DESIGN AND ANALYSIS OF AUTOMOTIVE CLUTCH SYSTEMS

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ABSTRACT

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This study is an attempt to include the engagement characteristics of clutches in vehicle performance studies. A computer simulation of engagement dynamics is used in order to study the clutch engagement. The simulation includes engine torque characteristics, diaphragm spring characteristics, driveline components, rotational inertias, vehicle parameters and road conditions. The engine throttle level and clutch pedal release scenarios are also included in the analysis. The vehicle performance in terms of vehicle acceleration, velocity and distance taken during the clutch engagement are analysed.

A mathematical model is used to determine sliding velocity between clutch side and engine side, torque transmitted through the clutch, engagement time, surface temperatures reached and energy dissipation of the clutch during slipping period. Numerical examples and detailed results of the clutch engagement are presented and discussed. The case studies have shown that the driver behaviour is the most effective parameter in the engagements. Low rpm engagements and thus low slipping periods are more advantageous when the clutch temperature and clutch life is considered. Gradient angle of the road has also a great effect on engagement characteristics and it is the prior parameter when selecting a clutch. Different clutch discs are analysed for the same cases and results are compared. Larger clutch discs are found to have shorter slipping times and reach lower temperatures for the same conditions.

Keywords: Clutches, Diaphragm Spring Clutches, Clutch Engagement, Friction, Friction Discs, Friction Clutches, Disc Clutches.

ÖZ

OTOMOTİV KAVRAMA SİSTEMLERİ TASARIM VE ANALİZİ

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Bu çalışmada araçların kavrama karakteristiklerin araç performansı üzerine etkisi incelenmiştir. Kavrama dinamiklerini esas alan bir bilgisayar programı hazırlanmıştır. Analizde motor tork karakteristik eğrileri, diyafram yay kuvvet karakteristikleri, güç aktarma elemanları, dönel ataletler ve yol koşulları gözönüne alınmıştır. Sürücünün gaz ve debriyaj pedalına basma senaryoları da analize eklenmiştir. Kavrama sırasındaki araç performansı ivme, hız ve katedilen yol olarak detaylı şekilde incelenmiştir.

Kurulan matematiksel model, sürtünme hızı, aktarılan tork, kavrama süresi, ulaşılan sıcaklıklar, ortaya çıkan ısı gibi değişkenleri değişik durumlarda incelemekte kullanılmıştır. Nümerik örnekler üzerinde detaylı kavrama analizleri yapılmış, sonuçlar üzerinde tartışılmıştır. Örnek analizlerde, kavramada en etkili parametrenin sürücü davranışları olduğu belirlenmiştir. Düşük motor devirlerinde tamamlanan kavramaların sıcaklıklar ve balata ömrü açısından avantajlı olduğu gözlemlenmiştir. Yolun eğim açısının da kavramada çok etkili bir parametre olduğu ortaya çıkmıştır. Farklı boyutlardaki debriyaj balataları ile yapılan analizler sonucu, büyük balataların daha kısa kavrama süresinde ve daha düşük sıcaklıklarda kavramayı tamamladıkları görülmüştür.

Anahtar Kelimeler: Kavrama, Debriyaj, Diyafram Yaylı Kavramalar, Sürtünme, Balata, Sürtünmeli Kavramalar, Debriyaj Balataları.

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a: Vehicle acceleration

aa: Tire rolling resistance coefficient

A: Friction Area

$A^{/}$: Area Exposed to atmosphere

$A_{f:}$ Area of the front view of the vehicle

BL: Bearing Load

c_1, c_2: Specific heat of metal and clutch facing respectively

C_D : Aerodynamic drag coefficient

$CR_{Max:}$ The maximum clutch release specified

d_1: Pressure plate thickness

d_2: Friction Disc thickness

d_3: Flywheel thickness

De: Outer diameter of diaphragm spring

Di: Inner diameter of diaphragm spring

Dt: Inner diameter of closed ring section of the diaphragm spring

E: Modulus of elasticity

Frr: Rolling Resistance

H: Free height of the diaphragm spring

I_1: Moment of Inertia of Engine

I_2: Moment of Inertia of transmission parts and vehicle mass

Ig: Moment of Inertia of transmission parts rotating at engine speed

Ip: Moment of Inertia of transmission parts rotating at shaft speed

Iw: Moment of Inertia of transmission parts rotating at wheel speed

k1, k2: Thermal conductivity of metal and clutch facing respectively

L: Length of air flow path

M: Total vehicle mass

N: Normal Force on the clutch plate

ρ_1, ρ_2: Density of metal and clutch facing respectively

p: Pressure on the clutch plate

Plimit: Limiting Max. pressure that the friction material can stand.

Pmax: Max Pressure on clutch plate

Qt: Total Energy Dissipated

R: Reynolds number

Ri: Clutch Friction Disc Inner radius

Ro: Clutch Friction Disc Outer radius

PPW: Pressure Plate Weight

r_f: Final gear ratio (Differantial ratio)

rg: Gearbox ratio

r_w: Wheel radius

S_1: Deflection of the closed ring section of the diaphragm spring

S_2: Deflection of the tongues of diaphragm spring

t: time

ts: total slipping time

T: Thickness of the diaphragm spring

T_a: Torque on clutch due to aerodynamic effects

T_C : Clutch Torque Capacity

T_g: Torque on Clutch Due to Gradient resistance

Tmax: Limiting temperature of clutch friction material

T_{rr}: Torque on clutch by tire rolling resistance

T_1: Engine Torque

T_2: Total resistance torque

Tambient: Ambient temperature

Temp: Clutch temperature

$V_{vehicle}$: Vehicle speed

W: Weight of the pressure plate

ω_1: Engine rotational speed

ω_2: Clutch rotational speed on the driven side

ΔTemp: Temperature increase of clutch surface

μ: Friction coefficient

μ_{static}: Static friction coefficient

μ_{High}: Dynamic friction coefficient

η: overall efficiency of the driveline components.

θ : Grade angle

υ: Poisson's ratio

CHAPTER 1

INTRODUCTION

Friction clutches are torque transmitting devices designed to be engaged and disengaged while driving or driven members rotate. Clutch is the essential component between the driven and driving plates that synchronises the rotational speeds.

The introduction of the internal combustion engine has had a substantial influence on the development of clutches. The automotive requirements has been of primary concern. The task of clutch is connecting the driven machinery to the prime mover, while maintaining the latter at a minimum operating speed with good torque response, and sufficient energy absorption capacity.

The driving and driven plates are the primary torque transmitting members of the clutch. Their two main functions are to generate the frictional torque transmitting capacity at the interface and to transmit the applied torque. During a clutch engagement, the torque applied to the plates is equal to the frictional torque capacity of the clutch until clutch slip has ceased; after the clutch is locked up, clutch torque is the torque applied by the prime mover or sometimes by the driven machinery.

Materials for clutch plates are under continual development. The major emphasis has been aimed at obtaining high friction coefficients, consistent friction coefficients, low wear rate for both the friction material and the matting surface, resistance to seizing, low swell, high temperature resistance, high energy absorption capacity, resilience, quiet and chatter-free engagement, and lately to eliminate asbestos. The plate interface is usually made up of a friction material and a ferrous opposing surface.

The clutch engagement analysis must be made from the clutch pedal to the engine. Every parameter between tires and the engine should be considered to make a realistic analysis. The large number of variables and the complexity of the analysis make the need of a computer aided design procedure. Using a small time interval solution procedure with numerical methods can simulate the engagement action. Iterative approaches should be used to converge the needed parameters.

A large number of clutch configurations result from the many methods of applying the clamp force to the plates. The commonly utilised force generating principles are: Manual, coil spring, diaphragm spring, air pressure, torque assist and combinations of these.

Before designing a clutch system, the input parameters should be well defined. These input variables are the engine torque-speed characteristics, vehicle specifications, design conditions and the operator variables. The operator controls both the clamping force (by clutch pedal) and the input torque (by the throttle pedal) in a vehicle. In order to make the analysis, these operator inputs must be considered in detail.

The engine and vehicle specifications must also be considered in detail to make a successful simulation. The vehicle inputs should contain gear ratios, masses, moment of inertia of the rotating parts, tire resistances, aerodynamic drag coefficient and so on. The design conditions should be taken as the worst conditions under which the engagement occurs. The number of engagements, the gradient and the ambient temperature are some of these conditions.

1.1 Literature Survey on Clutches

A large number of studies have been performed on clutch design. Different types of clutches (friction, wet friction, cone, jaw, gear, expanding rim, etc.) are used for engagement processes. The selection of the clutch type depends on its usage. For a smooth engagement, friction type clutches will be appropriate. If large torques are to be transmitted, wet clutches will be the right choice although they have more complex structures.

Clutch life is usually limited by the facing wear and it is highly depended upon the quality of the engagement process. Engagement time, torque to be transmitted and the variation of the torque transmission with time are the factors that designate the engagement quality. Engaging the clutch from a stand still requires the coordination and modulation of engine torque with the throttle pedal, torque transmitted through the clutch with the clutch pedal and vehicle speed. A large number of clutch facing and clutch control parameters have profound effects on the engagement characteristics, clutch performance and the overall performance of the vehicle. A poor engagement may be responsible for both vehicle performance problems and clutch failures.

1.1.1 Works on Engagement Characteristics and Energy Calculations

Szadkowski and Morfold [1] worked on clutch engagement and ideal engagement characteristics are defined. No lurge and no kill conditions are discussed and the objectives to perform an ideal engagement are listed. Using a mathematical modelling of the system, a relation between clutch torque and engine torque is expressed. Engagement without throttle is studied for typical examples of commercial trucks by using a computer simulation. Also the plate load, facing material properties, effect of number of plates, clutch size and clutch pressure is studied in this paper. A simulation of first gear start with throttle is made and the results are compared with experimental values. An important observation was made as a result of this study; fast aggressive starts do not produce greater temperature problems for dry friction materials. On the contrary thermal problems occur when the vehicle experiences a series of slow clutch engagements.

The work of Szadkowski and McNerney [2] is a continuation of engineering efforts on mathematical expressions and computer simulation presented in [1]. The modelling and study is extended on starting a vehicle with use of the throttle.

The basic mathematical models are modified in this paper because clutch engagement with throttle makes the researchers consider the human factors. Not only the clutch release but also the accelerator pedal is controlled by vehicle operator. The definition of ideal engagement is modified and throttle lead time and throttle level inputs are added in the mathematical model. Moreover, the model has been adjusted to predict low range characteristics of diesel and gas engines. Discussion of various simulation cases and results of extensive studies for various clutch parameters (such as the plate load, facing materials, pressure, damper size, damper stiffness, number of plates, engine parameters) are presented. Important extreme cases (repeated clutching, low loading rate etc.) are examined to determine when the clutch lock-up cannot be accomplished or the engine cannot be stalled. Important conclusions of this paper are: The vehicle operator should be able to keep the engagement speed under a certain level, which relates directly to the clutch plate-loading rate. It is a way to ensure ideal or close to optimum engagements. Larger starting throttles will result in longer engagement times and less favourable engagement characteristics. Also the plate loading rate corresponding to the optimum engagement decreases with throttle level increase. A quality engagement in a truck with a gasoline engine is more operator dependent than in diesel trucks. Not only too large but also too small throttles make smooth engagements difficult for the operator to perform and negatively affect the wear of the clutch. The throttle lead-time allows simplification of human factors in this mathematical model. The lead-time may be considered as a secondary factor in the design of engagement systems.

Michael [3] reviewed the use of disc type wet brakes and clutches in off-highway machines and pointing out some key design elements like torque capacity, energy capacity, cooling capacity and noise of brake and clutch design. Importance is given to torque and energy capacity design techniques in this study. Also the design data of different friction materials, calculations of clutch torque and absorbed energy by the clutch are given. Uniform facing pressure and uniform coefficient of friction assumption is made to calculate the clutch torque, clamping force and energy capacity. The formulation is nearly same with Szadkowski's work [1] but average diameter is used instead of using outer and inner diameters.

Disc type clutches are important components of many vehicle drive trains. Wet clutches are becoming widely used for heavy-duty applications where durability is important [3]. Traditionally clutches are designed to provide sufficient torque capacity using simple and well known techniques, but all other variables need to be considered. While the design analysis of clutches for energy capacity and chatter can be extremely complex. For most applications there are some relatively simple relationships that can be applied to obtain an adequate design.

Hermanns [4] worked on design refinements, which will improve the clutch life and operating characteristics. This study deals with clutch sizing, two plate clutches, heat effect, temperature increase, startability of the vehicles, characteristics of the friction materials, engagement characteristics and torque capacities. A large part of the paper is on the analysis of two plate clutches from the viewpoint of torque capacity and energy absorption. Benefits of the two plate disc clutches are discussed. Vehicle test data is given in this paper which indicates the amount of energy converted to heat by the clutch during a normal vehicle start. Start ability of vehicles that has a great effect on clutch life is investigated. Operator factor is found to be the most important parameter in determining clutch life.

Ünlüsoy and Akkök [5] presented a dynamic model of clutch engagement. Both the engine and the transmission dynamics are simulated with this model. A number of scenarios for the simulation of gas and clutch pedals are used. The motion of vehicle is determined by solving the model with these scenarios numerically. The engagement analysis is made both on the level road and uphill. The input and output speeds of clutch, vehicle speed and distance covered by the vehicle are calculated. With this model, engagement characteristics of a specified vehicle can be determined which can be very useful for the designers.

Willyard's [6] considered the ergonomic and operational advantages of single plate diaphragm spring type clutches over the twin plate coil spring type clutches. Comparisons between these two different clutch types are made. The functional differences between these two types are described and comparison of single plate clutches with twin plate clutches is given. In this work clutch pedal effort, torque capacity, shift effort comparisons, clutch wear life and clutch reactive torque comparisons are made between single plate diaphragm spring type clutches and twin plate coil spring type clutches. This work also includes the experimental studies of friction materials including temperature effects. These experimental studies include a long-term wear test that is performed with different vehicles. The typical diaphragm spring clamped load characteristic of a clutch is investigated in this study also. In all of these comparisons and tests, it is shown that the single plate diaphragm spring clutches are more convenient for the vehicles.

Hong [7] developed a computer-based simulation of a vehicle to design the power-train system of passenger cars aiming to operate at optimal performance. In this study, dynamic simulation of road vehicle under transient accelerating conditions has been accomplished through the combination of a dynamic engine model, a dynamic power-train model and a dynamic road load model. This dynamic simulation is capable of prediction of the transient performance of road vehicle power-train systems with a good accuracy and without inputting experimental engine performance maps. Hence, it is possible to parametrically determine the real engine characteristics and the engine-transmission matching with this model.

Burgan [8] developed a formula to calculate the instantaneous energy during engagement when coefficient of friction, applied pressure, slip speed are estimated. This paper claims the idea of controlling the energy capacity to prevent the clutch failure and to obtain a higher clutch life. In high-energy clutch applications, the designer must consider both the total energy absorbed by the clutch and the instantaneous rate at which energy is dissipated. The main conclusion is: if the designer can estimate the energy capacity, he also can control the energy capacity and thus the clutch life.

Schremmer [9] developed the force-deflection equations of slotted conical disc springs. The load-deflection diagrams are given and they are compared with the measured results taken by the testing machine. The deflection of the tongues and stress analysis of the spring is also given in this paper.

1.1.2 Works on Friction Materials

Wet clutch facings are mainly used in automatic transmissions of automobiles as annular multi disc clutches. The wet disc clutches used in automatic transmissions consist mainly of multiple separators and friction plates. During the engagement of clutches a large amount of friction heat is generated at the separator friction plate interface depending on the friction material, pressure applied and time.

In the friction clutches there is a considerable uncertainty about the distribution of axial load P over the contact surfaces. Between surfaces made of hard frictional materials such as sintered metals, contact is far from uniform and the pressure distribution is highly irregular. With soft friction surfaces of low elastic moduli such as certain asbestos – resin materials, fairly uniform contact can be assumed and hence the pressure is independent of the radius. Moreover when the clutch life is considered, the rate at which surface wear depends not only on the pressure but also the velocity of sliding between the surfaces.

Lucas and Mizon [10] defined the equations of motion of a two-degree of freedom system for clutch engagement. A numerical method was employed to solve the equations with a digital computer. Clutch torque and engine torque are taken as functions of time. Also practical tests are performed with different drivers and the results are compared with the analytical results. Different gradients are taken into consideration in these tests and a statistical table of driver behaviours is formed.

These statistical works have shown that generally driver applies a sufficient clamp load to hold the vehicle and this is held constant throughout the subsequent engagement until the slip speed is nearly zero. Then the rest of the clamp load is applied rapidly. The length of time in which the vehicle is held on the clutch can be quite long, of the order of 0.5 s on the level and more than 1 s on the gradient. Another conclusion of this paper is that a quick take-off obtained by using of higher initial engine speed and a higher clamping force generates more heat and temperatures. Less heat is generated if the driver allows the engine speed to drop during clutch engagement.

Lucas and Mizon [11] used the same mathematical model but a different friction model that includes the temperature, sliding speed and pressure effects. This model used provides information on the instantaneous torque output from the clutch, instantaneous speed and the instantaneous spatial temperature distributions through the clutch components during engagement period.

Gibson and Taccini [12] introduced a new type of friction material for the brake and clutch applications. This material is composed of carbon fibre reinforced carbon matrix composite, which has been subjected to high processing temperatures. The properties of this material are high coefficient of friction, low wear, low density, excellent stability at high temperatures and consistency of performance when using the same material against itself. This paper discusses the results obtained from dry friction applications.

Rhee [13] worked on three different friction materials to derive a friction force formulation, which involves the temperature, sliding speed and pressure effects. This paper also reviews past studies about friction force in the literature and searched the reliability of these equations. A new equation is given according to the test results including some constants. These constants are given as intervals to be appropriate for different type of friction materials.

1.1.3 Works on the Effect of Temperature and Contact Pressure

Many authors have stated that excessive temperatures and heat rates are major causes of clutch failures. Simulation of these temperatures is preferred over trial and error methods. The purpose of all simulation models is to describe the reality as closely as it is needed for the purposes without getting lost in complex details. Another approach to the problem of clutch temperature is to first calculate the generated heat under some circumstances as a function of time and then use this or a simplified heat characteristic as boundary condition for the Fourier equation to obtain the values of the temperatures.

Some of the most important conclusions are that both the dynamic and static coefficients of friction slightly decrease and the engagement time increases with increasing temperature. It was also found that the static and dynamic friction coefficients are not influenced by energy if the instant temperature is the same. Another important parameter is the stick-slip motion of the clutch, which depends on the difference of static and dynamic coefficient of frictions. Increasing the temperature will decrease the difference between the static and dynamic coefficient of friction and thus increase the risk of stick-slip.

Thermal loading in clutches is affected by normal pressure distributions on friction surfaces. When the pressure distributions are non-uniform, the distributions of heat flux generated on friction surfaces are also non-uniform. This can result in high local temperatures and high thermal stresses. The distribution of pressure depends on both design and material factors. They can also be subjected to changes during the clutch engagement due to thermal deformations [14].

During clutch engagements, the friction surface temperature reached becomes excessive and clutch failure may occur. To prevent the clutch failure, it is necessary to know the value of maximum surface temperature, how this depends on known conditions of loading, physical properties and dimensions of clutch plates and the degree of air-cooling. With repeated clutch engagements the temperature depends mainly on the coefficient of heat transfer from the clutch to the atmosphere and the area of the rubbing path. If the heat transfer coefficient is known or can be estimated, then by taking a limiting temperature, the area of clutch surface needed can be determined.

Zagrodzki [14] developed a theoretical model of a multiple disc clutch. With this model, distribution of initial pressure and temperatures are examined. Using the pressure distributions the temperatures and the stresses in the cross-sections of friction discs are calculated. The most important factor in thermal loading is the mean heat flux generated on friction discs during engagement. Experimentally determined values of heat flux are given for a specified material in this paper. The study also indicates that under non-uniform pressure distributions, stresses in discs can reach values several times higher than those reached under uniform distributions.

Wouters [15] presented a new approach; all variables mechanical and thermal must be calculated in the same loop. A computer program was developed to solve the heat transfer equations. In the study also the torque and energy equations are derived to calculate the generated heat rate. The graphical results of this computer program includes generated heat in clutch versus heat rate and temperature changes versus time.

Holgerson and Lundberg [16] investigated the influence of temperature and energy on various engagement conditions. A test setup, which operates with both drive torque and inertia was prepared to measure the characteristics of clutch engagement. Tests were performed with two different energy/friction area and with different starting speeds. Starting temperature is the clutch face temperature just before the engagement and it is adjusted by hot air. The results show that energy levels and sliding speeds do not affect the static and dynamic coefficient of frictions at the same instant temperatures. The difference between the static and dynamic coefficient of friction slightly decrease with increase in temperature and the engagement time increases while the torque capacity decreases with the increasing temperature

Newcomb [17] considered the torque capacity, rate of energy dissipation and temperatures reached in a single and multiple engagements. Expressions are given to determine the torque capacity of a clutch and rate at which heat is dissipated by dry friction during a single clutch engagement. In the study it is shown how the maximum friction surface temperatures varies with thickness of the pressure plate for design purposes. A rapid method of calculating temperatures when the slipping time is known is given. A good agreement between the theoretically and experimentally determined values prove the equations reliability. A formula is given to enable the area of the rubbing path to be calculated if the temperature is not to exceed a specific value.

Newcomb's other work [18] mostly dealt with drum brakes. The heat generated during braking, the transient temperatures during a single brake application are considered. A general expression for the temperature rise is given and its application to the determination of the area of friction path for a given vehicle is made. Newcomb [19] calculated the effect of the dimensions and thermal properties of the brake components on the rate of temperature rise and on the limiting temperatures reached during a series of applications.

1.2 Object of the Study

The aim of this study is to develop an efficient, user friendly and modular computer program to take into account all the necessary parameters of the vehicle system, starting from the drivers clutch and throttle pedals to the wheels for analysing vehicle performance during clutch engagement.

The previous works analysed the clutch parameters individually. Researchers worked on clutch torque capacity, clutch temperature, engagement characteristics and so on. But there is no available work in the literature that has worked on the complete analysis starting from the clutch pedal to the driving wheels. This study plays an important role in combining these analyses together and predicting the effects of different variables on vehicle performance.

In this study, a computer program is aimed to be developed with the following contents:

-During clutch engagement engine torque output will be determined from experimental engine torque characteristics.

-Clutch pedal displacement and engine throttle level scenarios will be used to simulate drivers feet motions.

-Thermal analysis will be carried out to predict the temperatures reached.

-The effect of temperature, sliding velocity and pressure on the coefficient of friction of clutch friction material will be taken into account.

-The program will include a vehicle model with total mass of the vehicle, rotating masses, gear ratios, tire radius and transmission efficiency to simulate a vehicle drive-train.

-The user will be able to compare the effects of different clutch system components and road conditions on different vehicles as well as drivers driving habits by plotting the vehicle performance parameters.

CHAPTER 2

INTRODUCTION TO PLATE CLUTCHES

2.1 Overview

The improvement of asbestos-based friction materials permitted the clutches to endure greater torque and above all, higher temperatures with smaller friction surfaces. The improvements on friction materials permitted reducing the number of the discs for a given type of use. At the end of the 20's and into the 30's, practically the only vehicles to retain the multiple-disc clutch were industrial vehicles, race cars and some high powered sedans. In single plate clutches, efficiency, performance and durability were prime objectives. Dry friction disc clutches offered the advantage that they were practically maintenance-free and, since they operated dry they didn't pose complex housing design problems.

Earlier plate clutches had shown problems similar to expanding shoe and cone clutches. However the torque capacity of the spring loaded dry plate clutch was predictable. The clutch was compact; it provided good heat absorption capacity and low output mass inertia. Therefore, it was readily accepted as the preferred clutch for automotive internal combustion engine application. Its attractiveness was further enhanced by successful development of low cost stamped steel cover plate assemblies.

2. 2 Coil Spring Clutches

This design offered some unquestionable advantages, among them a better distribution of the load on the trust plate and restrained axial dimensions. However, it was necessary to find the declutching system to work directly on the thrust disc. A very simple solution consisting of direct action from the control of a sleeve applied to the thrust disc had the disadvantage of requiring a very great actuating force. Then declutching system was invented that spread rapidly and remain in use for many years until the invention of the diaphragm spring clutch.

The design involved three levers arranged radially in equal distance positions. These levers rested on a thrust ring in the centre, which intern was activated by the pedal by suitable linkage, which conveniently multiplied the force exerted by the driver. One of the advantages of this system was declutching the levers through movement is directly against the flywheel. This meant that the thrust disc could simply come to rest on the levers during the declutching without having to be solidly connected to the mechanism.

