.35, (d) μ_1 = 0.4 and (e) μ_1 = 0.45.

amplification factor according to μ_2 and α can be determined as depicted in Figure 5.

As μ_1 increases, the amplification factor increases accordingly whereas, as μ_2 increases, the amplification factor decreases. As α becomes smaller, the corresponding amplification factor becomes not only larger but also very sensitive to the variation in μ_2. Therefore, the slope angle of the wedges should be designed so that stable operation of the clutch with an appropriate factor between 2 and 3 is guaranteed even with variation in μ_2. If α is larger than 30°, the self-energizing mechanism is marginally effective; thus, the amplification factor is less than 2 for general operating conditions.

Clutch locking

With reference to equation (8), it should be noted that the factor diverges when tan α = $[\mu_1-\mu_2(R_w/R_d)]/(\mu_1\mu_2 + R_w/R_d)$ or less. In this case, even if the magnetic force acting on the disc disappears after complete engagement, the disc does not return to its original position because it is locked by the wedges; this is referred to as the *locking phenomenon*. The slope angle α should be designed in order that the slope angle is larger than the critical angle in order to avoid the locking phenomenon and in order that the system always has a stable amplification factor of 2–3. The clutch locking phenomenon occurs when the amplification ratio caused by the self-energizing effect diverges or becomes negative. Hence, from equation (8), the locking avoidance condition is derived as

$$\tan\alpha > \frac{\mu_1 - \mu_2(R_w/R_d)}{\mu_1\mu_2 + R_w/R_d} \longleftrightarrow \alpha > \arctan\left[\frac{\mu_1 - \mu_2(R_w/R_d)}{\mu_1\mu_2 + R_w/R_d}\right] \tag{11}$$

When the nominal values of μ_1 and μ_2 are known empirically, the corresponding α to avoid clutch locking can be calculated. On the assumption that μ_1 is between 0.25 and 0.45 and μ_2 = 0.05–0.4, the minimum values of the slope angle α after which locking begins to occur can be calculated. The result is depicted in Figure 6.

Figure 6 presents the minimum slope angles of the wedge based on the variations in μ_1 and μ_2 for locking avoidance. For the extreme cases of μ_1 = 0.45 and μ_2 = 0.05, α should be larger than 21.37° in order to avoid clutch locking. Hence, α must be designed to be larger

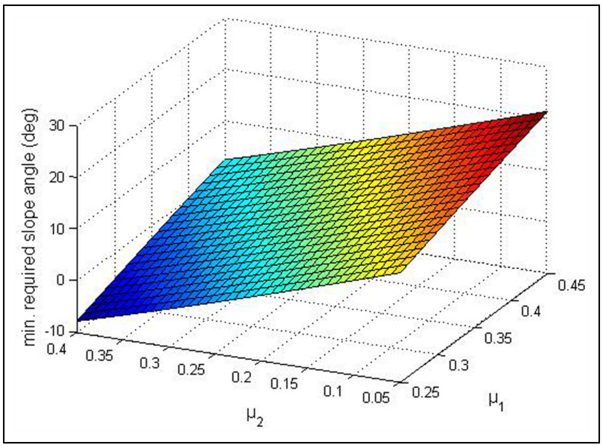

Figure 6. Minimum slope angle for locking avoidance.
min.: minimum.

than approximately 20° in order to avoid clutch locking in general operating conditions.

Dynamic-model-based analyses

In the previous section, the amplification factor of the engagement force was derived only for a static situation in which no slip occurs. However, it is also necessary to analyse the effects of the force amplification by the extra self-energizing mechanism in the system during the engagement process. The force amplification may cause the clutch to engage with too large a torque; this results in a poor impact on the surface of the clutch and, consequently, has a negative effect on its durability. In the next subsection, a dynamic model representing the engagement process of the clutch system was implemented in order to describe the behaviour of the wedge clutch system.

Dynamic modelling

When the torque balance relationships are considered, the dynamics of the clutch system are described as

$$J_p\dot{\omega}_p = T_p - T_c \tag{12}$$

$$J_{da}\dot{\omega}_d = T_c - T_L \tag{13}$$

where J_p is the inertia of the pulley, J_{da} is the inertia of the disc assembly that includes hubs, spring, and stopper structure, ω_p is the angular speed of the pulley, ω_d is the angular speed of the disc, T_p is the pulley torque delivered from the engine and T_c is the clutch torque.

When the current is flowing through the coil, the electromagnetic force F_M causes the disc to move in the direction of the force, and the displacement causes a restoring spring force F_{leaf} on the disc in the opposite direction. If δ_d is defined as the displacement of the disc and k_{leaf} is defined as the elastic modulus of the leaf spring, the leaf spring force is modelled as

$$F_{leaf} = k_{leaf}\delta_d \tag{14}$$

The net normal force acting on the disc is $F_M - F_{leaf}$. In general, the torque transmitted by the clutch is increased by the normal force during the engagement. These properties can be described as[16, 17]

$$\begin{aligned}\hat{T}_c &= \mu_1 F_N C_c \\ &= \mu_1 \left(F_M - F_{leaf}\right) C_c\end{aligned} \tag{15}$$

where C_c is the effective area of the clutch related to its geometry. Here, it is assumed that the dynamic friction coefficient of the clutch is slowly varying in order that it is constant during the gear shift. For a more detailed discussion on the dynamic friction coefficient characteristics, see the paper by Vasca et al.[16]

The compressor load torque T_L is determined by using equations (12) to (15), but it cannot be measured using sensors for production A/C systems. Furthermore, the compressor load varies because of many different factors; thus, it is difficult to estimate the load values accurately in real time. The detailed modelling process of the compressor torque is not in the scope of this study: see previous studies[18–20] for details of the modelling. When the actual value of the clutch torque is known, the compressor load during engagement can be calculated simply by using

$$\hat{T}_L = T_c - J_{da}\dot{\omega}_d \tag{16}$$

It may be argued that equation (16) is not practical because measuring the clutch torque in real time is not possible. However, equation (16) can be utilized for offline calculation of the compressor load when the clutch torque data are obtained from a test bench. This equation is useful for offline analysis of the clutch dynamic properties because of its simplicity.

Model validation

By combining equations (13), (14), (15) and (16), the rotational speed of the disc can be calculated. Comparison plots of the experimental results and simulations using the model are presented in Figure 7. The parametric values used in the clutch model are as follows: $\mu_1 = 0.32$, $k_{leaf} = 98.1$ N/mm, $C_c = 0.004485$ and $J_{da} = 0.000421$ kg m^2. In estimating the rotational speed of the disc, the model is set to maintain a constant speed once the disc is fully engaged, whereas the actual measured speed remains oscillatory. The results verify that the model can precisely describe the characteristics of the clutch torque and the disc speed during clutch engagement.

Model-based analyses of the wedge clutch system

The basic operating principles of the wedge clutch apparatus are the same as those of a conventional clutch. The only difference is that the wedges and some structural components such as stoppers are added to the wedge system; thus, the disc assembly inertia of the system differs slightly from that of a conventional

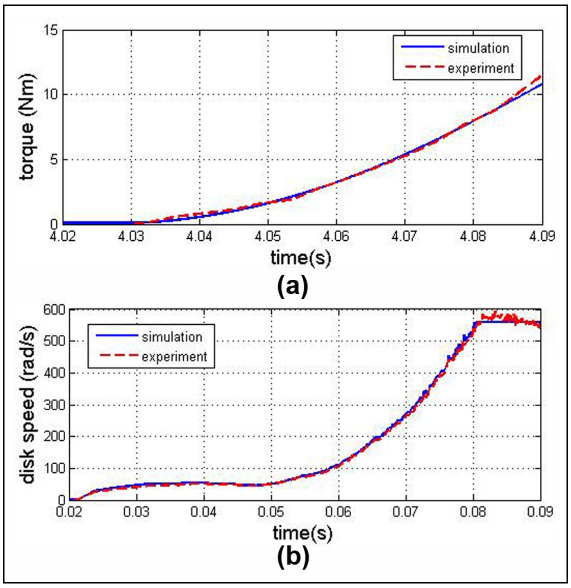

Figure 7. Model validation: (a) clutch torque; (b) disc speed.