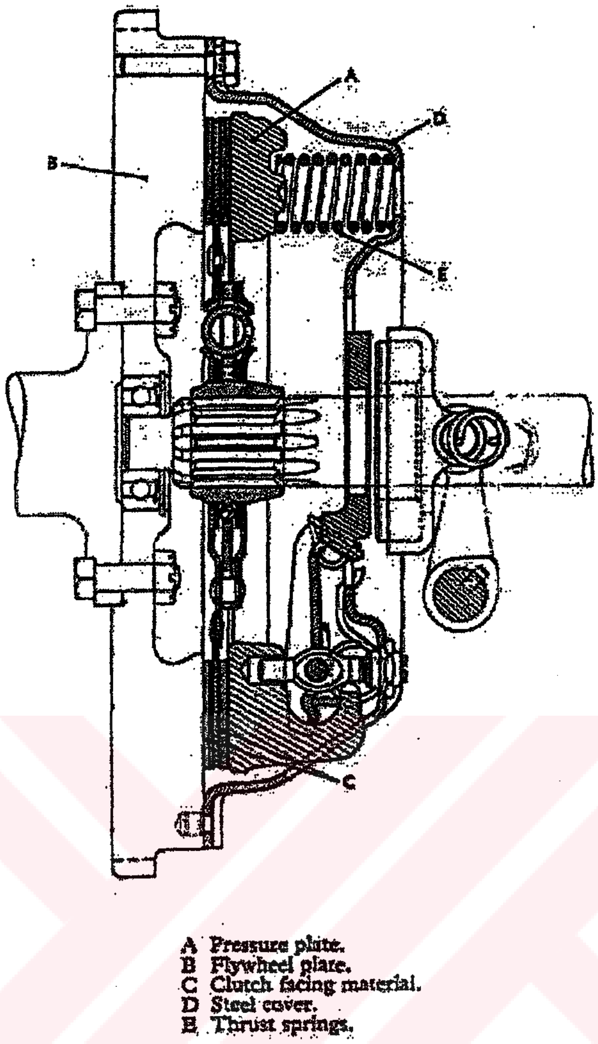


Fig 2.1. A Coil Spring Single Disc Clutch [20]

The coil spring clutch evolved from the single coil spring whose axis is coincident with the friction clutch axis of rotation, into a design with various springs arranged around in a circumference with the centre located on the axis of rotation as shown in Figure 2.1. Technically, functioning could be defined as an applied pedal force that surpassed 15-20 kg, therefore not too burdensome, and a 15-20 cm travel.

The cylindrical springs could not all be exactly equal and therefore the pressure on the plate was not uniform. And when there were initially different reactions to fatigue, there were different returns after a certain period. Moreover the cylindrical springs required an increasing force during the declutching phase. It was tiring to hold the pedal pressed down while rapidly losing load as the thrust disc plate approached to the disc. It was especially heavy when the disc is worn, because the clutch tended to slip more easily. Finally, with the increase in maximum engine speeds, the centrifugal effect increased on the declutching levers producing losses of the clutch useful load [20].

Coil spring clutches with cast iron covers in both single and twin plate designs were the universally accepted design worldwide for medium and heavy duty trucks until 1972. At that time, diaphragm spring clutches with stamped steel covers produced by Valeo were introduced by Renault Industrial vehicles. By 1987, usage of the older cast iron cover coil spring clutch was virtually eliminated in Europe in favour of the newer diaphragm spring designs with the great majority of these being the large single plate type.

2.3.Diaphragm Spring Clutches

The diaphragm spring clutch has almost completely replaced the traditional coil spring clutch in passenger cars, light trucks, and tractors. There are many reasons why the diaphragm spring clutch became so successful. Its simple, rotationally symmetrical design and the low axial height make it easier for the automotive designer to fit the clutch into today's shrinking clutch housings.

The single plate diaphragm spring dry clutch is composed of relatively few components as shown in Figure 2.2. These components are:

1. Clutch Cover
2. Pressure Plate
3. Diaphragm spring
4. Fulcrum rings
5. Fulcrum rivets
6. Rivets to attach drive straps to clutch cover and pressure plate.
7. Friction Disc

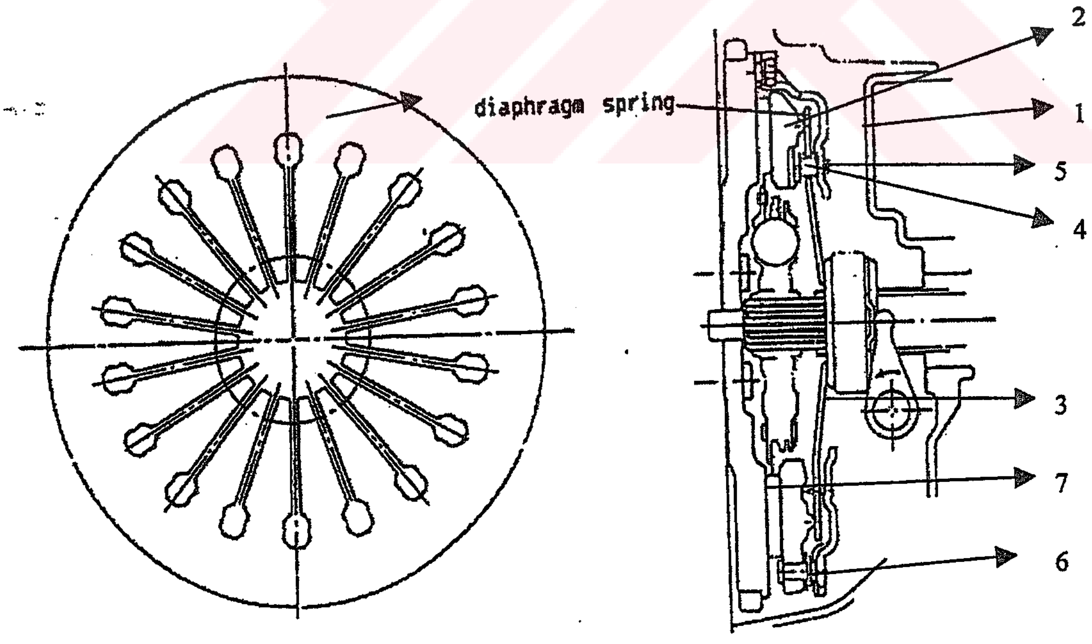


Fig 2.2. A Diaphragm Spring Clutch [20]

The flywheel is bolted firmly to the crankshaft, which transmits part of the engine torque to the clutch disc. The rest of the torque passes into the cover through the bolts holding the clutch to the flywheel. From there, the straps transfer it to the clutch pressure plate. The diaphragm spring clamps the pressure plate against the clutch disc to achieve positive lock-up with the flywheel. The clutch disc transmits the entire engine torque to the transmission with a splined connection to the transmission input shaft.

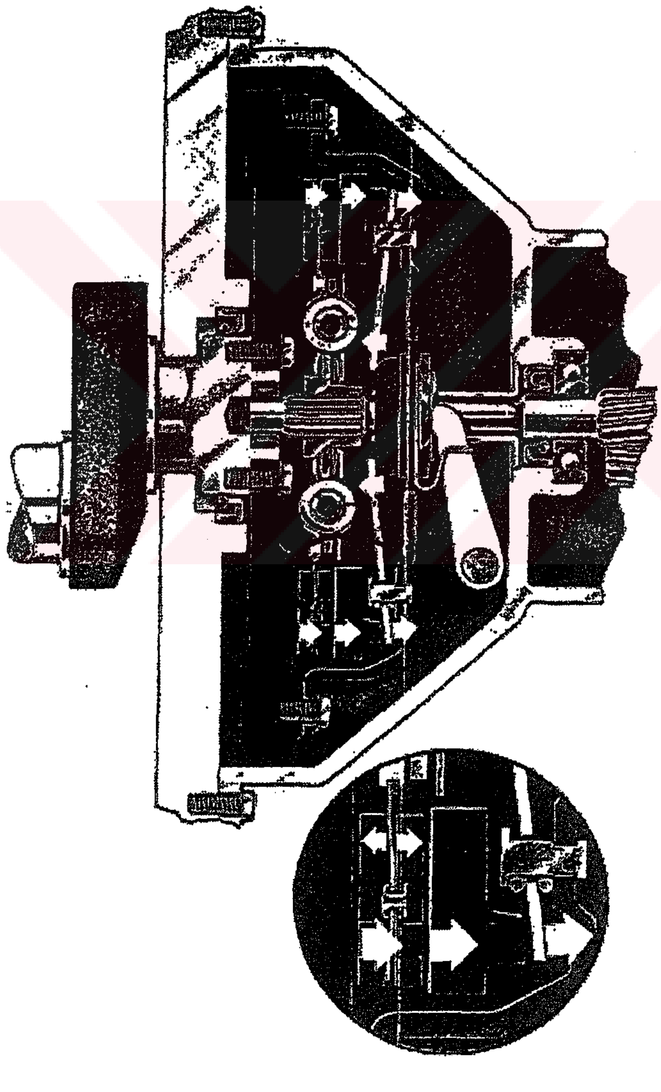


Fig 2.3. Disengagement of A Diaphragm Spring Clutch [20]

When the driver depresses the clutch pedal, the release bearing actuates the diaphragm spring levers as shown in Figure 2.3. The diaphragm spring is supported on a fulcrum ring attached to the clutch cover. The diaphragm spring pivots on this fulcrum ring, so that the outside diameter moves away from the pressure plate. This causes the preloaded drive straps to pull the pressure plate away from the clutch disc, interrupting torque transmission. When the driver releases the clutch pedal, the outer portion of the diaphragm spring acts on the pressure plate which clamps the clutch disc between the pressure plate and the flywheel.

2.4 Advantages of Diaphragm Spring Clutches

Design comparisons, installation measurements, drive evaluations, laboratory tests, fleet tests and theoretical analyses demonstrated the following advantages [20].

Ergonomic Advantages:

-Reduced pedal effort

-Improved clutch engagement quality

-Reduced transmission shift effort

-Improved noise and vibration problem solving capability

Operational Advantages:

-Easier installation

-Reduced and simplified in service maintenance adjustments

-Longer wear life

-Improved reliability

-Greater torque capacity under thermal loading

In the diaphragm spring clutch, the lever performs the release function. In spite of the high clamp load requirements, the diaphragm spring's regressive characteristic curve provides the low release loads demanded by the today's comfort minded drivers. Finally, when coupled with an appropriately tuned cushion characteristic in the clutch disc, the diaphragm spring offers excellent engagement performance and exceptional start-up comfort.

The coil spring clutch loses clamp load as the clutch disc wears because the coil spring is extending. The diaphragm spring gains clamp load as the clutch disc wears. The Figure 2.3 shows the load characteristics of the coil and diaphragm spring clutches.

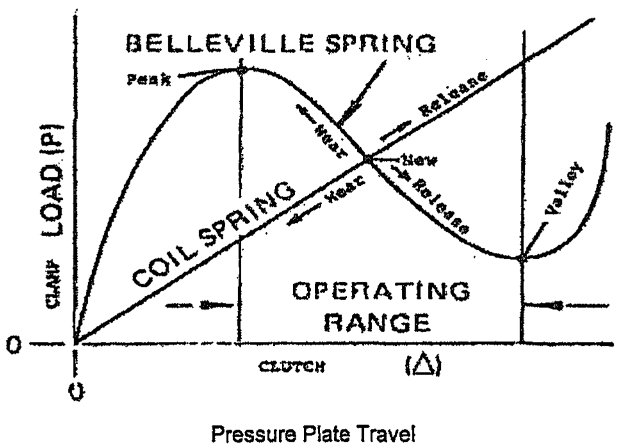


Fig 2.4. Load Characteristics of Diaphragm and Coil Spring [20]

The clutch is defined by three characteristic curves: the clamp load exerted on the pressure plate friction surfaces, release load and pressure plate lift as shown in Figure 2.5.

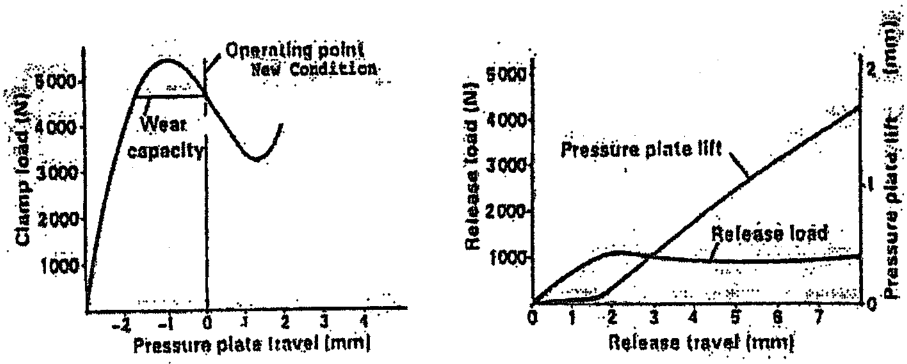


Fig 2.5. Diaphragm Spring Plate Load and Release Load Characteristics [20]

Clamp load is the force that presses on the pressure plate. Release load is the force on the clutch bearing. These two forces are linearly proportional to each other depending on the ratios of the diaphragm spring. (See Section 3.6).

Starting out in the new condition, the load of the diaphragm spring clutch slowly increases as the facings wear. Then it decreases to its original value until it reaches the wear limit. The clamp load remains nearly constant throughout the life of the clutch. The reason is in the special form of the diaphragm spring. Due to the internal lever ratio of a diaphragm spring clutch, the release load amounts to only 30 percent of the clamp load value.

The current trend is toward "low-lift designs" which reduce pedal loads even more than in the past. Here the internal lever ratio of the clutch has been further optimised, reducing lift off. The addition of the properly tuned cushion characteristic yields the good engagement performance that the customer demands. This design requires a special wear adjustment feature to retain the diaphragm spring.

The increased mechanical lever ratio used on the diaphragm spring type clutch results in greater clutch modulation control due to the slower rate of clutch pressure plate clamped load application during clutch engagement. Also it has predictable and repeatable pressure plate retraction and advancement versus release bearing position. This is because of the drive straps, which does not restrict pressure plate movement.

CHAPTER 3

DESIGN AND ANALYSIS OF FRICTION CLUTCHES

3.1 Torque Capacity of a Clutch

In a single plate clutch the elements of which are shown diagrammatically in Fig.3.1 p and N are the pressure and total axial load acting at the clutch interface respectively and R_i and R_o the internal and external disc radii.

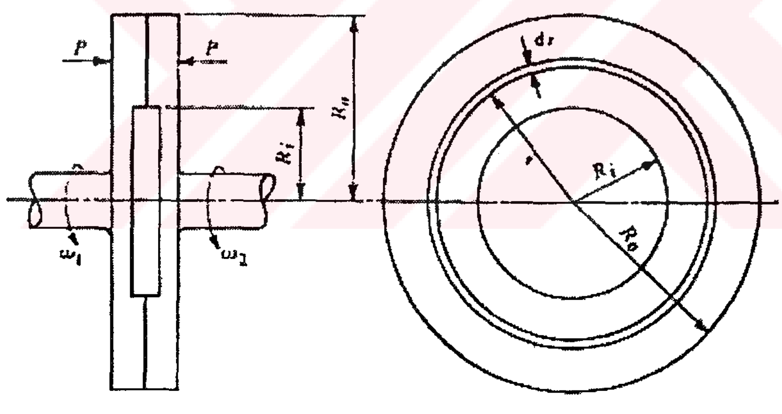


Fig 3.1. A Friction Disc [17]

If μ is the coulomb coefficient of friction between clutch face and disc, then the total axial load supported by the disc is:

$$N = 2\pi \int_{Ri}^{Ro} p.r.dr \qquad (3.1.1)$$

and the total torque capacity of the disc due to the tangential force on each radial element (clutch disc and pressure plate) is:

$$T = 2.\pi \int_{Ri}^{Ro} \mu.p.r^2 \qquad (3.1.2)$$

Before these equations can be integrated the manner in which p and μ vary with respect to radius (due to change in velocity) must be known. There is a considerable uncertainty about the distribution of axial load N over the contact surfaces. Between surfaces made of hard frictional materials such as sintered metals, contact is far from uniform and the pressure distribution therefore is highly irregular. With soft friction surfaces of low elastic moduli such as certain asbestos-resin materials, fairly uniform contact can be assumed and hence the pressure is independent of the radius.

Moreover, when the clutch life is considered, the rate at which surfaces wear depends not only on the pressure but also on the velocity of sliding between the surfaces. That is the rate of wear $f(p,v)$. Since there is little detailed knowledge as to the exact relation between the rate of wear, pressure and rubbing velocity, there are two alternatives. It will be assumed first that the rate of wear is directly proportional to the product $p.v$.

Thus,

$$Rate\ of\ wear = a_1.p.v \qquad (3.1.3)$$

and since v varies directly with r

$$Rate\ of\ wear = a_2.p.r \qquad (3.1.4)$$

If the coefficient of friction is assumed to be constant then equation (3.1.4) enables equations (3.1.1) and (3.1.2) to be integrated. In this instance when the rate of wear is uniform over the friction surface and μ is constant then

$$N = 2.\pi.p.r.(R_o - R_i) \tag{3.1.5}$$

$$T = \frac{N\mu}{2}(R_o + R_i) \tag{3.1.6}$$

for a single disc, and $T_n = n.T$ for n pairs of friction surfaces in contact providing each carries the same torque.

If however, p is constant

$$N = \pi.(R_o^2 - R_i^2).p \tag{3.1.7}$$

and if p is substituted in equation (3.1.2) and integrated then for a single disc;

$$T = \frac{2.N.\mu}{3}.\frac{(R_o^3 - R_i^3)}{(R_o^2 - R_i^2)} \tag{3.1.8}$$

With clutch discs in which R_o is not much larger than R_i, say R_o/R_i =1.25 to 1.35, then equations (3.1.6) and (3.1.8) give torque capacities which for all practical purposes are the same [17].

3.2 Analysis of the Vehicle Model

The vehicle model shown in Fig 3.2 is simplified to a two degree-of-freedom mechanical system containing input and output units having two rotating equivalent inertias I_1 and I_2 moving with two unequal (in general) rotational velocities ω_1 and ω_2 and loaded by torques T_1 and T_2.

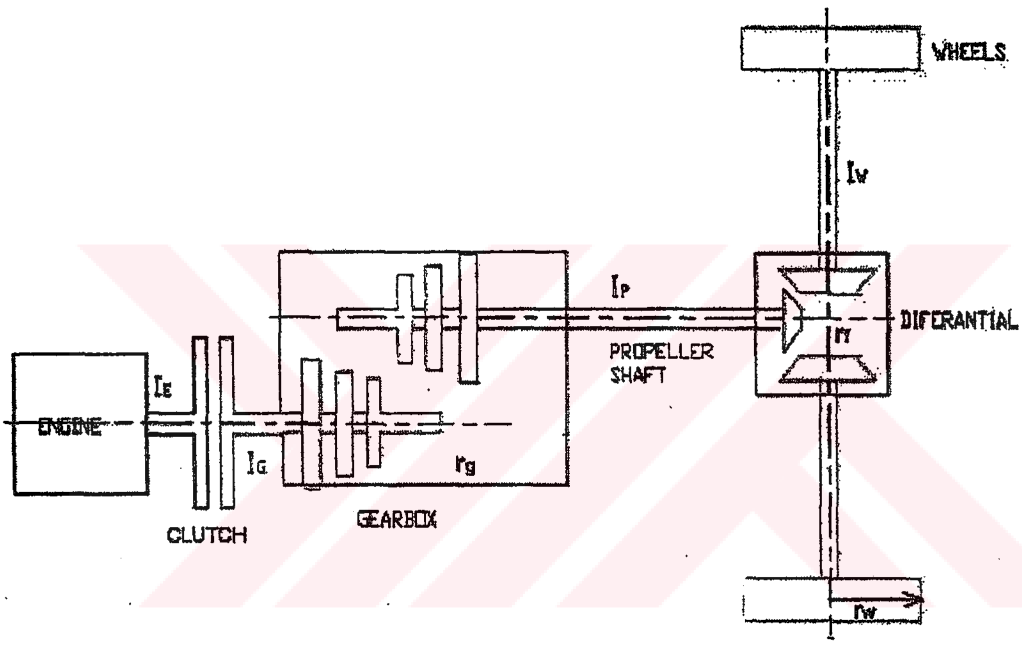


Fig 3.2. The Vehicle Model Used In the Analysis

Here I_2 is the moment of inertia of the clutch output side. It will be approximated by adding the moment of inertias of the parts rotating at propeller shaft speed and wheel speed reduced to the speed of the gearbox input shaft to the effective moment of inertia due to vehicle mass.. I_2 can be calculated as [25]:

$$I_2 = \frac{M.r_w^2}{r_g^2.r_f^2} + \frac{I_p}{r_g^2} + \frac{I_w}{r_g^2.r_f^2} + I_G \qquad (3.2.1)$$

The resistance torque T_2 is the torque on the clutch by rolling resistance, gradient resistance and the aerodynamic resistance of the vehicle. Here M is the total vehicle mass.

a) Torque on Clutch Due to Tire rolling resistance:

Tire rolling resistance can be expressed as: [25]

$$Frr = (a + b.V_{vehicle})Mg \qquad (3.2.2)$$

where a and b are constants depending on the type of the tires used. They are determined experimentally and given as:

$a = 0.011$ and $b = 0.000015$ for radial ply tires on asphalt road surface.

And $V_{Vehicle}$ is vehicle speed in kph.

This force creates a resistance torque on clutch axis:

$$T_{rr} = (r_w \,.\, F_{rr}) \,/\, (r_g \,.\, r_f \,.\, \eta) \qquad (3.2.3)$$

where η is the overall efficiency of the driveline components.

b) Torque on Clutch Due to Gradient resistance [25]:

$$T_g = (M \,.\, g \,.\, r_w \,.\, Sin(\theta)) \,/\, (r_g \,.\, r_f \,.\, \eta) \qquad (3.2.4)$$

c) Torque on Clutch Due to aerodynamic effects [25]:

$$T_a = (0.047.C_D.A_f.v_{vehicle}^2) \,/\, (r_g \,.\, r_f \,.\, \eta) \qquad (3.2.5)$$

where the vehicle speed is in kph . C_D is the aerodynamic drag coefficient, A_f is the area of front view of the vehicle. A_f can also be approximated by the expression;

$$A_f = 0.80.(Maximum\ height)(Maximum\ width) \qquad (3.2.6.)$$

The total driven side resistance torque on the clutch axis can be written as:

$$T_2 = T_g + T_{rr} + Ta \qquad (3.2.7)$$

The input unit in this model is an internal combustion engine that has a moment of inertia of I_1 and supplies a torque T_1. T_1 is a function of both the engine speed and the throttle level, which is controlled by the operator (driver). In order to properly calculate T_1, a detailed engine characteristic must be considered in the analysis. Figure 3.3 shows the torque characteristics of a gasoline engine for different throttle levels.

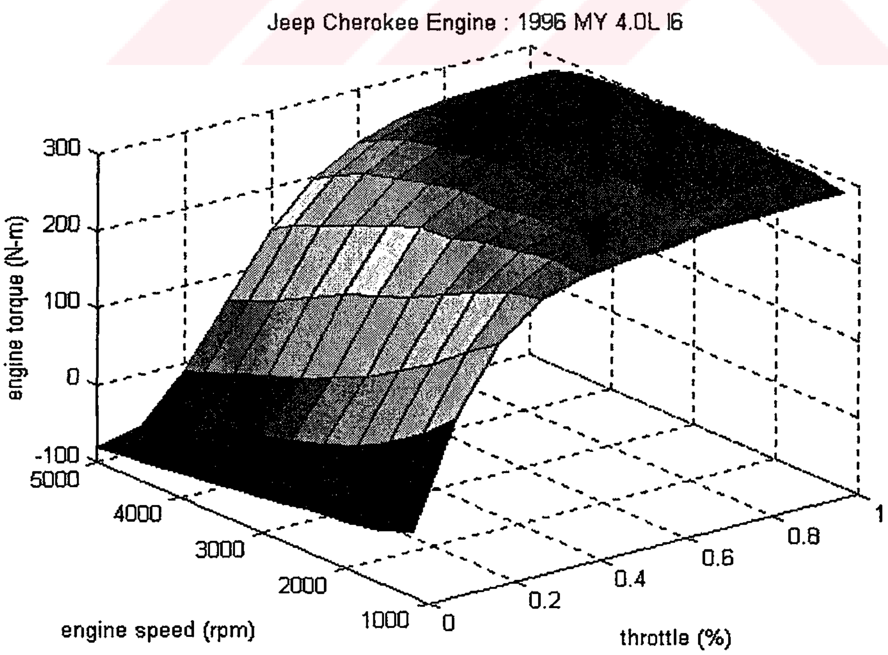


Fig 3.3. Torque Characteristics of 4.0 L. Jeep Cherokee Engine [27]

3.3 Clutch Engagement Model

Two rotating inertias I_1 and I_2 are involved which at the beginning, t=0, are moving with unequal rotational velocities ω_1 and ω_2 and at any instant t they are rotating at different angular velocities $\omega_1^{/}$ and $\omega_2^{/}$ respectively. Let I_1 and I_2 are the moment of inertias of the engine and the driven components respectively.