system. In addition,because of the self-energizing effect generated by the wedges, the clutch normal force is amplified when the same current input is applied.

Most parameters that affect the amplification factor, including the electromagnetic force and the angular speed of the disc, vary while the clutch is engaged. The original moment balance in equation (1) which considers the inertia effect is used to derive the amplification factor of the dynamic force. Combining equation (1) with equations (4) and (5) gives

$$F_R = \frac{\mu_1 R_d F_N - R_{tor} F_{tor} - J_d \dot{\omega}_d}{R_w(\sin\alpha + \mu_2 \cos\alpha)} \tag{17}$$

It should be noted that the variation in the displacement of the disc in the axial direction does not need to be considered because the displacement caused by the magnetic force is constant after the disc makes contact with the pulley. Hence, on substitution of equation (17) into equation (7), the dynamic amplification factor when $F_{tor} = 0$ is derived as

$$\begin{aligned} Y_d &= \frac{F_N}{F_M} \\ &= G\left(1 - I - \frac{F_{leaf}}{F_M}\right) \end{aligned} \tag{18}$$

where

$$I = \frac{J_d \dot{\omega}_d (1 - \mu_2 \tan\alpha)}{F_M R_w (\tan\alpha + \mu_2)}$$

Equation (18) gives the dynamic amplification factor with a reduced value owing to the inertia effect. In order to examine the effects of the self-energizing mechanism on engagement of the clutch, simulations with the developed model were performed.

Figure 8 presents the model responses of the three wedge clutch actuators with different angles when the current input in Figure 8(a) is applied to each system under the assumption that μ_1 and μ_2 are constant during engagement. The measured speed data of the disc used to calculate the compressor load were processed with a low-pass filter in order to remove the noise. With the self-energizing mechanism, the torque was amplified, which resulted in faster engagement of the clutch compared with that of the conventional clutch (Figure 8(c) and (d)). Thus, the time for complete engagement (the lock-up time) decreased together with an increase in the amplification factor. In order to investigate the effect of the torque amplification on the clutch durability, the frictional energy E_d dissipated in the clutch