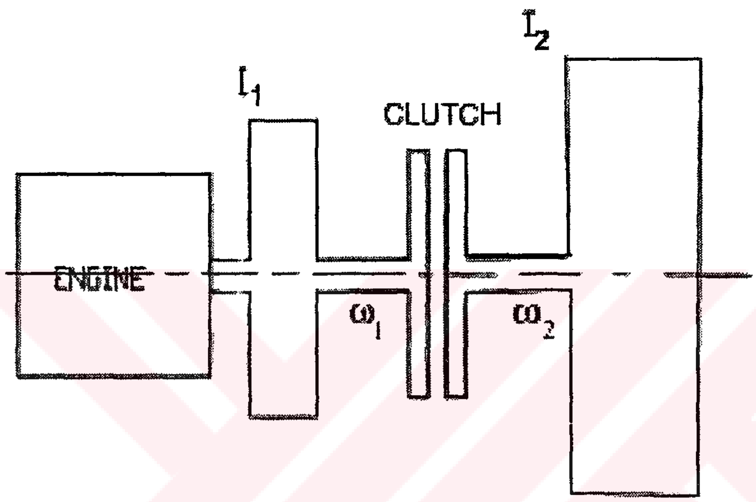


Fig 3.4. Simplified Dynamic Model

The clutch torque T_C is a function of coefficient of friction and normal force. The coefficient of friction is also a function of sliding speed, temperature and pressure. So a detailed coefficient of friction analysis should also be made. The clutch plate normal force depends on the clutch pedal travel and the diaphragm spring. This is a changing variable that depends on the driver throughout the engagement process.

The only way to analyse the engagement process is to investigate the process in very small time intervals and assuming clutch torque, engine torque, resistance torque and resistance inertia as constant in these time intervals. If the coefficient of friction and the normal force are constant and the system is assumed to be perfectly rigid, then the following dynamic relations exists [17],

$$T_1 - T_C = I_1 . \frac{d\omega_1}{dt} \tag{3.3.1}$$

$$T_C - T_2 = I_2 . \frac{d\omega_2}{dt} \tag{3.3.2}$$

Integrating these equations for constant time intervals and substituting the initial conditions give:

$$\omega_1^{/} = (\frac{T_1 - T_C}{I_1}).\Delta t + \omega_1 \tag{3.3.3}$$

$$\omega_2^{/} = (\frac{T_C - T_2}{I_2}).\Delta t + \omega_2 \tag{3.3.4}$$

By repeating these equations n times, the rotational speed of both the engine and the clutch can be calculated at time = $n.\Delta t$.

3.4 Models for Variation of Coefficient of Friction

Friction basicly arises from two sources: 1) Mechanical interlocking of asperities between reaction surfaces, and 2) Chemical adhesion of the reacting surfaces. The first phenomenon accounts for μ being a result of normal versus applied pressure. But the second and more elusive factor depends on the activation energies between molecules of the opposing surfaces, and it is temperature sensitive. Therefore in trying to determine the μ of a particular friction material one must consider not only an absolute characteristic number that can be applied to any new design, but one which is a function of many factors:

a. The composition and type of manufactured form of the friction material and opposing plate.

b. The temperature, speed, and pressure at which μ is determined.

c. Environmental contamination and prior usage history of the friction application.

There are two types of μ, dynamic (μ_d) and static (μ_s) which effect the smoothness of engagement in the transition from sliding to lock-up of the surfaces.

Most clutch design engineers will base their calculations on a minimum coefficient of friction of around 0.27 to 0.28 [20]. A typical range of friction found in clutch materials would be 0.3 to 0.4. Coefficient of friction of a friction material for a particular application is the single most sought-after number that a clutch designer needs. Unfortunately, it is often the most elusive and requires extensive lab and field experience with the particular design in order to achieve 90% confidence that the coefficient will fall within an often wider than desired range of values. It needs to be emphasised that dynamic coefficient of friction is not a fixed number even for a single and perfectly manufactured friction formulation. The coefficient varies with temperature, sliding speed, pressure, length of engagement, presence of wear particles, contamination, condition of mating surface, air flow in the assembly, and other parameters. In addition, variations in these parameters may influence other parameters. No successful statistical approach to quantifying and predicting these variables has yet been evident in the literature. Table 1 shows typical values for one commonly accepted clutch facing. Note how these values vary from one test to another.

During vehicle testing, it is not uncommon to find internal clutch housing air temperatures above 200°C while a facing is failing [16]. Dynamometer test stand measurements have exhibited facing surface temperature with standing spikes above 650°C for milliseconds, where extended exposure at these temperatures would have rapidly failed all conventional clutch facings [18].

Loss of friction due to increased temperature is one of the most important parameters. Decrease in coefficient can not be allowed beyond that minimum at which insufficient torque is developed to engage the maximum specified load of the clutch design.

As the coefficient decreases with heat, slip time increases and this leads to further temperature increase, which decrease the coefficient even more. Slippage and subsequent heat increases can result from overload, oil contamination, or heat build-up. Heat build-up can result from repetitive fast cycling such as those abusive conditions produced by vehicles attempting to move out of snow or mud.

Clutch chatter (also called clutch judder) is very annoying to the driver. Clutch facings must engage and disengage smoothly. This phenomenon has often been explained as an audible low frequency vibration created by the quickly alternating sticking and slipping conditions, which result from the differences between the static and dynamic engagements. Generally chatter is closely related to the ratio of static to dynamic coefficients [13]. According to this theory, the closer the ratio is to 1:1, the smoother engaging the clutch. This is a rule of thumb rather than a hard and fast rule. In some wet clutch systems, chatter has been attributed as much to a steep negative slope in the coefficient versus sliding velocity curve.

In the literature some empirical relations are formed by dynamometer tests which allows the extraction of the variation of coefficient of friction with sliding speed, temperature and clamp pressure for the particular configuration under test. A technique used by Lucas [11] is given as:

$$\mu = \frac{\mu_{static} \cdot \mu(p)\mu(T)\mu(v)}{\mu v \, \mu T \mu p} \qquad (3.4.1)$$

where $\mu(p)$, $\mu(T)$, $\mu(v)$ are the polynomials at the form

$$\mu = A_0 + A_1 \varpi + A_2 \varpi^2 ....$$

and μv, μT, μp are the are the normalising coefficients given and the constants A_0, A_1, A_2 .. are also given in this paper. The results taken with this equation are not quite realistic so a different approach is needed to be applied.

Newcomb [17] has given an empirical equation for the μ versus $v_{sliding}$ relation as:

$$\mu = \mu_{high} + (\mu_{static} - \mu_{high})\, e^{(-a.v)} \qquad (3.4.2)$$

where μ_{static} is the static coefficient of friction and μ_{high} is the coefficient of friction at high speed and "a" is a constant depending on the type of the friction material. This approach does not include the variation of coefficient of friction with temperature and pressure, so it must be used with some other equations.

μ versus. Temperature graphs given by Ortwein [21] can be used to determine the temperature effect on the friction coefficient. By curve fitting one can obtain polynomials of temperature for different friction materials. The pressure effect will be neglected in the analysis because there are no available equations to combine the pressure effect into the temperature and sliding speed equations in the literature.

3.5 Thermal Analysis of Clutch Engagement

Figure 3.5 shows a cross-section of a single plate clutch under the action of an axial clamping force N. The two outer plates are steel of different thicknesses d_1 (pressure plate) and d_3 (flywheel plate) whilst the inner plate consists of two friction plate materials riveted thin spring steel segments, the overall thickness being denoted by $2d_2$. Rubbing surfaces can be thought of as plane sources of heat, producing $q(t)$. At the rubbing interface the surface temperature of both plates 1 and 2 is the same. The rates of flow of heat into 1 and 2 are not equal but depend on the thermal properties of two contacting members.

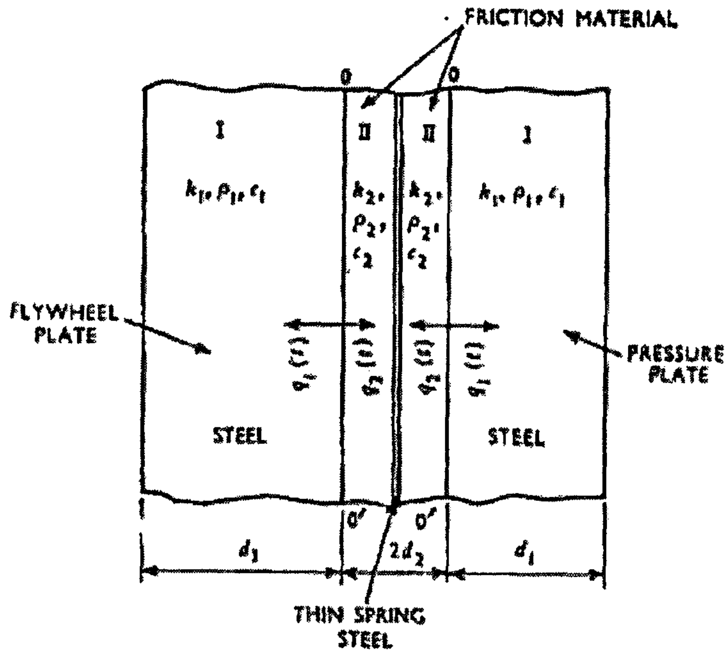


Fig 3.5. Section of Pressure Plate, Friction Disc and Flywheel Plate [17]

In normal clutch applications the slipping period t_s is at most a few seconds in duration and in this case the contacting bodies can be considered to be infinite in thickness since only the immediate vicinity of the friction surface is heated in time *ts*. This heat conduction problem has been considered by Newcomb [17, 18]. The surface temperature rise is given in the form:

For $t < ts$:

$$\Delta T = \frac{2\,NNC\,t^{1/2}}{\pi^{1/2}}(1 - \frac{2t}{3ts}) \tag{3.5.1}$$

and For $t > ts$:

$$\Delta T = \frac{2\,NNC}{\pi^{1/2}}(t^{1/2}(1 - \frac{2t}{3ts}) - (t - ts)^{1/2}\frac{2}{3}(1 - \frac{t}{ts}) \tag{3.5.2}$$

where,

$$C = \frac{1}{\sqrt{(k_1.\rho_1.c_1)} + \sqrt{(k_2.\rho_2.c_2)}} \tag{3.5.3}$$

$$NN = \frac{T_C}{A}(\omega_1 - \omega_2) \tag{3.5.4}$$

The equation (3.5.2) refers to the temperature in the cooling period due to conduction alone. When the equation (3.5.1) is applicable, the maximum temperature of the surface at the end of slip period is given by:

$$T = 0.53.NN.t_s^{1/2} \tag{3.5.5}$$

The bodies considered are a typical asbestos-friction material slipping against steel, where

k_1=44.982 W/m.K, ρ_1=7800 kg/m^3 , c_1=502.38 J/kg K for steel and

k_2=0.75386 W/m.K, ρ_2=1796.5 kg/m^3 , c_2=1255.95 J/kg K for the friction material [17].

The equations (3.5.1) and (3.5.2) allow the transient temperatures during a clutch engagement to be calculated. Eventually however, the steel members and clutch facing reach an average temperature, and knowledge of this value is important when repeated engagements are to be considered.

If the relative rotational velocity at the instant t is (ω_1-ω_2) then rate at which energy is dissipated in the clutch during slipping is given by [17]:

$$q(t) = \frac{T_C.(w_1 - w_2)}{A} \tag{3.5.6}$$

The total energy dissipated Q at time $t = n.\Delta t$ is

$$Q = \sum_{i=1}^{n} A.q(t) \tag{3.5.7}$$

If all the heat generated during one application is uniformly distributed throughout the contacting members, the rise in average temperature ΔT for a steel pressure disc of weight PPW and specific heat c_1 is:

$$\Delta T = \frac{\gamma Qt}{PPW\, c_1} \tag{3.5.8}$$

and

$$\Delta T = \frac{(1-\gamma)Qt}{Ad_2 c_2 \rho_2} \tag{3.5.9}$$

For clutch facing.

where $\gamma = \sqrt{k_1 \rho_1 c_1}\, C$ (3.5.10)

For a typical asbestos-friction material and steel engagement (using the parameters given above) and using d_2 = 3.6 mm with a pressure plate weight of 4.5 kg, $\Delta T = 0.804\ Qt$ for the friction disc. Thus both members approximately attain the same average temperature after the slipping period.

The previous equations were given for single engagements but in the normal operation of a friction clutch, repeated engagements are made and the average temperature of the assembly during operation under these conditions must be determined. In a single engagement the friction surface temperature is highly transient in nature since the slipping period is usually less than a second. The heat developed flows into the components of the assembly until all are at a uniform temperature. This occurs within a few seconds and afterwards the heat transfer is mainly convection and radiation [17]. In a well ventilated clutch, the principal method of heat transfer is by forced convection to the atmosphere, provided temperatures of the order of 500-600 °C are not exceeded when radiation loses must be included. In these circumstances Newton's law of cooling may be used provided that the following conditions are satisfied:

1. The temperature throughout the assembly, including the friction surfaces, is the same at any instant.
2. The coefficient of heat transfer at the exposed surface is a constant, independent of temperature.

These assumptions are approximately satisfied in practice if the thermal conductivities are high.

Newton's law of cooling states that when a solid body of weight W, specific heat c_l, and exposed surface area A^l cools slowly from an initial temperature $T_{initial}$, the temperature T at any subsequent time t is given by:

$$T - T_{Ambient} = (T_{initial} - T_{Ambient}).e^{(-b.t)} \qquad (3.5.11)$$

where $b = \dfrac{A^{/}.h}{W_1 c_1}$ and corresponding to the body considered. h is the coefficient of heat transfer, $T_{Ambient}$ is the ambient temperature. Applying the above law of cooling to intermittent clutch applications made at equal intervals of time, the temperature reached after the n^{th} $(n>=1)$ engagement is given by:

$$T = T_{Ambient} + \Delta T.\left[\frac{1-e^{-n.b.t_0}}{1-e^{-b.t_0}}\right] \tag{3.5.12}$$

ΔT is the increase in average temperature during a single engagement = $Qt/W.c_1$, Qt is the total heat entering the body and t_0 is the regular time interval between clutch applications.

The steady state values of T is obtained by taking n large. Thus

$$T_{Limit} = T_{Ambient} + \frac{\Delta T}{1-e^{(-b.t_0)}} \tag{3.5.13}$$

In general $b.t_0$ is small and a good approximation can be made as:

$$T_{Limit} = T_{Ambient} + \frac{\Delta T}{b t_0} \tag{3.5.14}$$

which reduces to:

$$T_{Limit} = \frac{Qt}{A^{/} h.t_0} \tag{3.5.15}$$

where coefficient of heat transfer can be determined experimentally by measuring the rate of cooling of the assembly when operating at similar conditions to those during continuous sliding. Thus

$$h = 0.055\frac{k}{L}R^{0.75} \tag{3.5.16}$$

which virtually implies that k is proportional to 0.75 power of the air velocity past the body since other terms are substantially constant [17]. If a typical automotive clutch pressure disc is considered L can be assumed as 54 mm and R can be taken as a average value of 21110. Then h = 43.17 W/m^2 K can be used as a typical value for the calculations.

From the above equations it can be seen that the temperature of the clutch components during repeated cycling depends mainly on the following factors:

1. The energy dissipated, Qt into the system during a cycle.

2. The coefficient of heat transfer. This has a considerable influence on the temperatures attained and depends on how well the surfaces are exposed to the air stream. In a well ventilated clutch cooled by turbulent air flow the coefficient of heat transfer is proportional to $v^{0.75}$.

3. Thermal capacity of the component considered, for bulk temperature equation contains this term and shows that an increase in value of $W.c_1$, that is a decrease in the value of b, causes a reduction in the rate of increase in average temperature.

Figure 3.6 shows the variation of bulk temperature with n (number of engagements) for pressure plates weighing 4.5 , 6.8 , 9 , 13.6 *kg* when the areas exposed to atmosphere are 86770 mm^2, 91788 mm^2 , 96804 mm^2 , 106745 mm^2 respectively and same Qt is used.

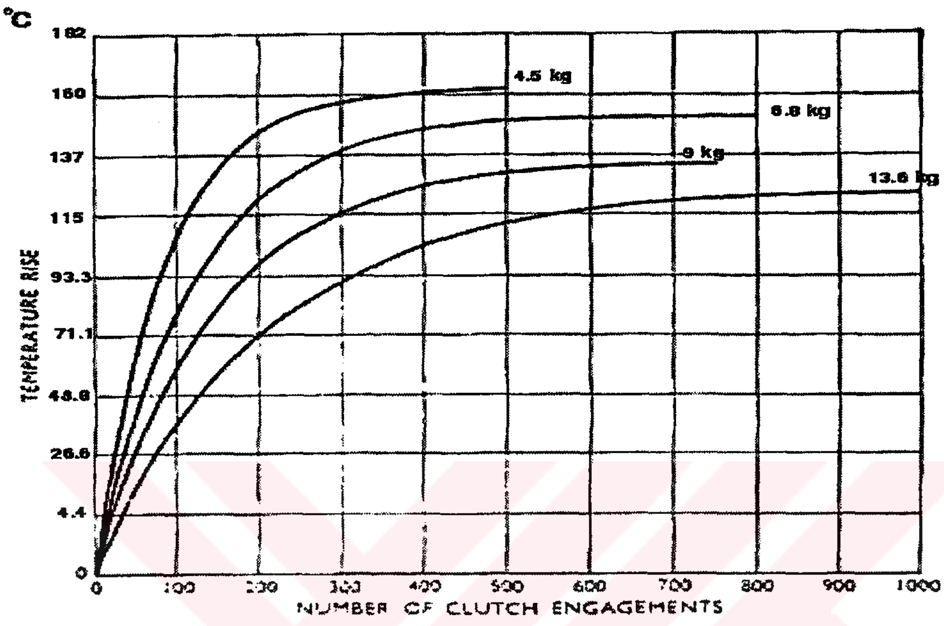


Fig 3.6. Steady State Temperatures Reached by Different Size Pressure Plates [17]

These curves show that the more massive the pressure plate, the less rapid the rate of rise in bulk temperature with number of clutch engagements for the same t_0. Furthermore the limiting temperature attained decreases with increase in the weight of the pressure plate.

3.6 Diaphragm Spring Design

3.6.1 Characteristics

The diaphragm spring curve resembles that of a Belleville spring, because it is a modified Belleville spring with long levers.

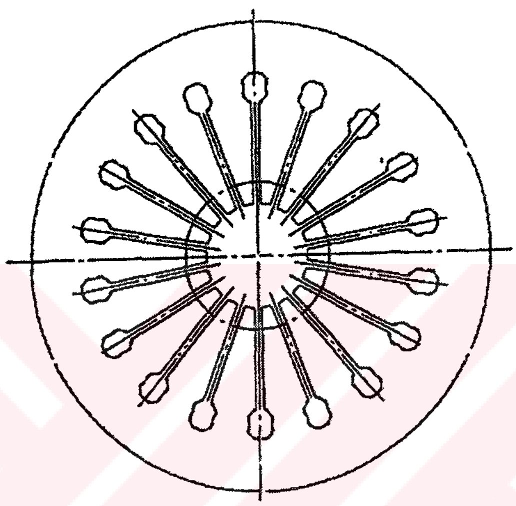


Fig 3.7. A Diaphragm Spring [20]

To properly design the diaphragm spring, two factors to be considered are the clamp load requirements and the bearing travel. These factors influence the basic size of the diaphragm spring. An advantage of the diaphragm spring clutch design is that it achieves a relatively high clamp load with low deflections. The diaphragm spring is a stamped steel ring with multiple levers. The spring is heat treated to achieve the desired operational characteristics.

Often supported at the edges only, these springs can be loaded beyond flat. Therefore have an even larger deflection range with very small load variation, making them ideally suited for many clutch applications.

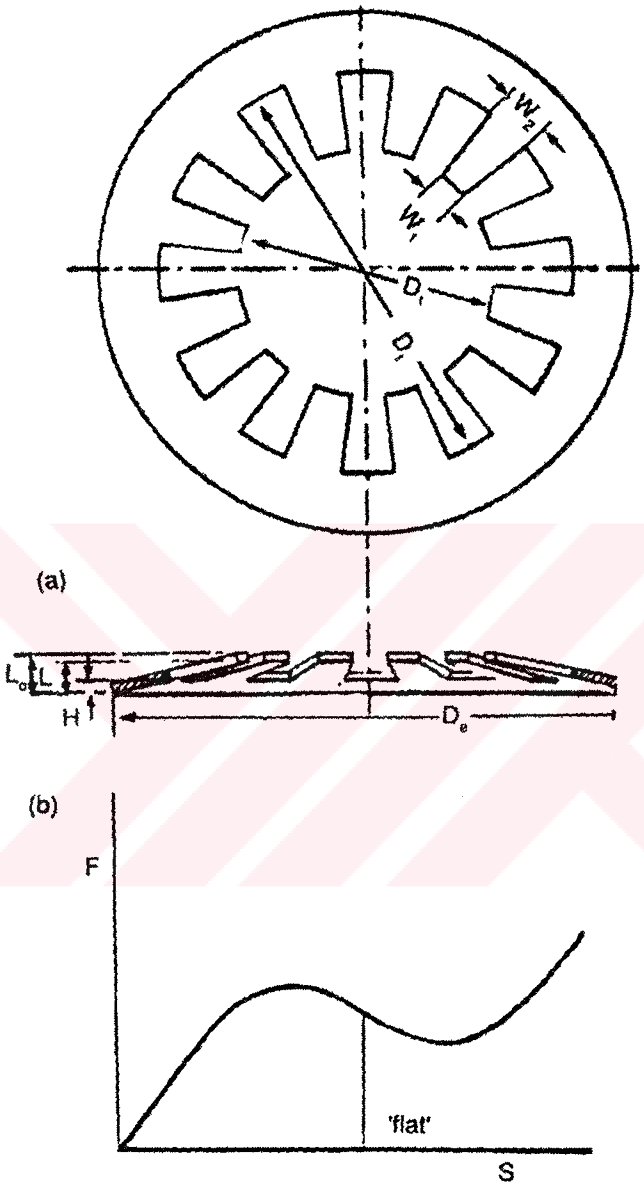


Fig 3.8. Diaphragm Spring Dimensions and Load Characteristics [22]

Theoretically, this spring can be treated as a standard disc spring with the load acting "outside" the inner diameter *Dt*. Such an approach would neglect the effect of bending of the fingers or tongues. The following design equations apply when the bending of the tongues is taken into account and the load is applied at the edges [22].

Load F is as a function of material, dimensions and deflection S_1, which is the deflection of the closed ring section. That is the same deflection as if the inside of the closed ring section is pivoted and outer side of the spring is deflected.

F is given with the following equation:

$$F = \frac{4.E}{(1-\upsilon^2)} \frac{T^3 S_1}{K_1.De^2}\left[1+(\frac{H}{T}-\frac{S_1}{T}).(\frac{H}{T}-\frac{S_1}{2.T})\right]\left[(1-\frac{D_t}{D_e})/(1-\frac{D_i}{D_e})\right] \quad (3.6.1)$$

where K_1 is a factor and defined as:

$$K_1 = \frac{1}{\pi}.\frac{(\frac{R-1}{R^2})^2}{\frac{R+1}{R-1}-\frac{2}{\ln R}} \quad \text{or} \quad K_1 = \frac{6}{\pi.\ln R}\left[\frac{(R-1)^2}{R^2}\right] \quad (3.6.2)$$

and,

$$R = \frac{De}{Dt}$$

Both equations for K_1 can be used. The results are the same for all practical purposes [22].

This type of spring will show a significantly larger deflection than the regular disc spring. The closed ring section can be designed with a ratio H/T=1.4 which provides an equal force transmission in the closed ring section. In other words if

H/T ratio is equal to 1.4, the force applied on the inner part of the closed ring section is equal to the force on the outer part of the closed ring. The Figure 3.9 shows the calculated characteristics for disc springs with different ratios H/T.

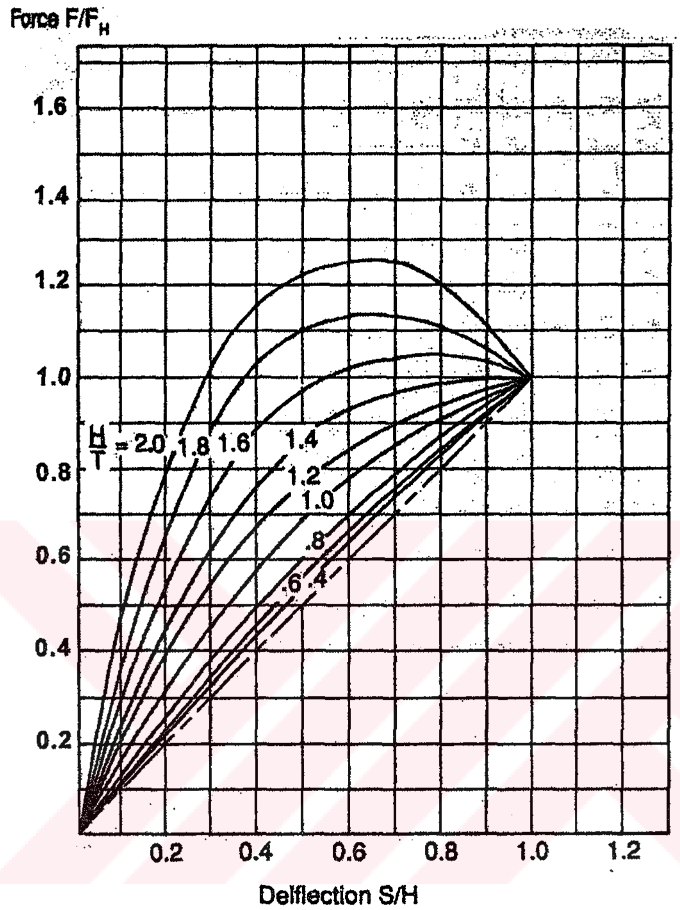


Fig 3.9. Effect of H/T Ratio on Spring Force [22]

3.6.2 Clamp Load

The clamp load characteristics of a diaphragm spring clutch is not a linear function like a coil spring lever type clutch. The clamp load curve naturally resembles that of a Belleville spring. Figure 3.10 shows a typical clamp load curve [20]. Normally clamp load is plotted against the pressure plate travel, or diaphragm spring deflection.

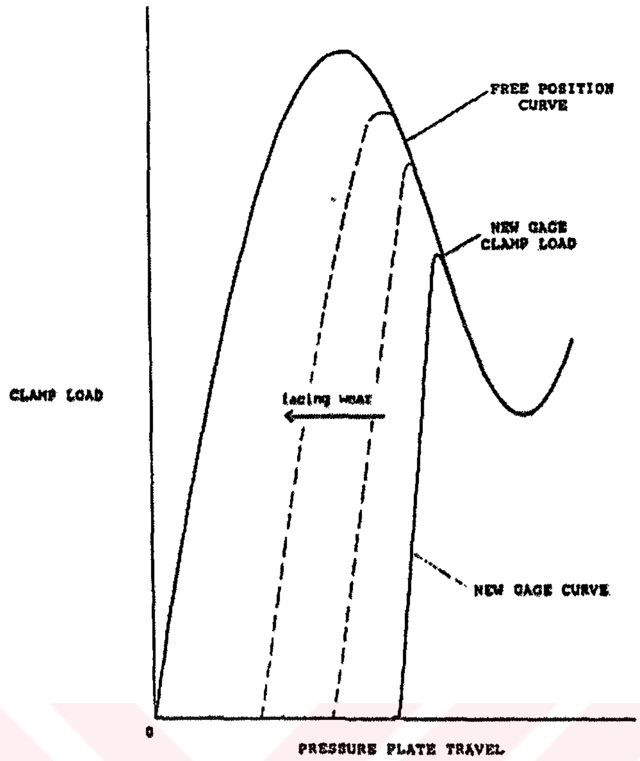


Fig 3.10. Clamp Load Curve of a Diaphragm Spring [20]

The clamp load curve in the Figure 3.10 represents a measurement taken without a clutch disc or gages installed under the pressure plate. This condition is described as the free position. When the clamp load curve is taken with gages representing the compressed thickness of a new clutch disc installed under the pressure plate, this condition is described as the new gage position. The new gage curve indicates a clamp load for a specified new clutch disc compressed thickness.

When the free position curve and the new gage curve are plotted together, the point of intersection represents the clamp load for a fully engaged new clutch disc. The new gage curve represents one of an entire family of curves that can exist inside the basic free position curve. As the gage height changes, the clamp load will change, the thinner gages causing clamp load values move to left on the clamp load curve, which is represented by the dashed lines in the Figure 3.10. In case of the automotive diaphragm spring clutch, as the clutch facings wear, the clamp load will change. One of the unique characteristics of the diaphragm spring clutch is that during the initial stages of clutch facing wear, the clamp load will increase. Continued wear may eventually result in decreasing clamp loads if the clamp load goes over the hump in the curve. If sufficient wear travel is available in the clutch, clamp load during wear will eventually decrease to a value below the minimum required. This point would indicate the maximum wear limit due to clamp load for this particular package. The total amount of wear provision available in a clutch cover assembly due to clamp load can be maximised by designing diaphragm springs capable of large deflections, limited only by maximum stress [20].

Figure 3.11 shows the load limits of a diaphragm spring. As seen in the Figure the clamp load increases significantly with the increasing facing wear. Most automotive clutch applications have diaphragm spring wear provisions around 2 mm. Diaphragm springs designed with 2 mm wear provision can usually maintain an acceptable pedal effort throughout the life of the clutch.

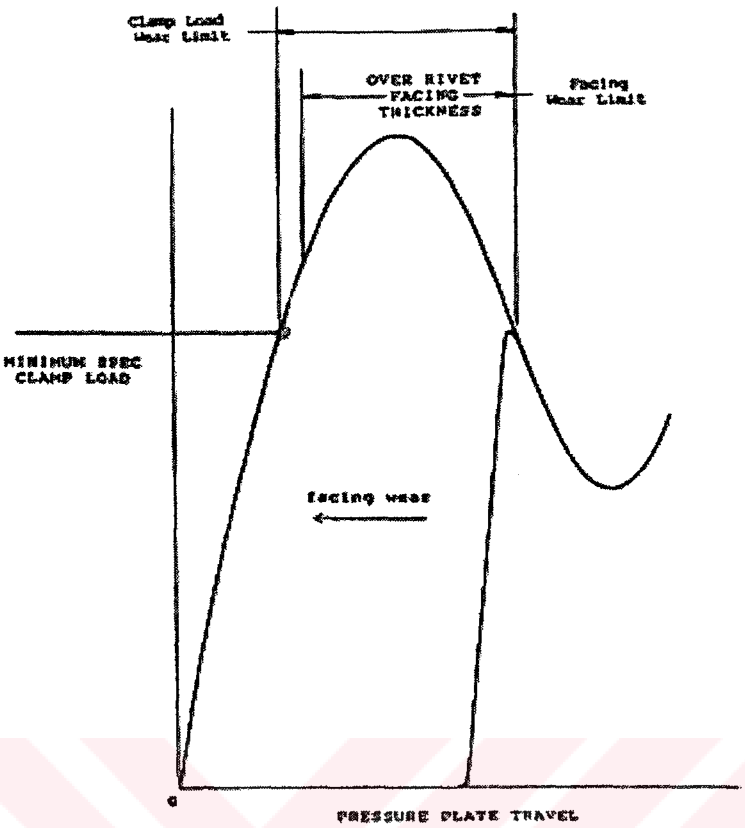


Fig 3.11. Facing Wear Effect on Diaphragm Spring Force Characteristics [20]

The clamp load curves displayed in figures represent the clamp loads that can be achieved during a complete engagement condition. They do not represent the clamp load applied on the clutch disc during clutch modulation. The clamp load applied to the clutch disc during modulation is a function of the clutch disc cushion curve.

3.6.3 Release Load (Bearing Load)

Release load is the force required to actuate the clutch cover assembly. This force is usually applied through the release bearing. The bearing release load is a function of clamp load and the lever ratio. The release load curve shape looks similar to that of the clamp load curve as shown in Figure 3.11.

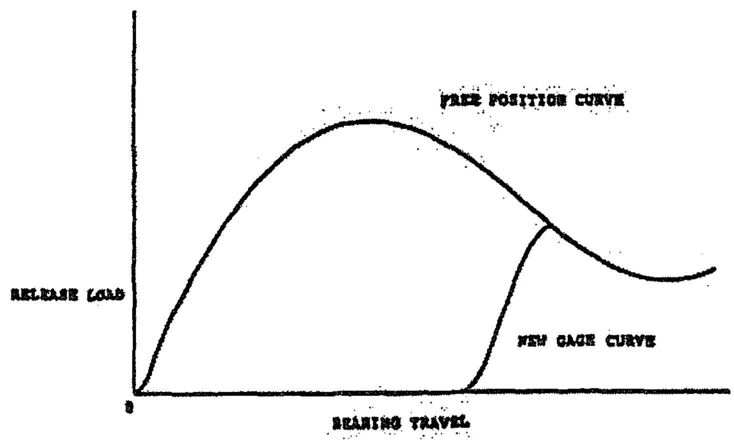


Fig 3.11. Bearing Load Characteristics of A Diaphragm Spring Clutch [20]

The basic characteristics and behaviour of the release load are also similar to that of the clamp load. The release load, when measured over gages, will indicate the load characteristics for a specified condition. During clutch facing wear, the release load will also behave similarly to the clamp load.

The release load is the clutch characteristic that is most noticeable to the vehicle operator. The release load is directly related to clutch pedal efforts, with higher release loads developing higher clutch pedal efforts. As the clutch pedal is depressed, the pedal effort initially starts to increase. Then part way through the pedal stroke, the load begins to decrease. This is a unique characteristic of the diaphragm spring clutch. The actual clutch pedal hold down effort is quite manageable even with high peak clutch pedal efforts. Conversely a conventional coil spring/lever type clutch will have a consistently increasing pedal effort as the clutch is depressed.

The addition of fingers provides the diaphragm spring with an increased distance of travel. The fingers act as a number of cantilever beams extending from the inner diameter of the belleville spring. These fingers allow for the spring to gradually load as the fingers are deflected. Each finger has the effect of restricting the natural circumferential strain of the belleville washer at the inner diameter of the washer. This effect shifts the effective inner diameter of the washer towards the center of the spring. This theory defines the existence of an effective diameter, which can be evaluated with the following equation.

$$Deff = Di - \left\lfloor Di\ 10^{-3}\,(25.6 - 0.483\,.\ No\ of\ fingers)\right\rfloor \tag{3.6.3}$$

Bearing load can be calculated by using the diaphragm spring dimensions.

$$BL = \frac{4\pi\, E\, t\, s1\, (Log(\frac{De}{Di})\left[(H - \frac{s1}{2})(H - s1) + t^2\right]}{6\, fulcratio\left[\frac{(De - Di)^2}{4}\right]} \tag{3.6.4}$$

Here $fulcratio = \dfrac{(De - Di)}{(Dt - 2\, Deff)}$ (3.6.5)

Deflection of the tongues is given by [22],

$$S_2 = C\frac{(Dt - Di)^3 (1 - \mu\ )}{2 E t^3 W_2 Z} BL \tag{3.6.6}$$

where C is a constant depending on the ratio W_1/W_2 and Z is the number of tongues.

Table 3.1 Deflection Coefficient of Tongues [22]

W_1/W_2	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
C	1.31	1.25	1.20	1.16	1.12	1.08	1.05	1.03	1.0

3.7 Clutch Pedal Travel

The maximum distance the clutch pedal travels from top to bottom of the stroke is known as pedal travel. There are several factors which determine an optimised pedal travel. Considering existing ergonomic data from independent sources, the recommended maximum clutch travel should not exceed 175 mm [20]. In general, current passenger car clutch travels average around 150 mm. This distance appears to be adequate for the normal range of vehicle operators.

There are several critical areas of clutch pedal travel that affect the perceived clutch engagement quality and overall ease of operation. These are clutch pedal reserve distance, the first engagement point, and the engagement modulation zone. These zones are shown in Figure 3.12 in detail. Clutch pedal reserve is the distance from point where the clutch is completely disengaged, to the end of the clutch pedal travel. This point is normally encountered and measured on the clutch pedal downstroke. The clutch pedal first engagement point where the clutch is transmitting enough torque to move the vehicle from standstill. This point is normally encountered and measured on the clutch pedal upstroke. The clutch pedal engagement modulation zone is the distance from the first engagement point to the point where the clutch is completely engaged (Engine speed equals transmission input speed). This area is normally encountered and measured on the clutch pedal upstroke.

A clutch pedal reserve of at least 25 mm is desirable in order to ensure a positive clutch release at all times [20]. The clutch and actuation system can experience additional deflection when the clutch system is hot. This situation will decrease the reserve distance. A very low reserve distance may not allow the clutch to completely release unless the clutch pedal is depressed all the way to the floor. The lack of reserve distance can cause gear lash, higher shifting efforts and synchroniser wear when shifting. The reduced disengagement zone caused by low reserve also requires a much more precise coordination of clutch, throttle and shift lever in order to perform a smooth shift.

The clutch pedal engagement or modulation zone starts at the first engagement point and ends when the clutch torque capacity varies from zero (at the first engagement point) to maximum clutch torque (near the top of the pedal travel), the engagement zone will vary along the pedal travel. Complete engagement at maximum engine torque occurs near the top of the pedal travel while complete engagement for part throttle low engine torque occurs closer to the first engagement point. The torque transmitted through the clutch during any modulation of the pressure plate is largely a function of the clutch disc cushion curve. Since most cushion curves start out with a low spring rate and progress to a very steep rate near their maximum deflection, in the normal driving range most clutch engagements will occur in a very narrow pedal travel range. The clutch pedal reserve point depends on many parameters and it is also an adjustable parameter. In this study only the clutch modulation zone will be taken into consideration. The clutch pedal travel zones are shown in Figure 3.13.

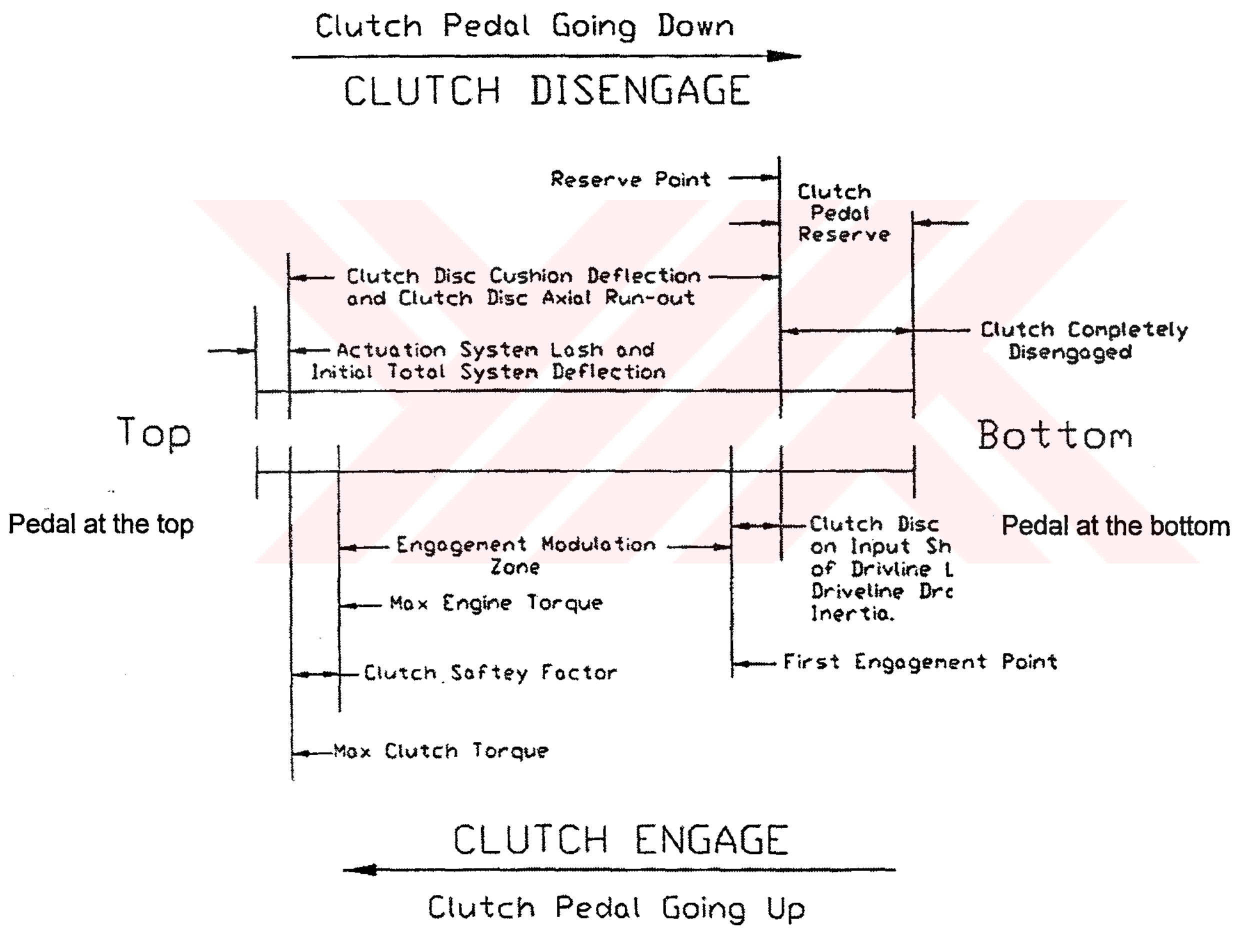


Fig 3.13 Clutch Pedal Travel Zones [20]

3.8 Clutch Actuation System

The clutch actuation system converts the force and travel of the operator's (driver's) foot on the clutch pedal into a larger force and smaller travel by the release bearing acting on the clutch release levers.

There are several types of clutch actuation systems. These actuation systems can either push or pull on the clutch release levers. The decision as to which system is best suited for a particular application will be based on performance needs, service life, reliability, available vehicle space, ease of assembly to the vehicle, weight and cost. Some examples of these actuation systems are seen in Figures 3.14 to 3.16.

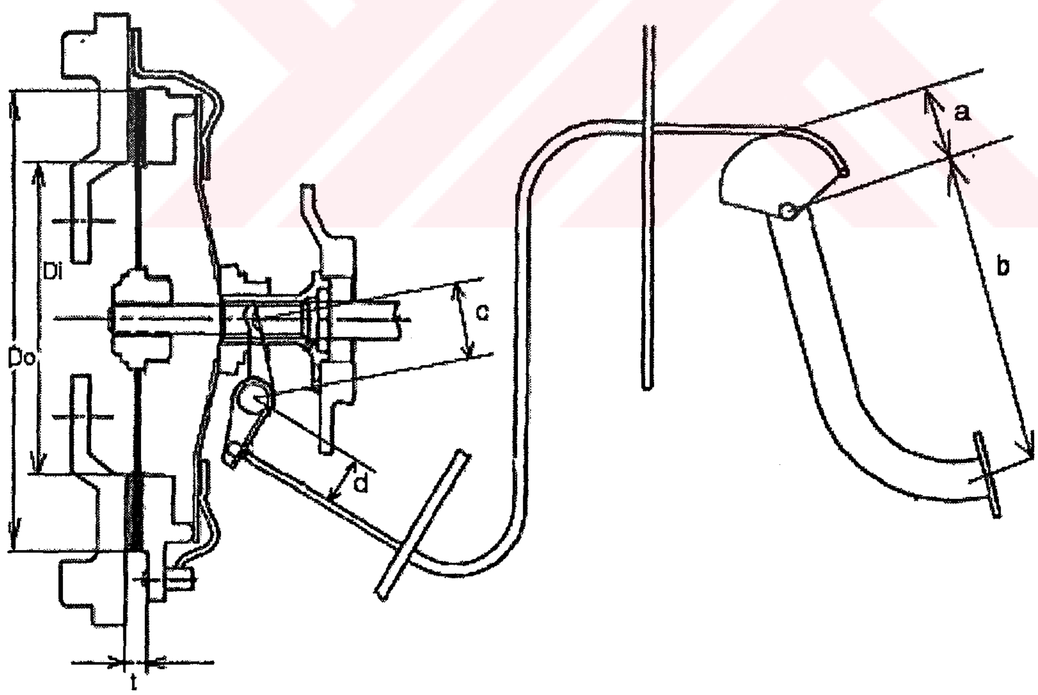


Fig 3.14. Direct Lever Mechanical Actuation System [20]

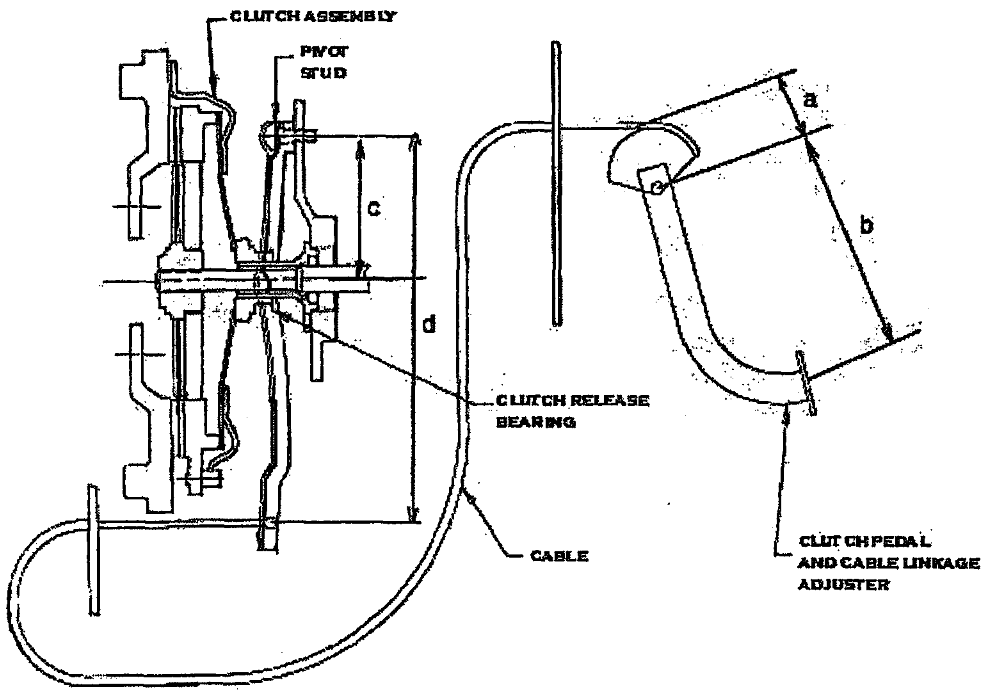


Fig 3.15. Pivoted Lever Mechanical Actuation System [20]

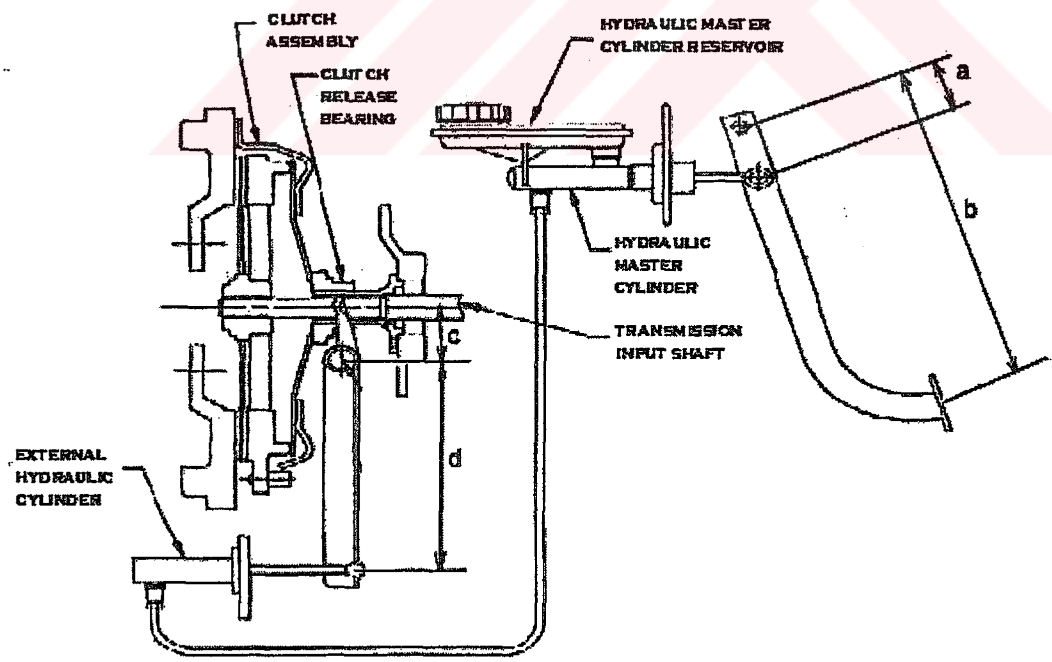


Fig 3.16 External Hydraulic Clutch Actuation System [20]

The comparison of the actual input and output values of force and travel yields the clutch actuation system force and travel ratios. These ratios include the system losses (both irreversible and reversible) like friction and deflection of the individual system components. A comparison of the actual system force and travel ratios to the theoretical ratios determines the force and travel efficiency of the clutch actuation system. In an ideal actuation system (no frictional losses or deflection), the theoretical and actual force and travel ratios would be the same.

As described above, the clutch actuation system experiences losses in efficiency due to friction and deflection of the individual components. These losses are experienced when the actuation system is used to disengage and engage the clutch. The operator experiences these losses as a lower pedal effort (force) and as an increased pedal return travel before engagement of the clutch occurs. This is known as system hysteresis and it is shown in Figure 3.17.

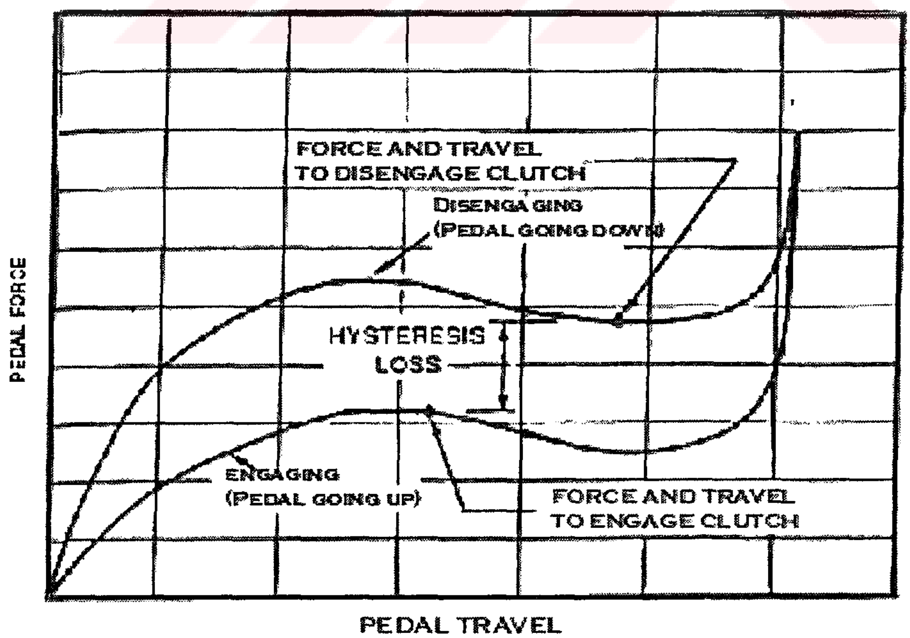


Fig 3.17. Hysteresis Loss in a Clutch System [20]

The design process should include this hysteresis effect. Since it is complex and it depends on the actuation system, the designer can use a multiplier for calculating the pedal force . The pedal force itself can be calculated by: [20]

$$PF = \frac{K(BearingLoad)}{leverratio1\ leverratio2} \tag{3.7.1}$$

Here the lever-ratio 1 (b/a) is the ratio of the lever at the pedal side and lever-ratio $2\ (c/d)$ is the ratio of the lever at the clutch side as shown in Fig 3.14. K is the hysteresis factor.