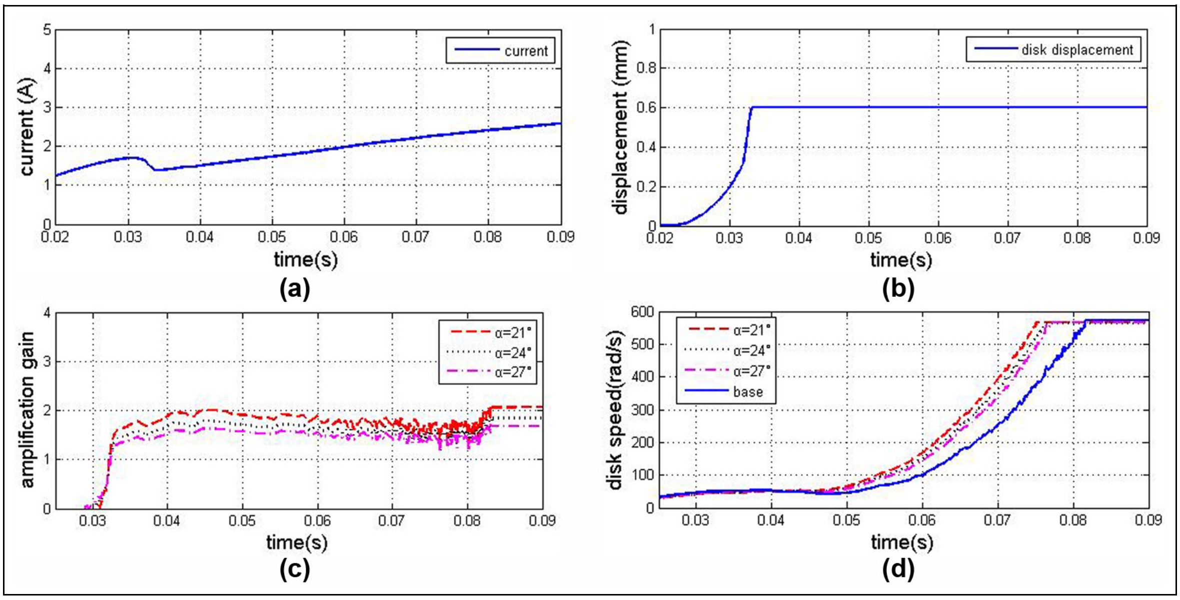

Figure 8. Model responses during engagement (F_{tor} = 0, μ_2 = 0.18 and μ_1 = 0.32): (a) current; (b) disc displacement; (c) dynamic force amplification factor; (d) disc speed.

Figure 9. System prototypes and measurement devices.

Table 1. Comparison of the clutch dissipated energies with respect to the wedge angles.

α (deg)	t_l (s)	$T_c(t_l)$ (N m)	E_d (J)
21	0.0751	10.3	54.4
24	0.0762	9.8	52.9
27	0.0771	9.6	50.9
Base	0.0816	8.1	44.6

Table 2. Measurements for the base clutch.

Current (A)	Electromagnetic force (N)	Torque capacity (N m)
1	427	0.27
1.5	1103	1.8
2	1676	3.18

during the slipping phase should be considered; it is given by

$$E_d = \int_0^{t_l} T_c \omega_{dp} \, dt \tag{19}$$

where $\omega_{dp} = \omega_p - \omega_d$ is the slip speed of the pulley and disc and t_l is the lock-up time of the clutch.

The lock-up time t_l, the transmitted torque $T_c(t_l)$ at lock-up and the corresponding clutch dissipated energy for each clutch including the conventional clutch are presented in Table 1.

Although the lock-up time became shorter (i.e. a shorter slipping phase) with a larger amplification factor, more frictional energy was dissipated because the transmitted clutch torque was significantly larger than the base torque. The results demonstrate that an excessively large amplification factor due to the self-energizing effect can cause ill effects in the clutch durability. However, the transmitted torque was not sufficiently high and the lock-up time of the clutch was not sufficiently long that the dissipated energy from them was insignificant. As described in Table 1, when the wedge angle was the smallest, i.e. $\alpha = 21°$ (the value for which the self-energizing effect was maximized for the three cases), the increase in the dissipated energy was approximately 10.2 J. Therefore, it is claimed that it had a negligible effect on the frictional losses of the clutch, at least when the wedge angle was in that range. Thus, it can be concluded from the model-based analyses that the torque amplification does not have a negative impact on the clutch durability during engagement.

Experimental analysis

Experimental set-up

The base clutch used for the experiments was a dry friction clutch installed in a production A/C compressor from the Halla Climate Control Company. The diameter of the base clutch was 119 mm. The slope angles of the prototypes were 21°, 24° and 27°, in accordance with the analytical studies in the previous sections. The field coil and pulley assemblies of the system were the same as those of conventional production systems. The torque transmitted by the clutch was measured using a torque meter.

In order to verify the capability of the torque amplification of the proposed system, a comparative study of the proposed wedge clutch system was undertaken with a conventional system. The torque capacities of both systems were used as the measure because they determine which clutch can deliver more torque when the same current is applied.

Experimental results

The magnetic force and the torque capacity of the base clutch in the steady state were measured. The results corresponding to currents of 1 A, 1.5 A and 2 A are presented in Table 2. Next, the torque capacities of the

Table 3. Experimental results: comparison of the clutch torque capacities.