The specific form of each component in the actuation system is not considered in the calculations. Each system is the sum total of the individual components. The theoretical system ratio is computed by multiplying the individual component ratios $(C_{total} = C_{pedal}\ C_{cable}\ C_{fork}$ etc.$)$. In reality, there are two system ratios (ie, travel and force) due to the effects of deflection and friction. The actual system travel and force ratios are based on the known input travel and force compared to the measured output travel and force based on system components.

The travel and force efficiencies of the system can be calculated by comparing actual ratios to the theoretical system ratio. It should be noted that the mounting surfaces for individual components (i.e. clutch pedal bracket and clutch pedal bracket mounting surface, hydraulic master cylinder and external or concentric slave cylinder mounting surface, pivot stud mounting surface, etc.) may deflect under loading. However, these parts are considered to be rigid in this work.

CHAPTER 4

COMPUTER IMPLEMENTATION

A computer program is developed in Visual Basic 6.0 to operate on a PC that makes a non-linear clutch engagement analysis with the formulations used. Euler method is used as the method of solution.

The program is user friendly which is easy to use. It has some libraries to help users select the inputs. (Clutch sizes , engine characteristics etc.)

In this program:

-A vehicle model is developed including rotating masses, gear ratios, tire radius, and transmission efficiency.

-Engine characteristics are asked in a tabular form to determine the correct torque output from the engine for the corresponding engine speed.

-Clutch motion and throttle level are the operator dependent input variables and they are fitted into scererios that the user can easily define.

-A thermal analysis is made to see the interface and bulk temperatures throughout the engagement process.

-Diaphragm spring analysis is made to determine the clamp load and bearing force on the clutch.

-The actuation system is taken into consideration basicly.

-A friction coefficient analysis is made to fit alternating conditions.

-The outputs are taken out in a graphical form to show the effects of variation of the inputs.

4.1 Flowchart of the program

A detailed flowchart of the program is given in the Appendix A. The program inputs are arranged in 5 different groups of inputs. These are friction disc and clutch actuation system inputs, diaphragm spring inputs, vehicle parameters inputs, engine torque characteristics inputs and driver scenario inputs. The parameters are calculated as time dependent and the outputs are also given as time dependent.

4.2 Description of the Computer Program and the Displays

4.2.1. Clutch Disc and Actuation System Input Screen

The clutch disc and actuation system parameters are entered from the first input window of the program shown in Figure 4.1. The friction disc diameters, thickness and initial temperature is asked in this window. A complete diaphragm spring clutch and the actuation system are shown in the first window of the program. The lever ratios of the actuation system are indicated and the hysteresis coefficient is asked. The hysteresis coefficient is explained in detail in section 3.8. If a hydraulic actuating system is going to be used then the lever ratio at the pedal side must be considered as the hydraulic cylinder diameter ratio. If repeated engagements are considered for the temperature analysis, then the number of repeated engagements and time interval between these repetitive engagements must be entered. If only one engagement is going to be analysed then these spaces must be left as “1”.

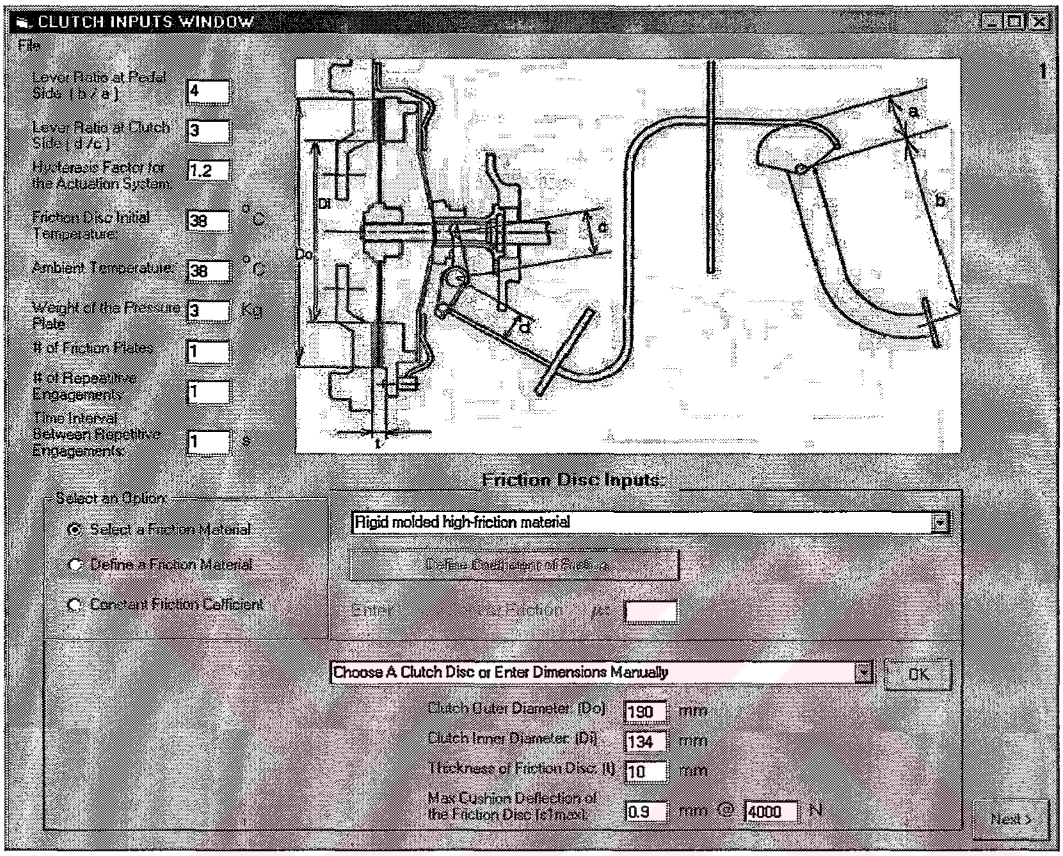


Fig 4.1 Clutch Disc and Actuation System Input Screen

Below the figure there are friction disc inputs. The friction material is selected in this window. The user can select one of the most commonly used five friction materials listed in the listbox, define a friction material or enter a constant friction coefficient. For friction materials used in clutches, the variations of the coefficient of friction with temperature are given in Appendix C. If a constant friction coefficient is entered, the program will get this value and use it throughout the analysis. The friction disc dimensions and cushion specifications can either be entered by selecting one of the friction discs manufactured by

Valeo® or by entering manually. Some of the Valeo® clutches used in passenger cars are given in Appendix E.

By the "File" menu, user can save the inputs, open the previously saved inputs and print all inputs. "Save" command includes all the inputs so after filling all the inputs, user should view this screen and save the inputs.

4.2.2 Diaphragm Spring Input Window

The second input window asks the user to enter the diaphragm spring parameters shown on the window. The user can select one of the Valeo® springs or enter the dimensions of the diaphragm spring manually. If one of the springs in the list is selected, the user should press on the "OK" button and see the new values of the variables in the white blanks before going on to the next step. The diaphragm spring input parameters are shown in Figure 4.2.

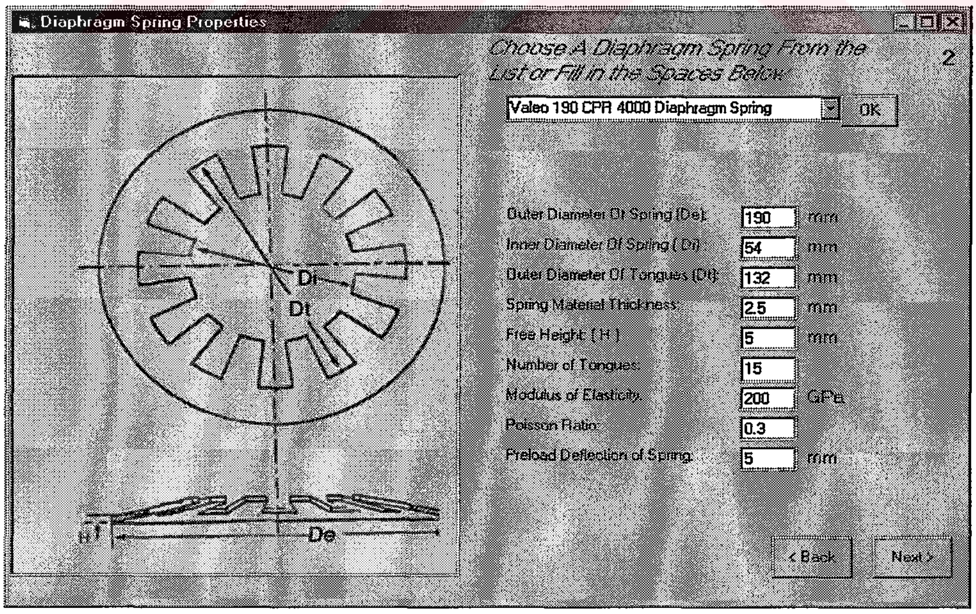


Fig 4.2 Diaphragm Spring Input Screen

Since the clamping force is directly related on the dimensions of the diaphragm spring, all the dimensions should be entered carefully. Preload deflection of the spring is the initial loading deflection of the spring, which corresponds to full release deflection from the free position of the spring.

4.2.3 Vehicle Parameters Input Window

This screen shows a basic vehicle model used in the program. The necessary vehicle parameters such as gear ratios, wheel radius, vehicle weight are entered in this window. The drive-line components, which have different rotational speeds are represented with different axis in the Figure 4.3. Clutch and gearbox input shaft is shown on the left of the gearbox. Gearbox output shaft, propeller shaft and the differential input gear are shown on the left of the gearbox. The differential output gears, axes and tires are represented with the vertical axis on the right. The moment of inertias of these transmission parts should be entered individually. Also the moment of inertia of the engine (including the flywheel) should be entered in this window.

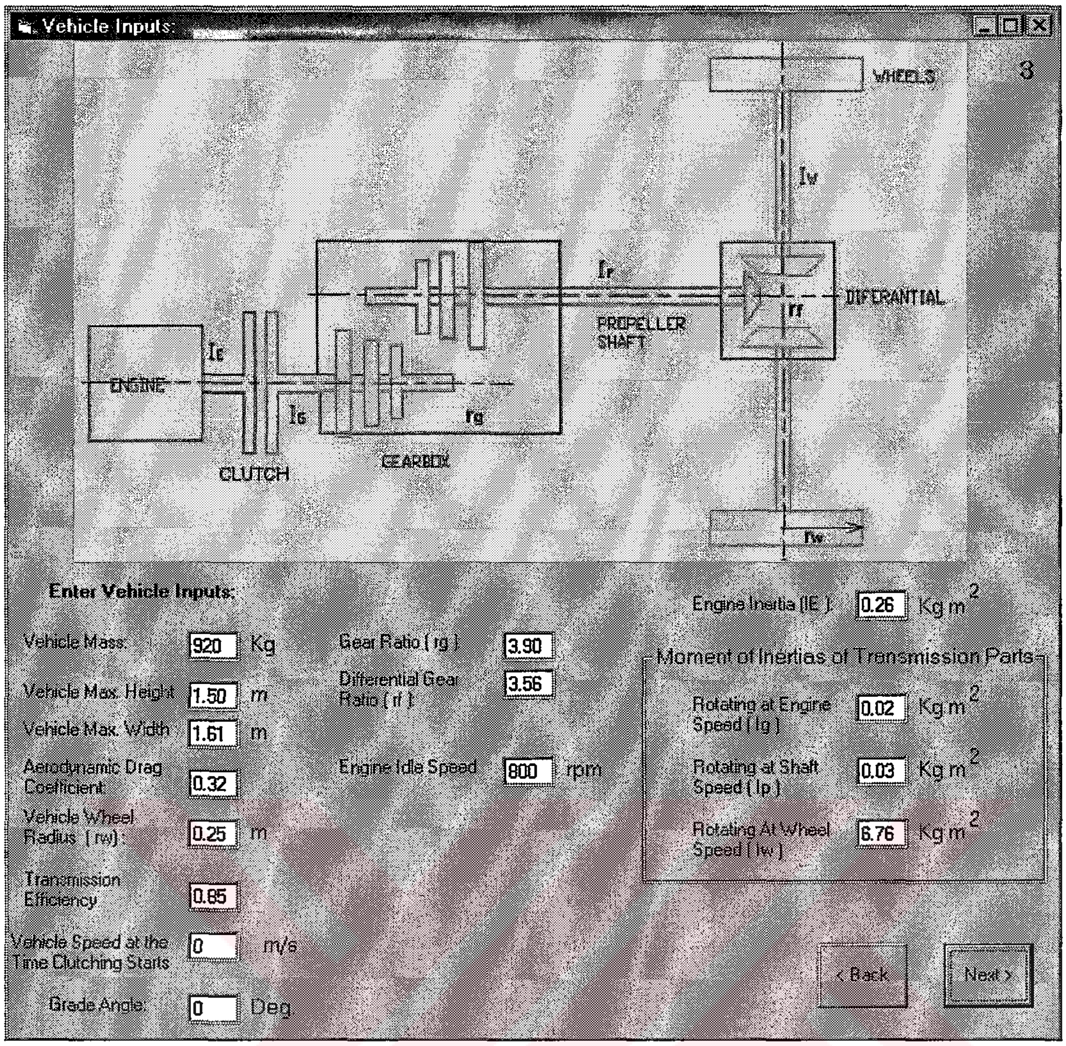


Fig 4.3 Vehicle Parameters Input Screen

The gearbox ratio entered here can be any gear ratio of the vehicle to be analysed. With this program not only the start-up condition but also the moving vehicle condition can be analysed. If the vehicle is in motion during gear shifting, the vehicle speed at this instant should be entered in this window. The road that the analysis is going to be made can either be a level road or a road with gradient. If a level road is considered, the grade angle should be left as "0". Otherwise the grade angle should be entered in degrees. After all the necessary vehicle inputs are entered, the user can go on to the next step by pressing the button on the bottom left corner.

4.2.4 Engine Torque Characteristics Input Screen

In order to get the engine torque depending on the rotational speed of the engine and the throttle level (which is adjusted by the driver), a detailed engine torque characteristic analysis must be made. For that reason a part throttle input screen is added to the program.

The user fills the white blanks according to the engines torque characteristics. Program enables the user, to enter maximum 8 different throttle level characteristics. For every different throttle level, the user must enter at least 5 torque values of the engine with the corresponding rotational speeds in order to generate a sufficient torque map of the engine. In the screenshot given in Figure 4.4, a small size gasoline engine part throttle characteristics are shown. For 100 % throttle (Full Throttle) The engine gives 80 Nm torque @ 1000 rpm, 105 Nm torque @ 4000 rpm and 92 Nm torque @6000 rpm. The program fits a curve for the entered values and the user can see the torque-rpm graph by entering the throttle level to the "Wanted Throttle" space and press on the "Draw" button. The user should enter at least 2 different throttle characteristics in this window.

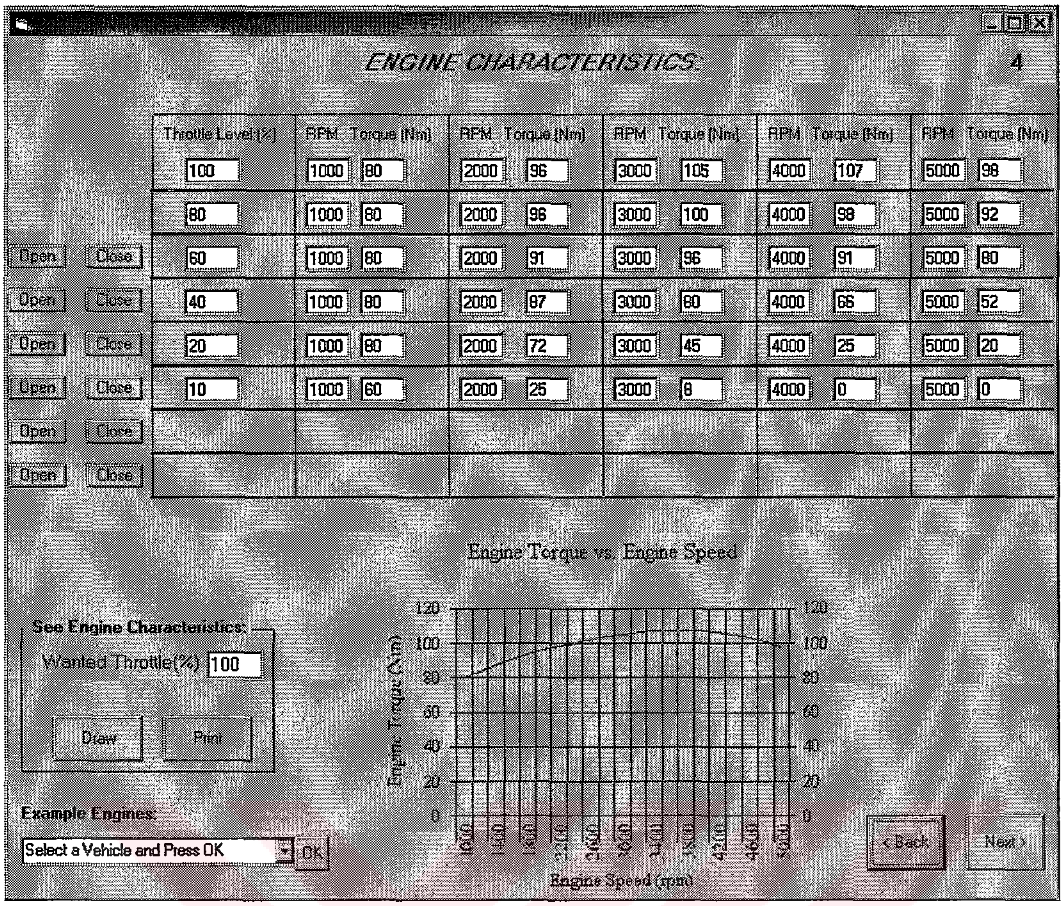


Fig 4.4 Engine Torque Characteristics Input Window

If the torque characteristics at only two different throttle levels are available, then the other spaces should be closed with the close button near them. If the torque characteristics at eight throttle levels are available then all the spaces should be open and filled. After filling all the available torque characteristics (Six data sets is available in Figure 4.4), the user can enter any throttle value from 1 to 100 to the "Wanted Throttle" space and see the fitted curve by pressing on the "Draw" button. The program first curve fits the available data (If four set of data is given then four different curves are fitted) by "Newton's Interpolating Polynomials" method. After that, the program makes linear interpolation between these curves to calculate the data between these curves. For example in the Figure

4.4, 100 % and 70 % throttle values are entered by the user. When the program needs the torque at 85% throttle and 1500 rpm, the torque values at 100% throttle and 80% throttle at 1500 rpm are calculated. Then the program makes an interpolation between these values to get the torque value at 85% throttle and 1500 rpm.

4.2.5 Operator Input Screen

This window enables the user to simulate drivers foot motions for the throttle level and the clutch release. Previous statistical works [5], [10] have shown that the driver applies sufficient clamp load to hold the vehicle and this is held constant throughout the subsequent engagement until the slip speed is nearly zero. Then the rest of the clamp load is applied rapidly. This is far from the application of the full clamp load in zero time as it is often used in the mathematical models in the literature.

The length of time in which the vehicle is held on the clutch can be of 0.5 s on the level road and more than 1 s on a road with steep gradient. It is important to take the heat generated and the temperatures into account when simulating clutch engagement mathematically. The simulation in Figure 4.5 includes this holding period, which can be adjusted by the user. The clutch release behaviour of the operator is defined with 3 different curves. Clutch release scenario number 2 is linear increasing while 1 and 3 are exponential increasing. The power factors asked, determines the shape of the increase.

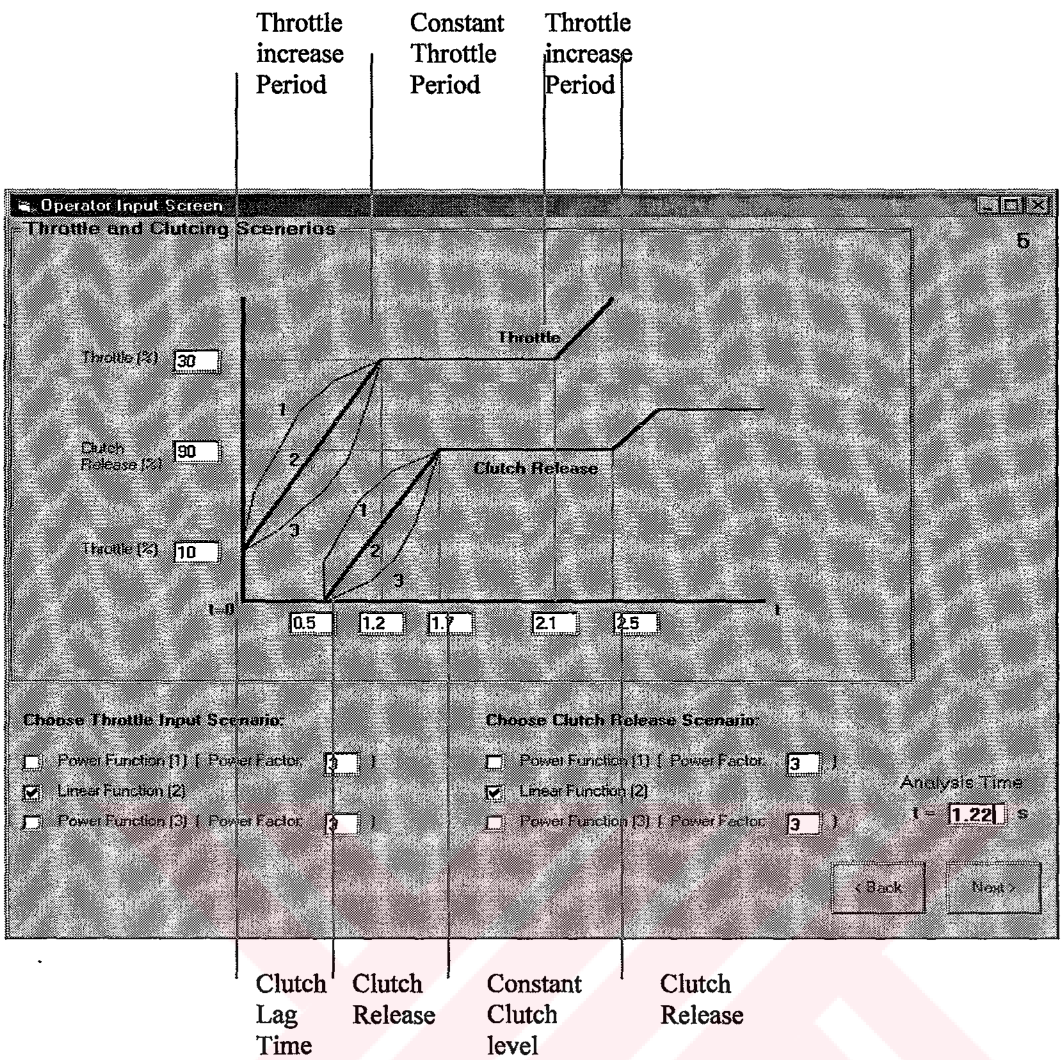
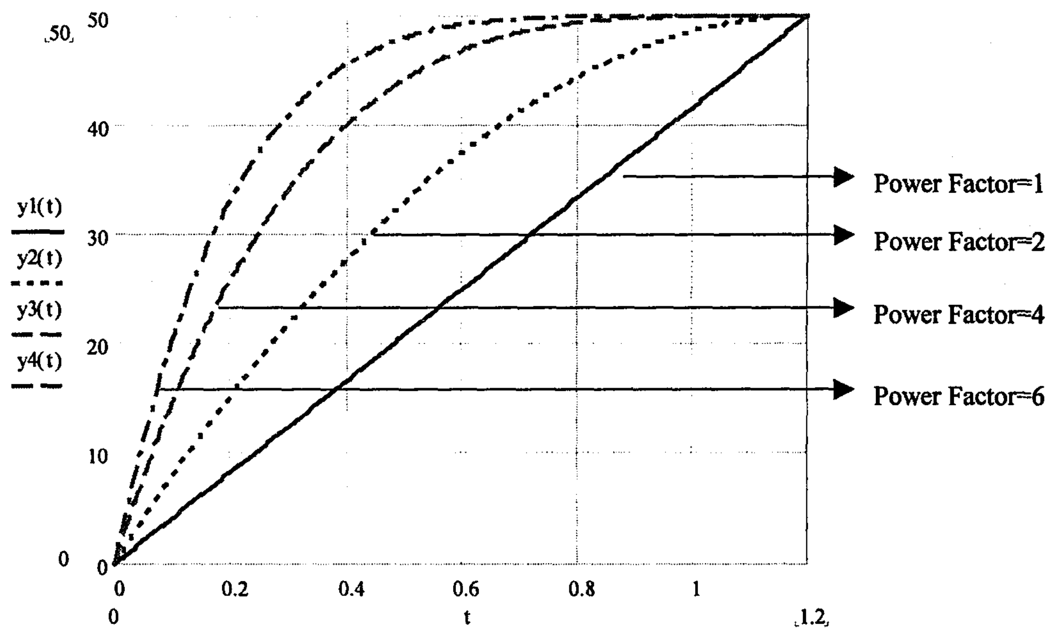