Current (A)	Clutch torque capacity (N m)			
	Base clutch	Clutch A (α = 21°)	Clutch B (α = 24°)	Clutch C (α = 27°)
1	0.27	2.32	0.97	1.3
1.5	1.8	5.22	4.76	3.93
2	3.18	6	5.57	5.26
Locking?	No	No	No	No

Table 4. Estimated values of μ_{1s_w}.

Current (A)	μ_{1s_o}	μ_2	Estimated μ_{1s_w}		
			α = 21°	α = 24°	α = 27°
1	0.11	0.18	0.39	0.33	0.29
1.5	0.33	0.18	0.39	0.39	0.39
2	0.39	0.18	0.39	0.39	0.39

three wedge clutches were measured using a torque meter when the same current inputs were applied to the systems. In order to improve the measurement accuracy, every measurment procedure was repeated five times, and the final measured value was the average of the five values for each clutch. The results are presented in Table 3.

As the slope angle of the wedges was reduced, the torque amplification factor due to the self-energizing effect increased as expected, which agreed well with the analysis results in the previous section. Furthermore, in comparison with that of the base clutch, the torque capacity increased significantly for all cases with different wedge angles and different applied currents. Consequently, the amplification of the torque can lead to a reduction in the electric power consumption and potentially to a magnetic coil design with a reduced weight and a reduced cost. Figure 10 compares the torque amplification factors for different wedge clutches and the conventional base clutch.

It is critical to guarantee that the self-energizing wedged clutch returns to the disengaged position when the current is switched off. As predicted by the analysis in the previous section, the discs of all wedge clutches returned to their original positions safely when the current input was disconnected. It should be noted that the torque amplification factor increased as the applied current decreased because, when the current is sufficiently small, μ_{1s_o} in equation (9) is so small that the torque factor becomes significantly larger than the corresponding force factor. Moreover, a large variation in the torque amplification factor during the operation of the clutch system adversely affects it and can cause unstable operation. According to Figure 10, the torque amplification factor is highly sensitive to the operating conditions for $\alpha = 21°$ and $\alpha = 24°$. In contrast, the wedge clutch with $\alpha = 27°$ exhibited torque amplification with an appropriate factor of 2–3 for all cases. Hence, it is concluded that a slope angle of near 27° is appropriate for the wedge design of the target A/C friction clutch. The experimental analysis also validated the effectiveness and the feasibility of the proposed clutch system to control the clutch engagement in an energy-efficient way.

Validation of amplification factor model

Because equation (9) was used to analyse the system design in the previous section, its accuracy of modelling the torque amplification factor of a wedge clutch must be validated. In order to use equation (9), all friction coefficient values including μ_{1s_o}, μ_{1s_w} and μ_2 should be known. However, the static friction coefficient μ_{1s_w} of the pulley surface in a wedge clutch cannot be directly measured because the corresponding amplified normal force is unknown. However, it can be estimated on the basis of the static friction characteristic curve in Figure 4. For example, when the current applied to a wedge clutch is larger than 1.5 A, the corresponding value of μ_{1s_w} is assumed to be 0.39 because the amplified normal force is larger than 1700 N. On the assumptions that the nominal value of μ_2 is 0.18 and that it does not change during the experiments, the estimated values of μ_{1s_w} are presented in Table 4. The friction coefficients are assumed to be 0.39, except for two extreme cases when the current is 1 A. Using the known μ_2 and the analysis results in the third section, μ_{1s_w} is determined to be 0.29 for $\alpha = 27°$ and 0.31 for $\alpha = 24°$. Using the parametric values in Table 4, the corresponding static torque factor can be calculated using equation (9), as depicted in Figure 11.