Fig 4.5 Operator Inputs Screen

Figure 4.6 shows behaviour of throttle level and clutch release increase for different power factors. The curves in Figure 4.6 are obtained by using equations 4.1 and 4.2. Here the first figure shows the power function (1) and second figure shows the power function (3).

$$y(t) = CR_{Max} - CR_{Max}.(\frac{t2-t}{t2})^{Power.Factor} \tag{4.1}$$

$$y(t) = CR_{Max}.(\frac{t}{t2})^{Power.Factor} \tag{4.2}$$



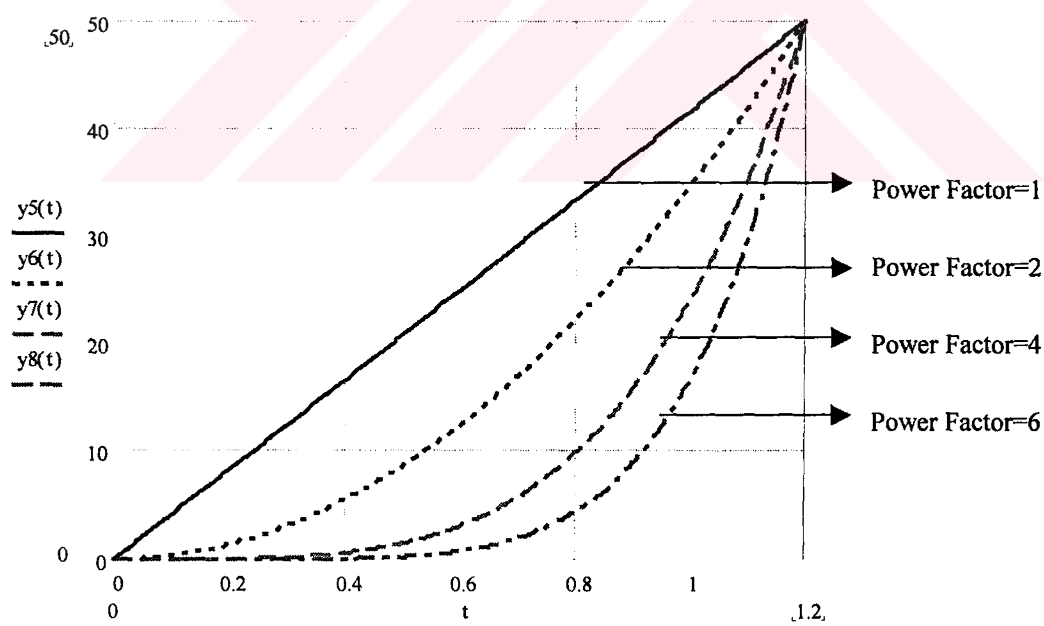


Fig 4.6 Effect of Power Factor on Throttle and Clutching Scenarios

In equations 4.1 and 4.2, t_2 is the time at the end of clutch release period and specified as 1.2 s in Figure 4.6. CR_{Max} is the maximum clutch release percentage specified.

The same input style is used for the throttle input, which allows user to enter most of the real life scenarios. There is also a holding period for the throttle input scenario. The user can lengthen the first part (increase part) if no holding period wanted. If always constant throttle is wanted then both the throttle values (minimum and holding) can be made same as the wanted constant throttle.

In the scenario given in Figure 4.5, the operator starts to press the accelerator from the minimum throttle level of the vehicle (idle speed) to 30 % throttle level in the first 1.2 seconds. In the first 0.5 seconds there is no clutch action and the engine accelerates. Between 1.2 and 1.7 seconds the clutch is released by 90% linearly. (2nd scenario input for the throttle and clutch is selected) After 1.7 s the clutch is held at this level (90 %) and the throttle is still at 30 % . After 2.1 s the throttle is depressed linearly to 100 % with the same slope of the first part. This last part of the throttle scenario is added to see the vehicle motion after the engagement. Generally it happens after the engagement. But it can also be added in the clutching action by shortening the holding period. This can give opportunities to the user to create different scenarios.

"Analysis time" input space enables the user to make the analysis from t=0 to the specified time. If this time value is larger than the engagement period than the user can see the vehicle variables (velocity, acceleration etc.) after the engagement.

4.2.6 Output Parameters Screen

This screen shows the outputs taken by the program at the specified analysing time. This time can be changed also in this screen. By pressing the refresh button, the outputs at the specified time can be seen. User can print both the inputs and the outputs in this window. Some of these outputs seen are time dependent and their variation with time can be viewed by going to the "Charts" window by pressing on the button on the bottom left corner.

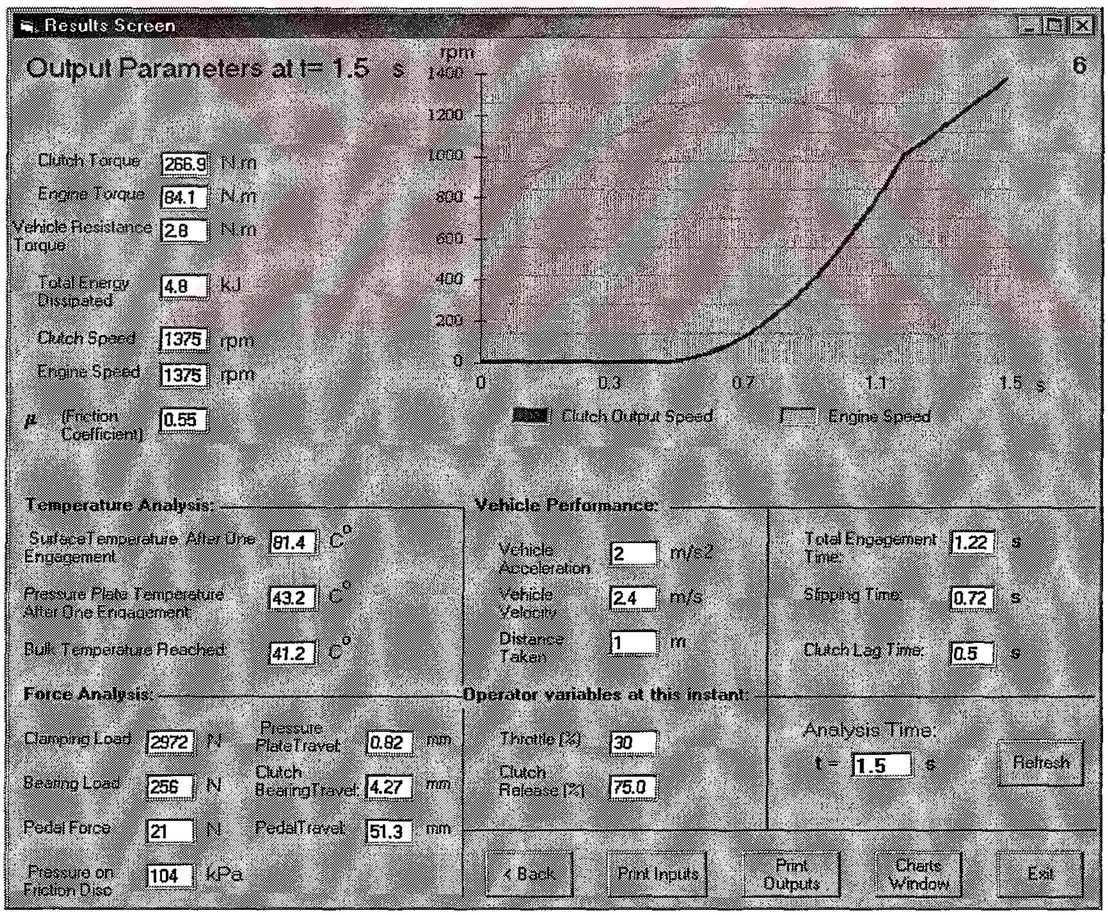


Fig 4.7 Output Parameters Screen

In the Figure 4.7, the engagement is completed and the vehicle continues to accelerate. Engine speed and clutch speed are both same. This means the contact has ceased and they are rotating together. From the operator inputs at 1.5 *s* the clutch release is at 75 % and the corresponding clutch friction torque provided by the diaphragm spring selected is 267 Nm. Engine torque at 1375 rpm is 84.1 Nm at 30 % throttle. Friction coefficient just before the engagement is 0.55.

From the temperature analysis section, clutch disc interface temperature is reached to 81 °C from 38 °C at 1.2 seconds (The end of engagement). The clamping load on the pressure plate exerted by the diaphragm spring at 75% release by the operator is 2972 N and this force is controlled by 256 N clutch bearing force. With the specified lever ratios of the actuation system, the operator needs 21 N force to control the bearing and thus the pressure plate. The clamping load on the pressure disc creates 104 kPa pressure on the clutch disc.

The specified vehicle's acceleration is 2.0 m/s^2, and the speed at 1.5 s is 2.4 m/s. Distance covered by the vehicle in 1.5 s is 1.0 m (No movement in the first 0.5 seconds).

By the "Print Inputs" and "Print outputs" buttons, user can print all of the inputs entered in previous windows and the outputs calculated.

4.2.7 Charts Window:

This Window includes some buttons that has the description of the chart it includes. Related charts are combined in the same screen so the user can see the relations between the parameters. All of the charts can be printed by the print button near the chart. The detailed explanation of these charts will be given in the case studies section.

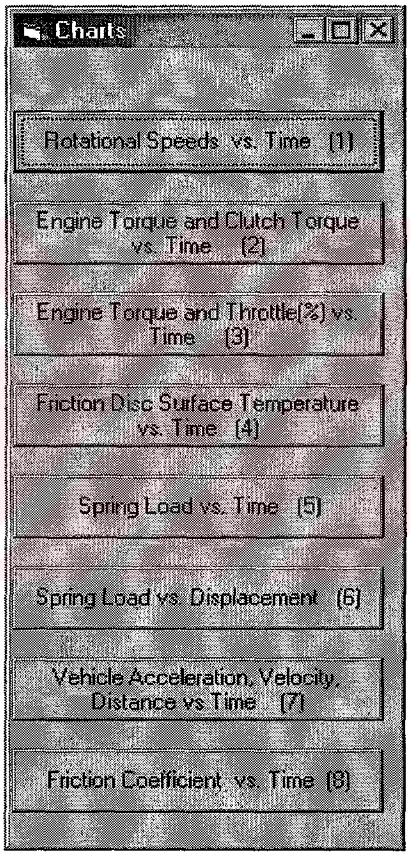


Fig 4.8 Charts Window

CHAPTER 5

CASE STUDIES

In this chapter a number of sample applications of the program will be given in order to compare the effects of different parameters on the clutching performance of a vehicle. The vehicle parameters, clutch parameters and the road conditions are defined. Results are given in terms of vehicle performance parameters. The input and output parameters are given in tables and charts, discussions are made.

Vehicle and Clutch Parameters

The vehicle to be analysed is a small passenger car which has a 845 kg empty weight. The parameters of the vehicle considered are:

Vehicle Mass:	845 kg
Vehicle wheel radius:	0.250 m
Vehicle Height:	1.501 m
Vehicle Width:	1.610 m
Drag Coefficient:	0.32
Gearbox Ratio:	3.90
Final Drive Ratio:	3.56
Transmission Efficiency:	0.85
Engine Idle Speed:	800 rpm
Moment of Inertia of Engine:	0.28 $kg.m^2$
Moment of Inertia of Transmission	
Rotating at Engine Speed:	0.02 $kg.m^2$
Rotating at Shaft Speed:	0.03 $kg.m^2$
Rotating at Wheel Speed:	6.76 $kg.m^2$

The clutch disc and corresponding clutch diaphragm spring to be used for this vehicle is listed below.

Friction Disc:

Friction Disc Model:	Valeo 190 K22 AX/408
Friction Material:	Rigid molded high-friction material
Outer Diameter of Clutch Disc:	190 mm
Outer Diameter of Diaphragm Spring:	134 mm
Friction Disc Thickness	10 mm
Max Cushion Deflection of Friction Disc	0.9 mm @ 4000 N
Initial Temperature of Friction Disc	38 °C
Ambient Temperature	38 °C
Lever ratio of actuation system at pedal side:	4
Lever Ratio of actuation system at clutch side:	3

Diaphragm Spring:

Diaphragm Spring:	Valeo 190 CPR 4000 Diaphragm Spring
Outer Diameter of Diaphragm Spring:	190
Inner Diameter of Diaphragm Spring:	54 mm
Outer Diameter of Tongues:	132 mm
Spring Material Thickness:	2.5 mm
Spring Free Height:	5 mm
Number of Tongues	15
Modulus of Elasticity:	200 GPa
Poisson's Ratio:	0.3
Preload Deflection	4 mm

These values are entered in the first three input windows (ie. Clutch Input Window, Diaphragm Spring Input Window and Vehicle Inputs Window) of the computer program

The engine torque map of the vehicle can be entered in the engine characteristics window. In Figures 5.1 and 5.2, the engine torque characteristics of the vehicle and its application in the computer program is given.

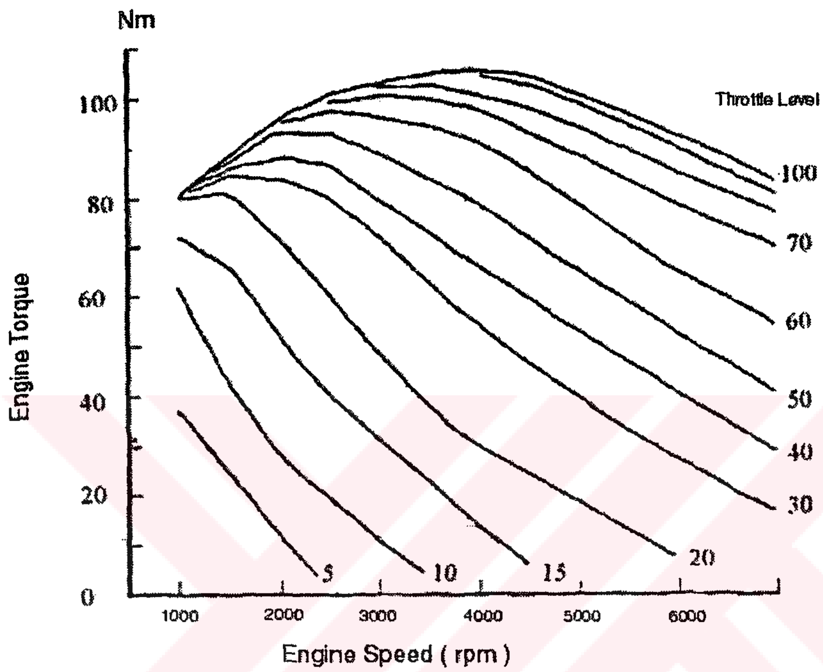


Fig 5.1 Actual Torque Map of The Engine

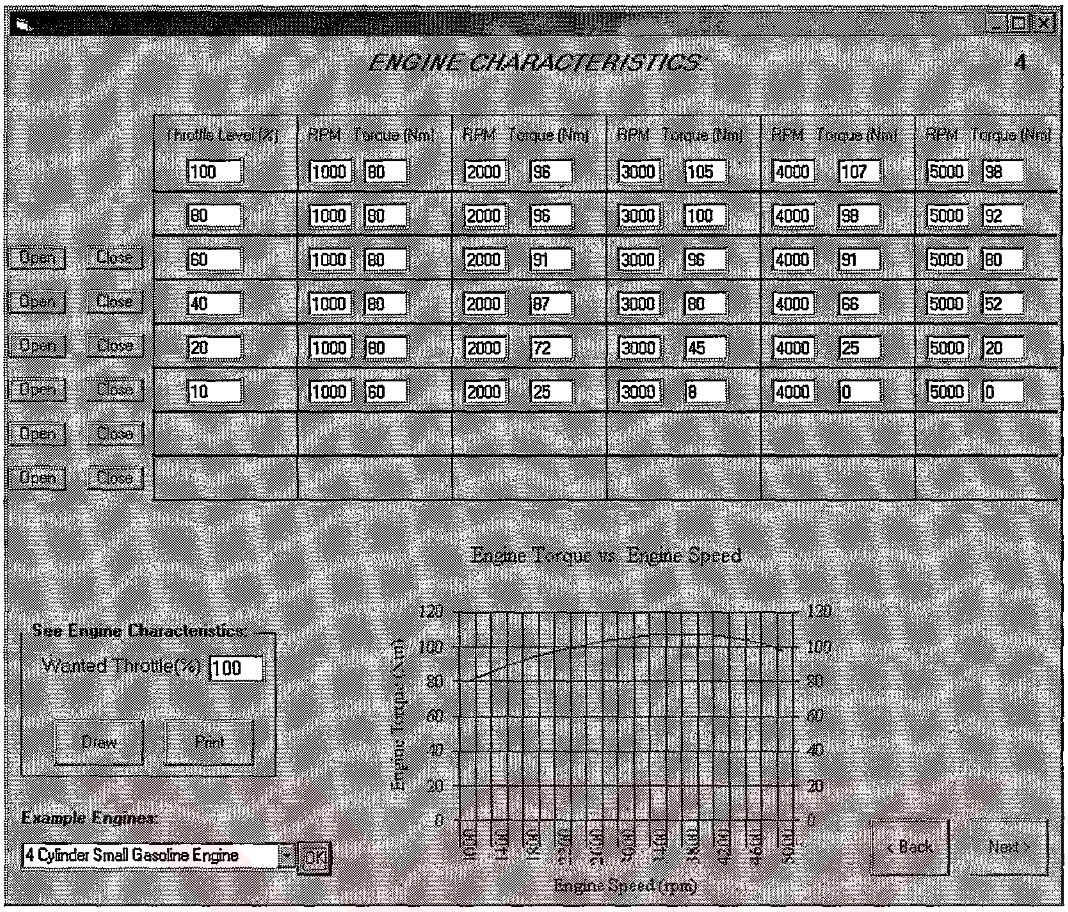


Fig 5.2 Engine Torque Characteristics Input Window

Case Study 1: A Short Slipping Time Engagement

In the analysis, slip period starts when the contact between the clutch and flywheel starts and ends when the clutch and the flywheel rotate at the same speed. Engagement period is the period which starts at $t = 0$ and ends when the clutch fully engages the flywheel. Engagement period consists the initial lag time.

The first analysis will be done on level road and with only the driver inside the vehicle. An average driver weight of 75 kg will be used in the analysis so the total vehicle weight is 920 kg. In case studies 1 and 2, the effect of clutch release period on the vehicle performance is investigated by considering a typical throttle scenario and a 0.5 s clutch engagement lag period. The drivers behaviour is entered by the operator input screen given in Figure 5.3.

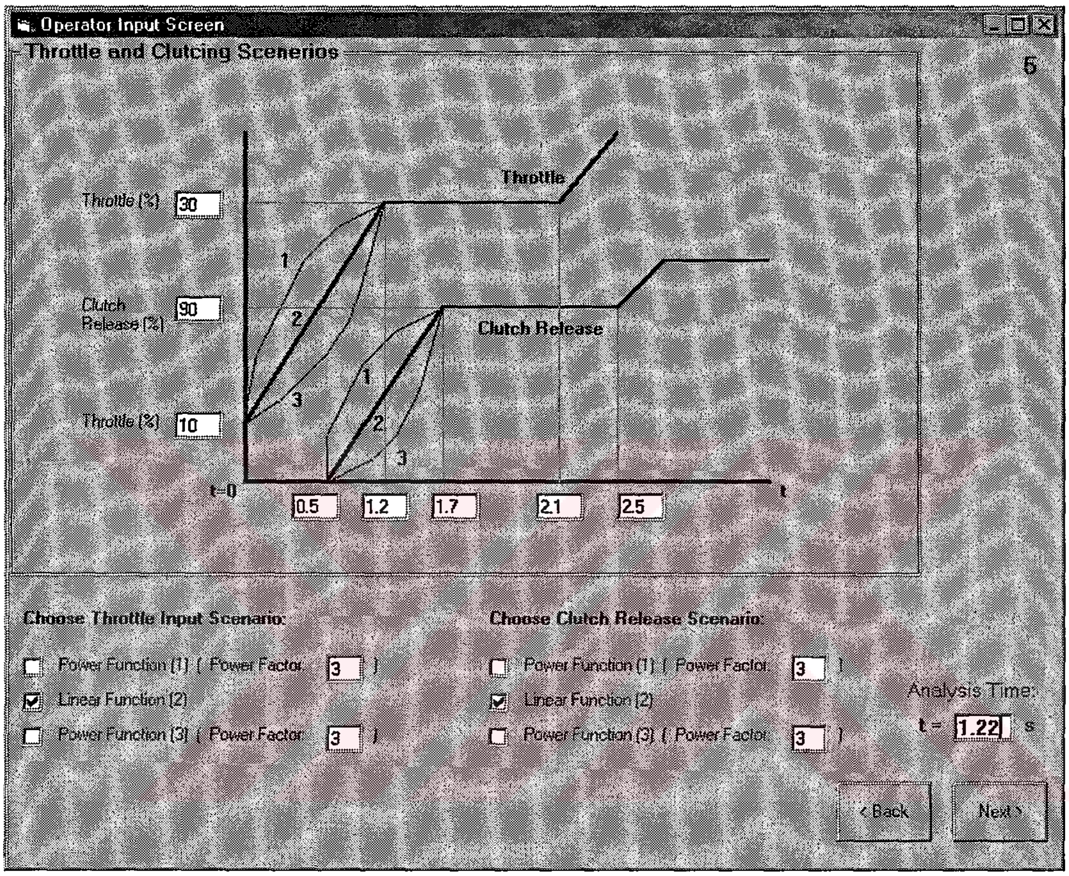


Fig 5.3 The First Driver Input Scenario for Case Study 1

First a linear increase in both the throttle level and clutch release will be used as given in Figure 5.3. In the first 0.5 seconds there is no clutch action but the engine accelerates by the linearly increasing throttle level. Between 0.5 and 1.7 seconds, the clutch is released by 90% and the throttle level increases until 1.2 seconds then remains constant at 30 % level. In this case the simulation results obtained from the program is given in Figure 5.4.

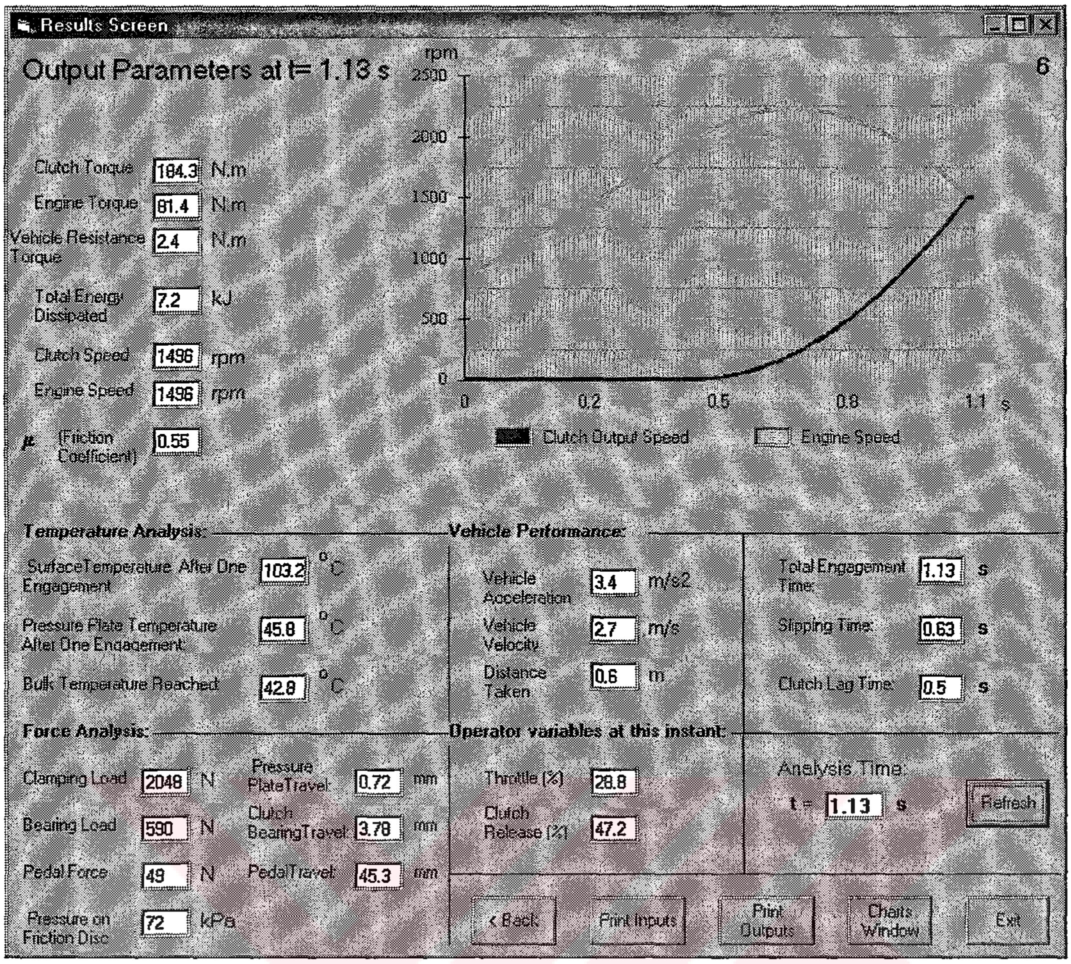


Fig 5.4 Output window of Case Study 1

The clutch output speed starts to increase at 0.5 *s* where the engine speed increased up to about 2000 rpm. The clutch engagement ends when the throttle level reaches to 30 % constant level. The engagement takes about 1.17 seconds with the slip period of 0.63 seconds. 7.2 kJ energy is dissipated by the clutch during the slip period. The clutch torque at the end of clutch engagement is 184 Nm corresponding to 47 % clutch release by the operator which means that engagement ceases before the full release of the clutch. The throttle is at 29 % and the clamping force on the friction disc at 47 % release is 2048 N. The bearing load to control this clamping force is 590 N and the pedal force to control this bearing load is 49 N.

The vehicle performance during clutch engagement period is shown in Figure 5.5 in terms of vehicle acceleration, velocity and distance versus time graphs.

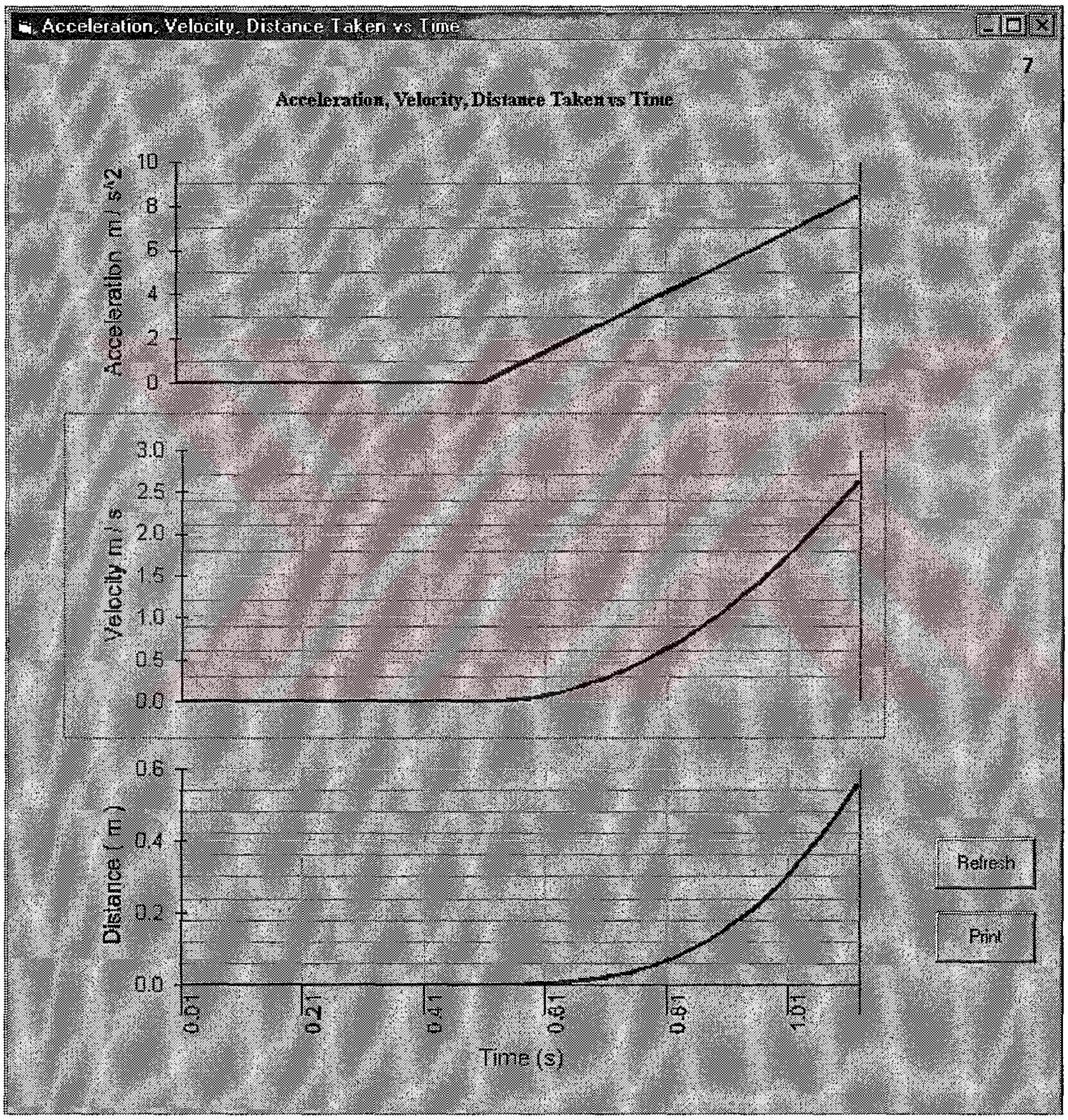


Fig 5.5 Acceleration, Velocity and Distance Graphs for the Case Study 1

This graph here is given until the end of engagement period. But after the engagement can also be viewed by setting the Analysis time longer than the total engagement time. As shown in Figure 5.5, in the first 0.5 seconds there is no movement as expected. After the engagement starts, the vehicle accelerates and takes a distance of 0.6 m in 0.63 seconds.

The Figure 5.6 shows the engine torque and clutch torque variations during clutch engagement. The clutch torque is increasing almost linearly by the effect of linear clutch release input and diaphragm spring force characteristics. The engine torque variation in the graph is interesting but the reason is the decreasing torques with the increasing rpm (in low throttle level situations).

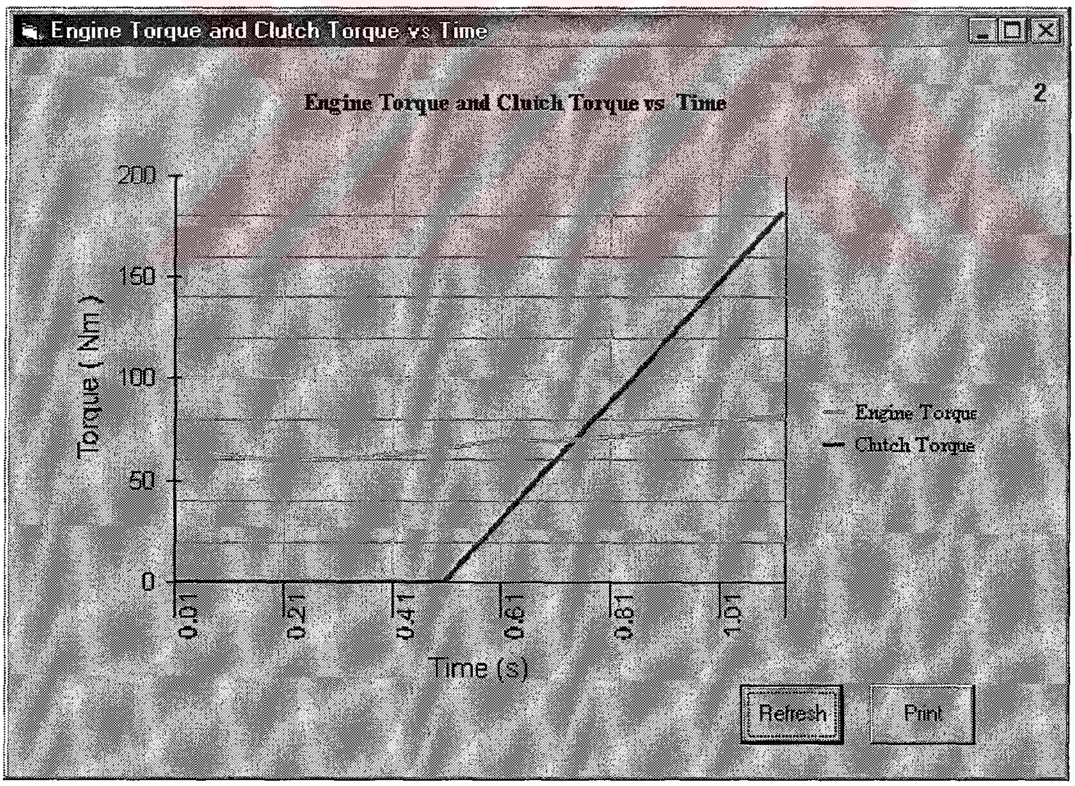


Fig 5.6 Engine and Clutch Torque Variation with Time in Case Study 1

Case Study 2: A Long Slipping Time Engagement

A low clutch release rate engagement situation will be analysed in this scenario for the same vehicle and same conditions. As it is seen in Figure 5.7, the engagement starts at 0.5 s as it was in the previous example. But the clutch release time is increased to 3.5 s to get a slower engagement. Other variables in the scenario are the same as given in Case Study 1.

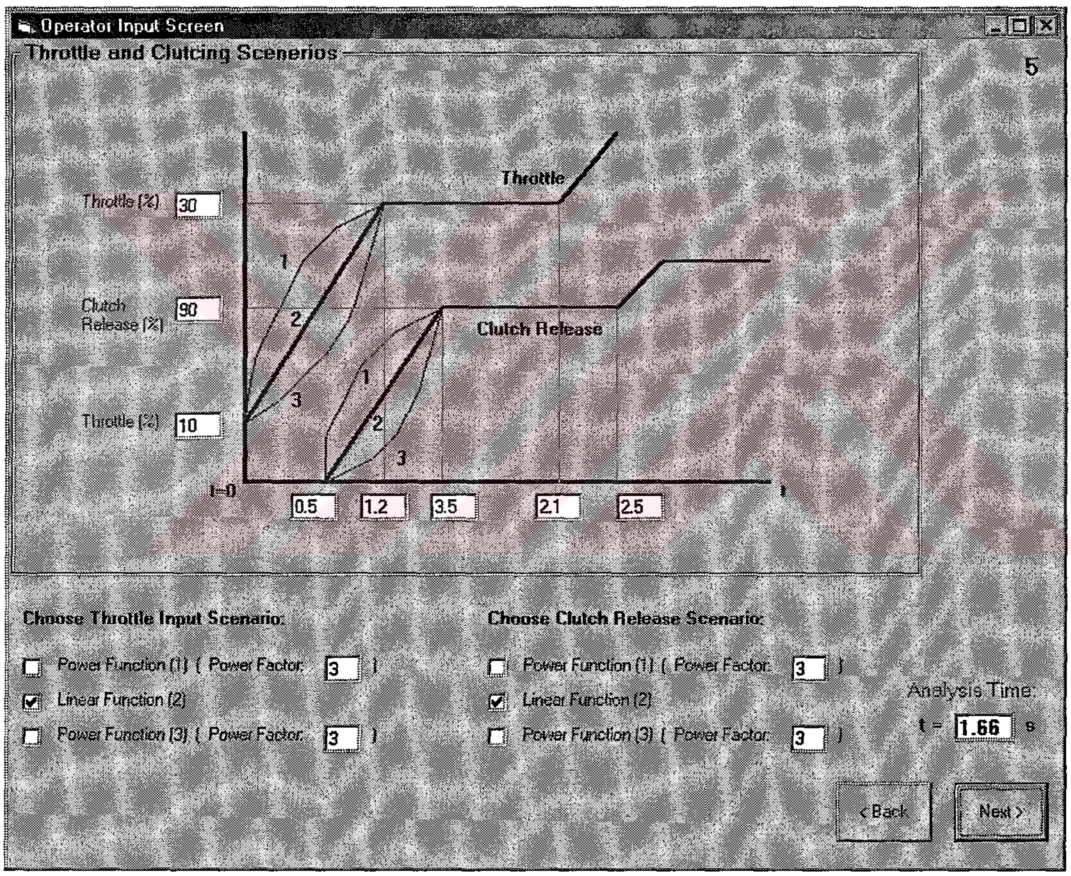


Fig 5.7 The Driver Input Scenario of Case Study 2

The engagement is completed at 1.66 s at 2040 rpm of engine speed. The engine speed is increased until 2700 rpm and the slow clutch release decreases the engine speed to 2040 rpm when the engagement is completed. In this longer engagement period 11.6 kJ energy is dissipated (61 % more compared with the scenario given in Case Study 1). Friction surface temperature is 143 °C after the engagement which is again higher when compared with the Case Study 1. The output screen at this situation is given in Figure 5.8.

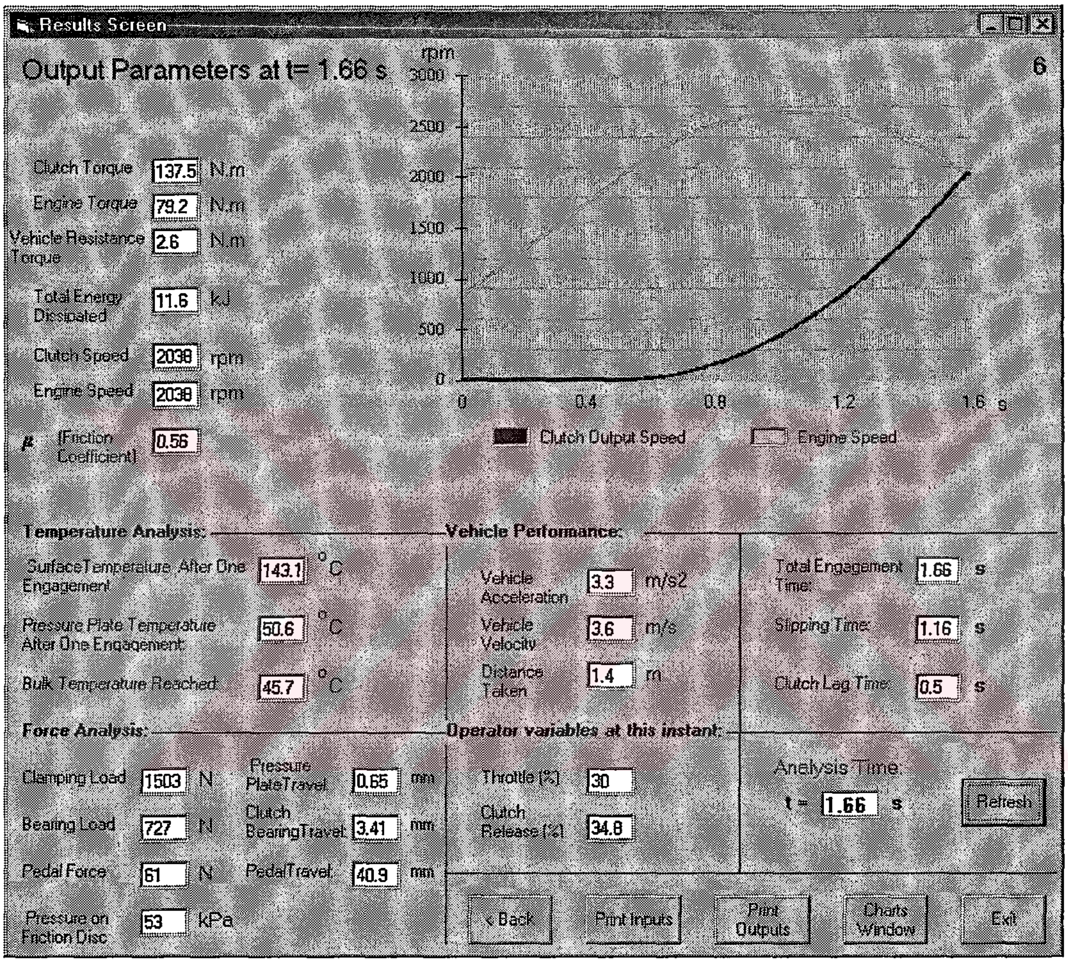


Fig 5.8 Output Screen of Case Study 2

The vehicle performance during the engagement can be seen in Figure 5.9. All variables are higher in this case but the time passed is also higher. In order to compare both cases 1.66 s analysis should be made in the first case. When the analysis is made, it is seen that 1.4 m of distance is covered by the vehicle during short engaging period (In long engagement case it is about 0.6 m). In Figure 5.10, the 1.66 s analysis for Case Study 1 is given in order to compare output parameters of Case Study 1 and Case Study 2 for the same analysis time.

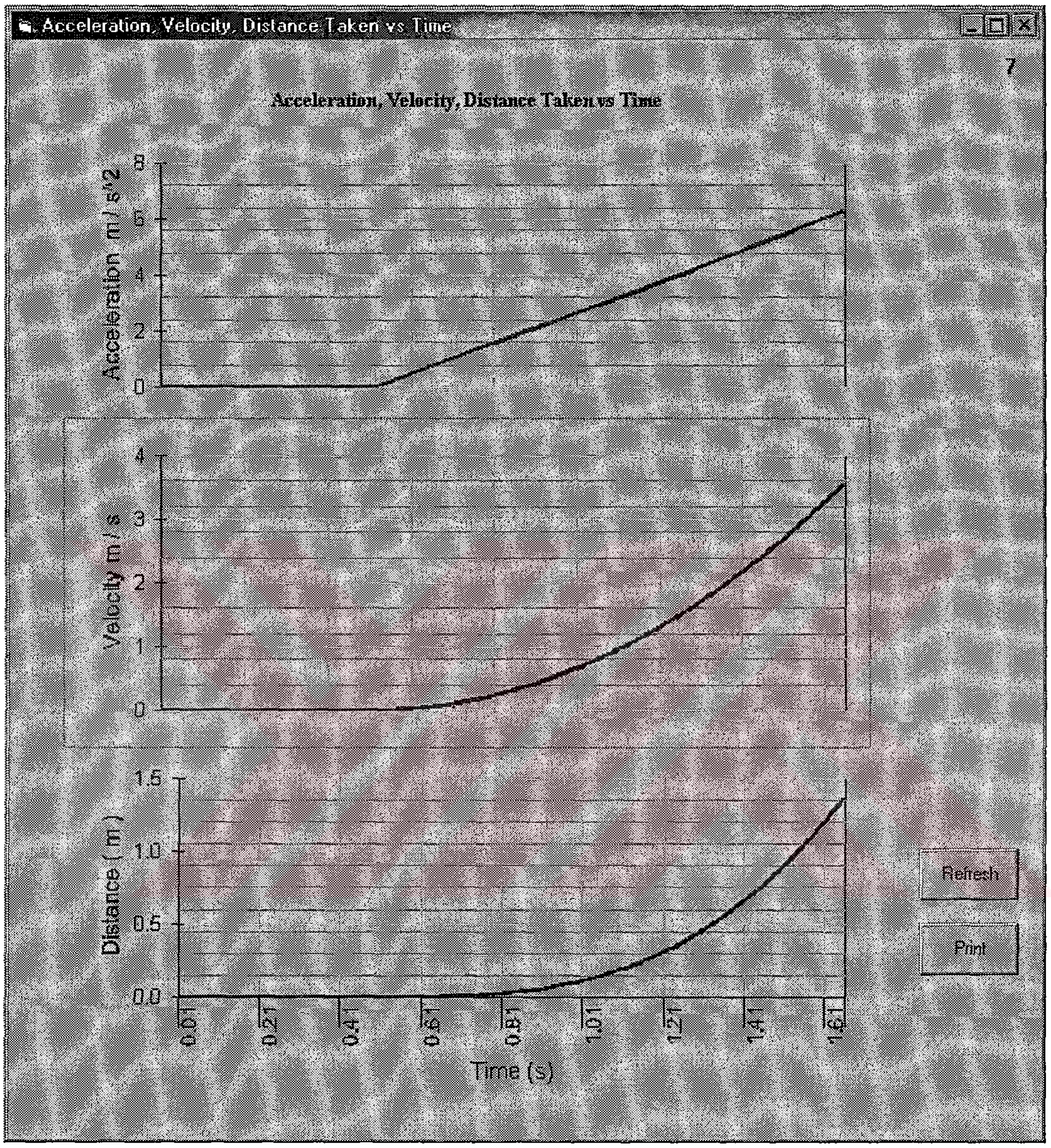


Fig 5.9 Vehicle Performance for Case Study 2

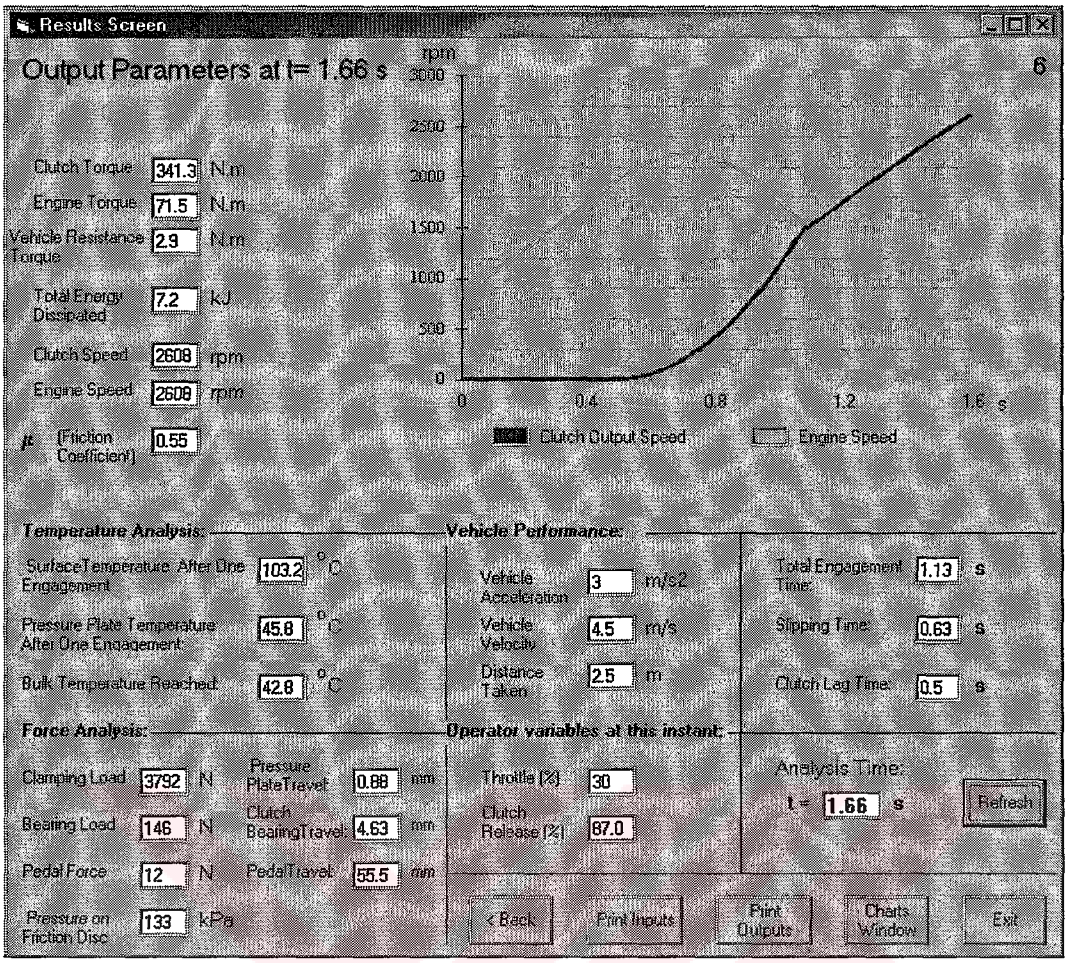


Fig 5.10 Output Screen for Case Study 1 for 1.66 s

The engine torque increases slowly by the effect of low throttle characteristics of the engine. The variation of the engine torque and the clutch torque is shown in Figure 5.11. The clutch torque increases almost linearly by the effect of the selected scenario and the diaphragm spring properties. The clutch disc surface temperature increase versus time is given in Figure 5.12.