Figure 11 compares the calculated torque factors with the experimentally measured values, which validate the fact that the torque factor modelled in equation (9) is accurate. Even though the estimated values of μ_{1s_w} were used in the model, equation (9) predicted the torque amplification factors well for all cases. The friction coefficient μ_2 of the wedges can be varied in

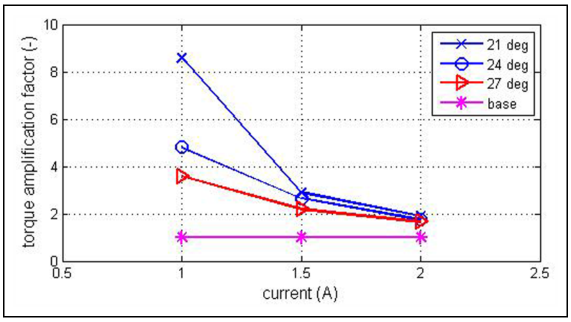

Figure 10. Comparison of the torque amplification factors.

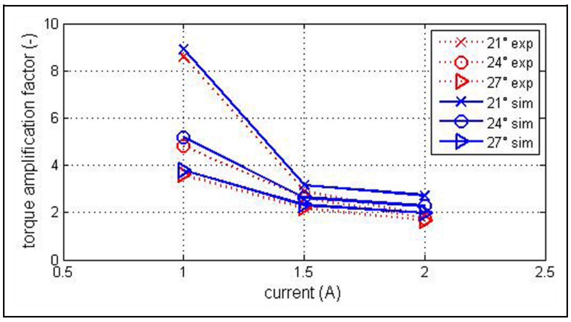

Figure 11. Validation of the torque amplification factor model. exp: experimental data; sim: simulated data.

each experiment; thus, the assumption that μ_2 is a constant may be a primary cause of the model errors, particularly when the current is 2 A. However, the results prove that the self-energizing effect of the proposed system is well described by equations (8) and (9), and the equations can be used for further analysis and design of such systems.

Conclusion

This paper proposed a new clutch actuator design using the self-energizing effect to reduce the power consumption of the actuation. The proposed clutch actuator is based on a dry clutch for vehicle A/C systems; however, the design methodology can be easily applied to other friction clutches for vehicle applications. The influences of the pulley surface friction coefficient, the wedge surface friction coefficient and the leaf spring force on the amplification factor of the engagement force were analysed in order to determine the appropriate slope angle of the wedges. From the model analysis and the experimental results, it was proposed that the slope angle of the wedges for the target clutch should be near 27° in order to obtain the appropriate amplification factor while avoiding self-locking. Dynamic modelling and analyses based on the developed model were also performed in order to investigate the transient characteristics of the modified structure. The experimental results on a test bench with three different prototypes indicated that the wedge clutch actuator system was significantly more efficient than the conventional clutch because it can significantly amplify the engagement force, thus reducing the actuation energy and potentially lowering the weight and decreasing the cost of the actuation system.

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Appendix 1

Notation

C_c	clutch geometry constant
F_{f1}	frictional force on the disc
F_{f2}	frictional force on the wedge surface
F_{leaf}	leaf spring force
F_M	electromagnetic force
F_N	normal force on the disc
F_R	reaction force on the wedge surface
F_{tor}	torsion spring force
J_d	inertia of the disc itself
J_{da}	inertia of the disc assembly
J_p	inertia of the pulley
k_{leaf}	elastic modulus of leaf spring
R_d	effective radius of the friction lining on the disc
R_{tor}	effective radius of the torsion spring
R_w	effective radius of the wedge location
T_c	clutch torque
T_L	compressor load
T_p	pulley torque
α	slope angle of the wedges
δ_d	displacement of the disc
μ_1	dynamic friction coefficient of the disc
μ_{1s}	static friction coefficient of the disc
μ_2	dynamic friction coefficient of the wedge surface
ω_d	angular velocity of the disc
ω_p	angular velocity of the pulley
Y_d	dynamic amplification factor of the force
Y_s	static amplification factor of the force