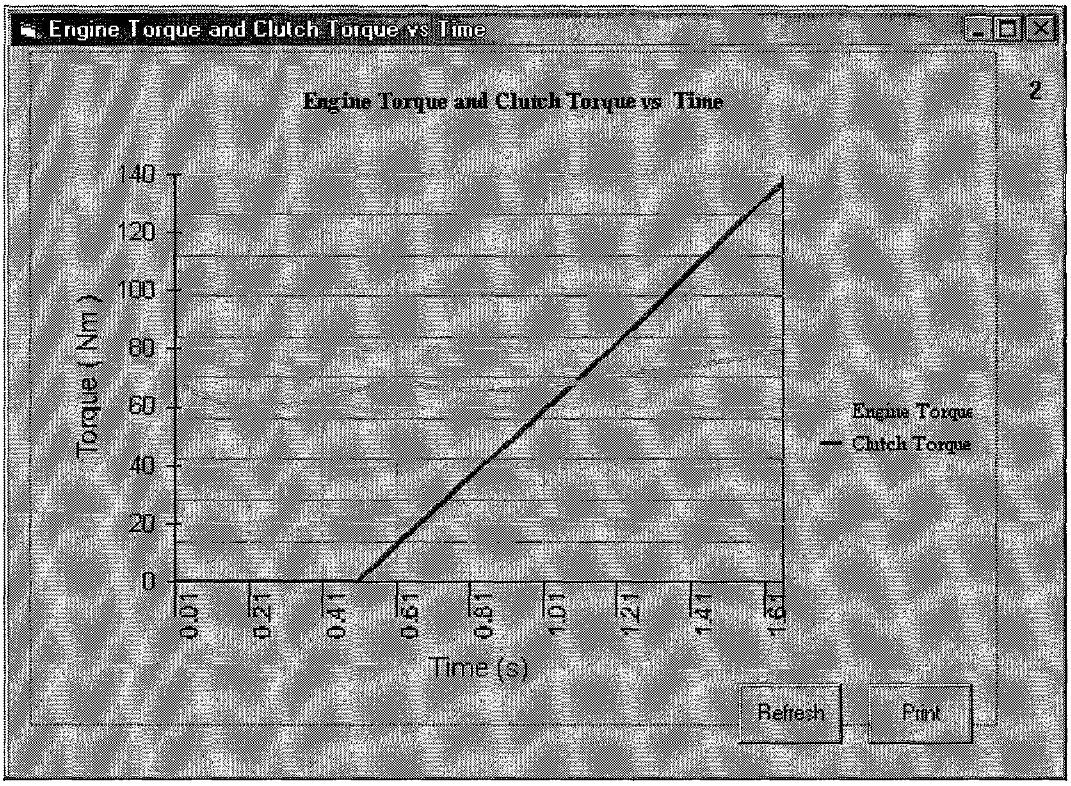


Fig 5.11 Engine and Clutch Torque Variation for Case Study 2

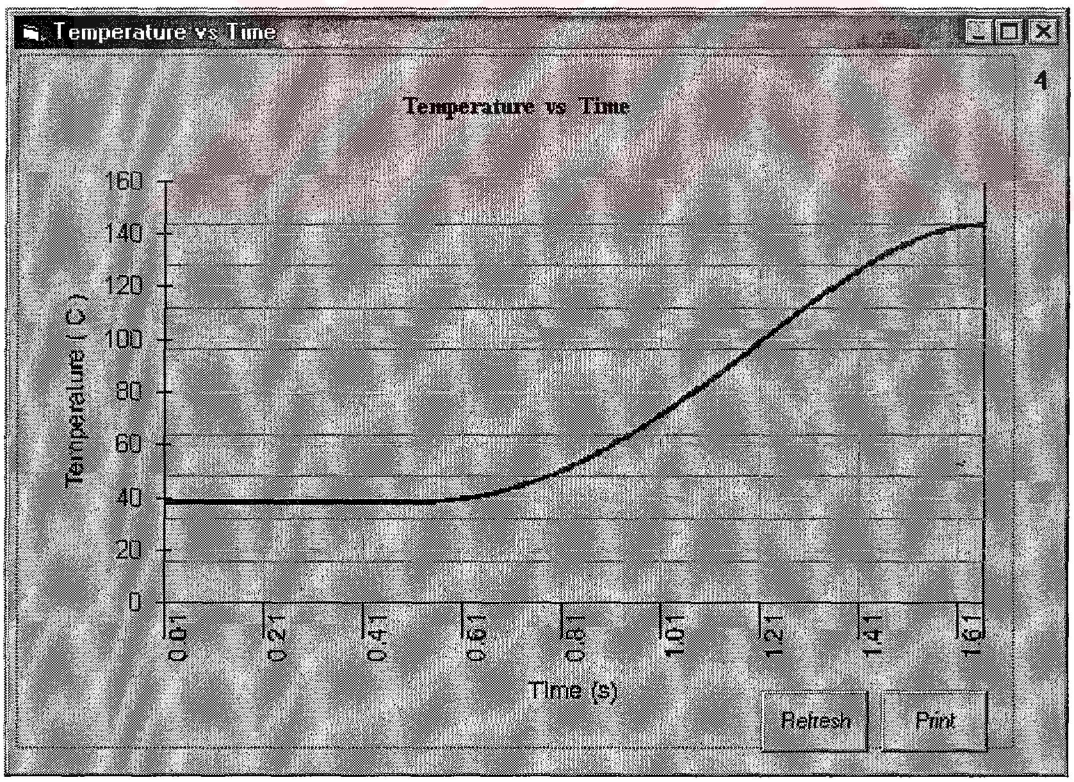


Fig 5.12 Clutch Disc Surface Temperature for the Case Study 2

Short engagement period has many advantages. These are less heat generation during engagement, lower temperatures reached (both the interface and the bulk temperatures), higher velocity reached and longer distance is covered by the vehicle in the same time period. The main reason for that is the low torque characteristics of the engine for low throttle levels.

Case 3: Effect of Clutch Lag Time

In order to investigate the effect of the initial lag time on vehicle performance, the throttle and clutch release scenarios are kept fixed while the clutch lag time is increased. The screen shown in Figure 5.13 was used for 0.5 s of lag time. The slope of the clutch engagement period is kept constant by taking 1 second for the clutch release period. Linearly increasing throttle and clutch release scenarios are considered.

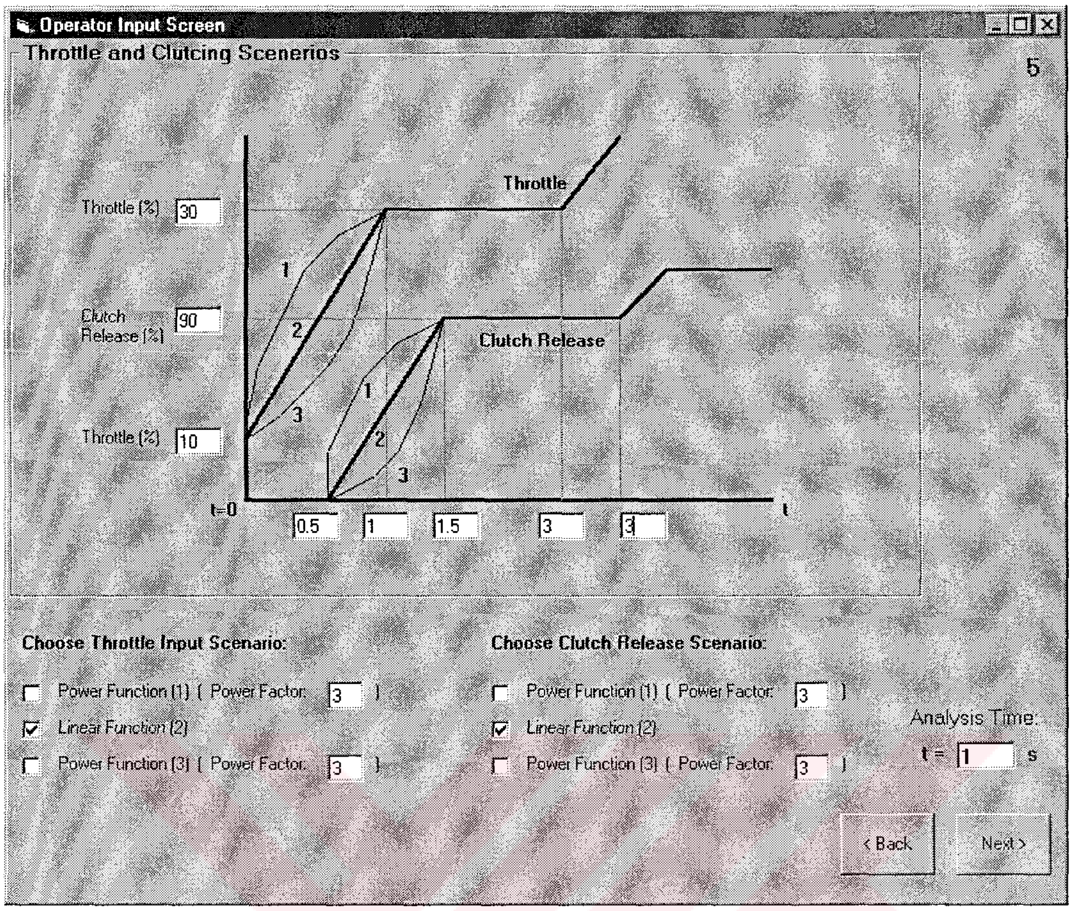


Fig 5.13 The Scenario Used for Calculating Lag Time Effects

The Figures 5.14 to 5.19 show the effect of clutching lag time on slipping time, the distance covered by the vehicle during engagement, the vehicle velocity at the end of the engagement period, average vehicle acceleration, engagement speed and clutch disc surface temperature.

As the initial engine speed gets higher, the slipping time increases. By the increase in difference between the engine flywheel and the clutch disc rotational speeds, the engagement time lengthens.

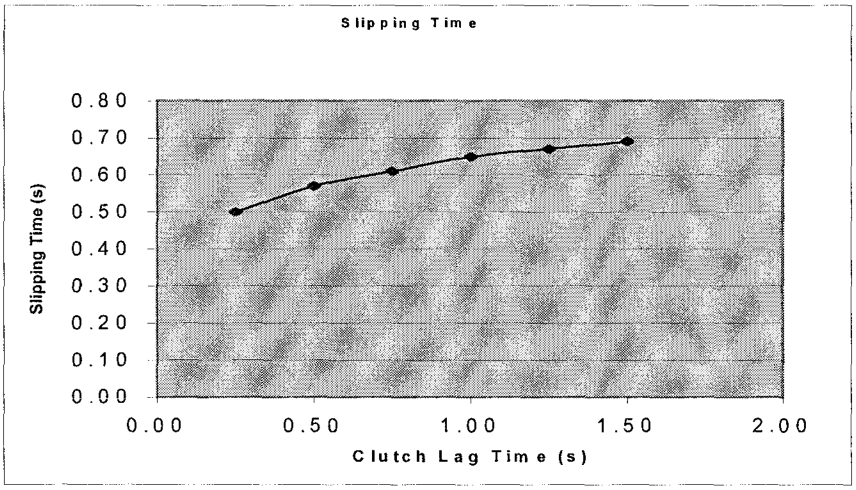


Fig 5.14 Effect of Clutch Lag Time on Clutch Slipping Time

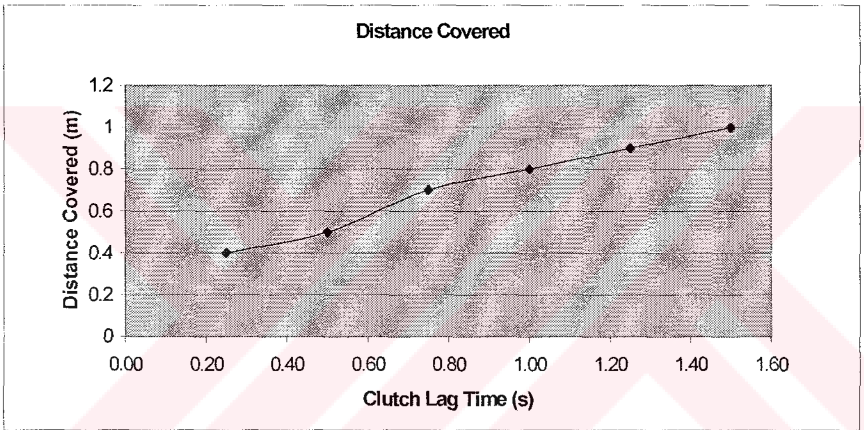


Fig 5.15 Effect of Clutch Lag Time on the Distance Covered by the vehicle

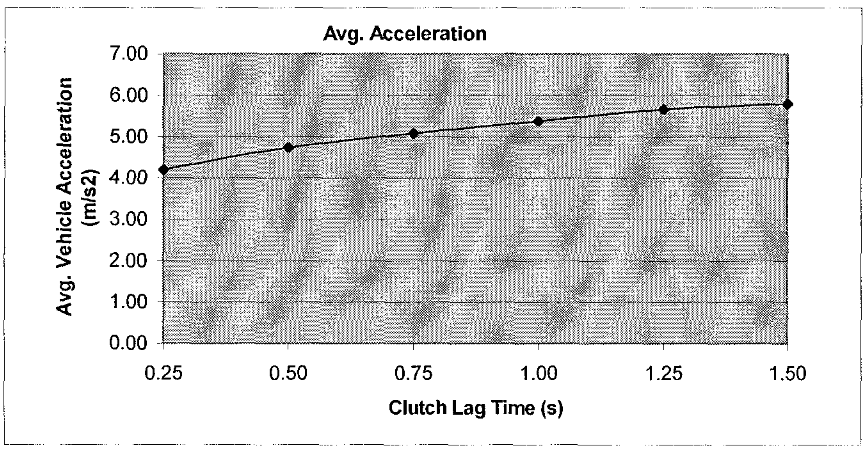


Fig 5.16 Effect of Clutch Lag Time on the Average Vehicle Acceleration

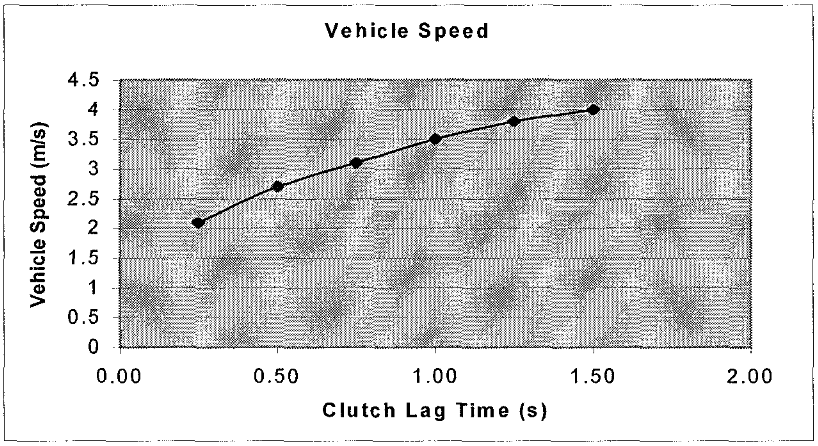


Fig 5.17 Effect of Clutch Lag Time on Vehicle Engagement Speed

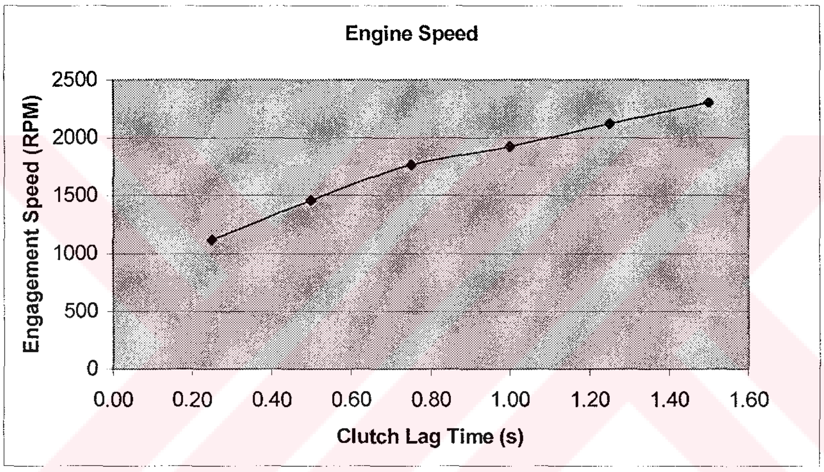


Fig 5.18 Effect of Clutch Lag Time on Clutch Engagement Speed

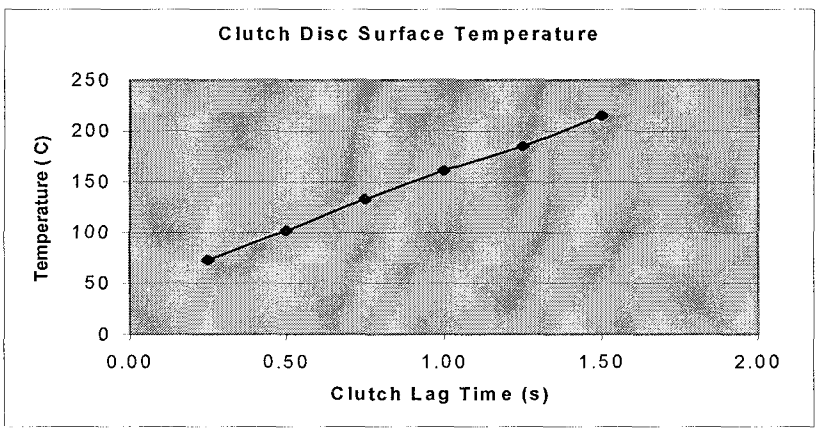


Fig 5.19 Effect of Clutch Lag Time on Clutch Disc Surface Temperature

Decreasing the speed of engine is easier than increasing the clutch speed. Because the total moment of inertia of the engine is much less when compared with the equivalent inertia of the vehicle on clutch disc. In other words, accelerating the vehicle requires more energy than deceleration of the engine by overcoming the engine torque. Theoretically, an ideal engagement happens in the lowest engine speed possible. As in Figure 5.19, the clutch disc surface temperature reached is much less in the short lag time engagement because of the lower initial engine speed.

In terms of vehicle performance the distance covered by the vehicle in a long engagement process may seem long but in fact in the short lag time process, the engagement finishes earlier and the vehicle accelerates as fully engaged. If the same time period is considered for both cases, (short clutch lag time and long clutch lag time) short lag time case may be more advantageous in terms of vehicle performance because in short lag time case the slipping period time is shorter and rest of the time is spent on vehicle acceleration. Additionally the lag time is saved. But the torque characteristic of the engine plays an important role here. If the low speed torque characteristics of the engine is low then the engagement should end in a high rotational speed to make use of the high torque values at high speed.

Case Study 4: Effect of Clutch Release Scenario Characteristics

In order to see the effect of the clutch release scenario, all the clutch release scenario parameters are kept constant and only the characteristics of clutch release motion is changed (Explained in Section 4.2.5). Figure 5.20 is the scenario used for this analysis. The lag time is fixed at 0.8 s and the clutch release period is fixed at 1.2 s. Scenario 1 represents initially a high clutch release rate and expressed with a 3rd order power curve. Scenario 2 represents a constant clutch release rate. Scenario 3 represents an initially lower clutch release rate and expressed with a 3rd order curve.

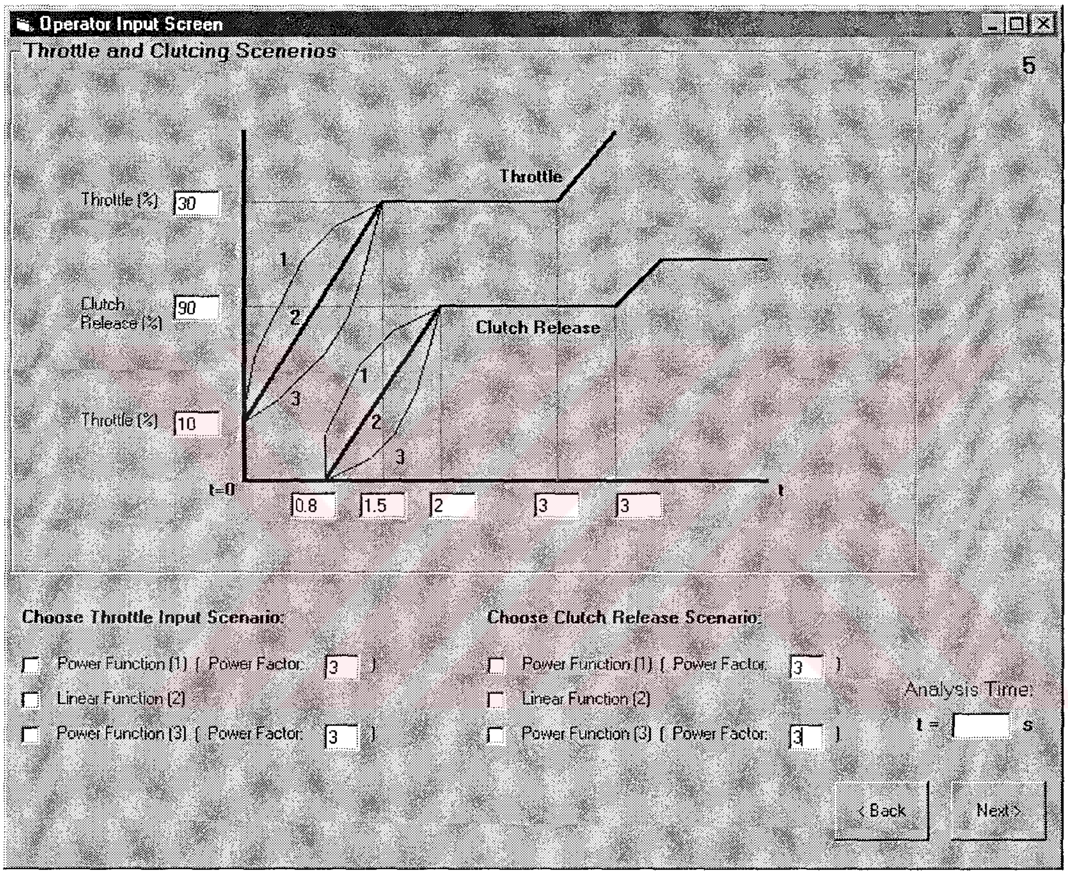


Fig 5.20 Scenario, Used for Clutch Release Characteristics Comparison

The output screen of the scenario shown in Figure 5.20 is given in Figure 5.21. It shows the clutch release motion scenario "1" which is the initial high clutch release rate scenario. The response is quick and the slipping time is 0.4 seconds. The speed of the engine at the end of engagement period is 1480 rpm. The engine may stall if a faster engagement is applied and the engine is not accelerated to a certain value before the engagement starts. The acceleration of the vehicle is high because of the great energy transmission in small time but this may be uncomfortable for the passengers. At the time engagement finishes, the acceleration decreases suddenly to a low value since the engine speed is very low and the torque of the engine at that speed is low.

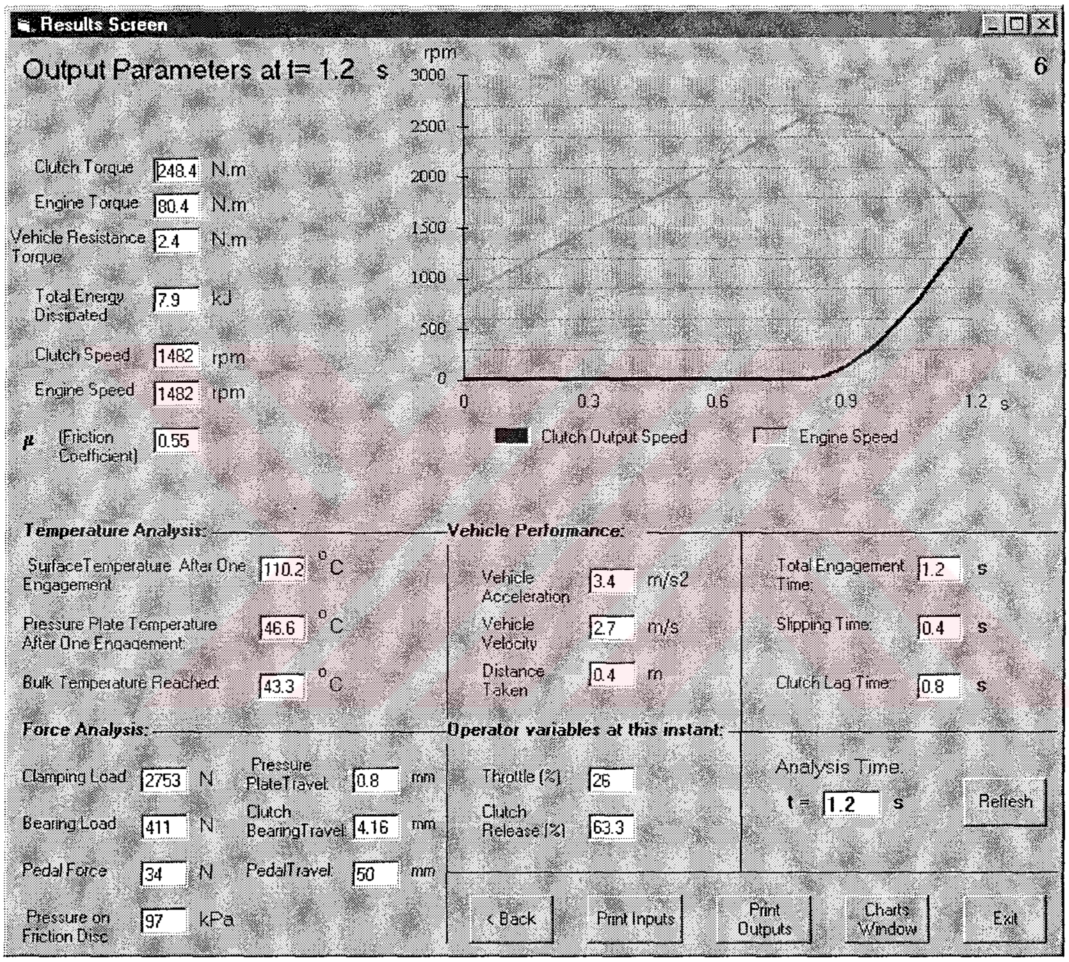


Fig. 5.21 The outputs for clutch release scenario "1"

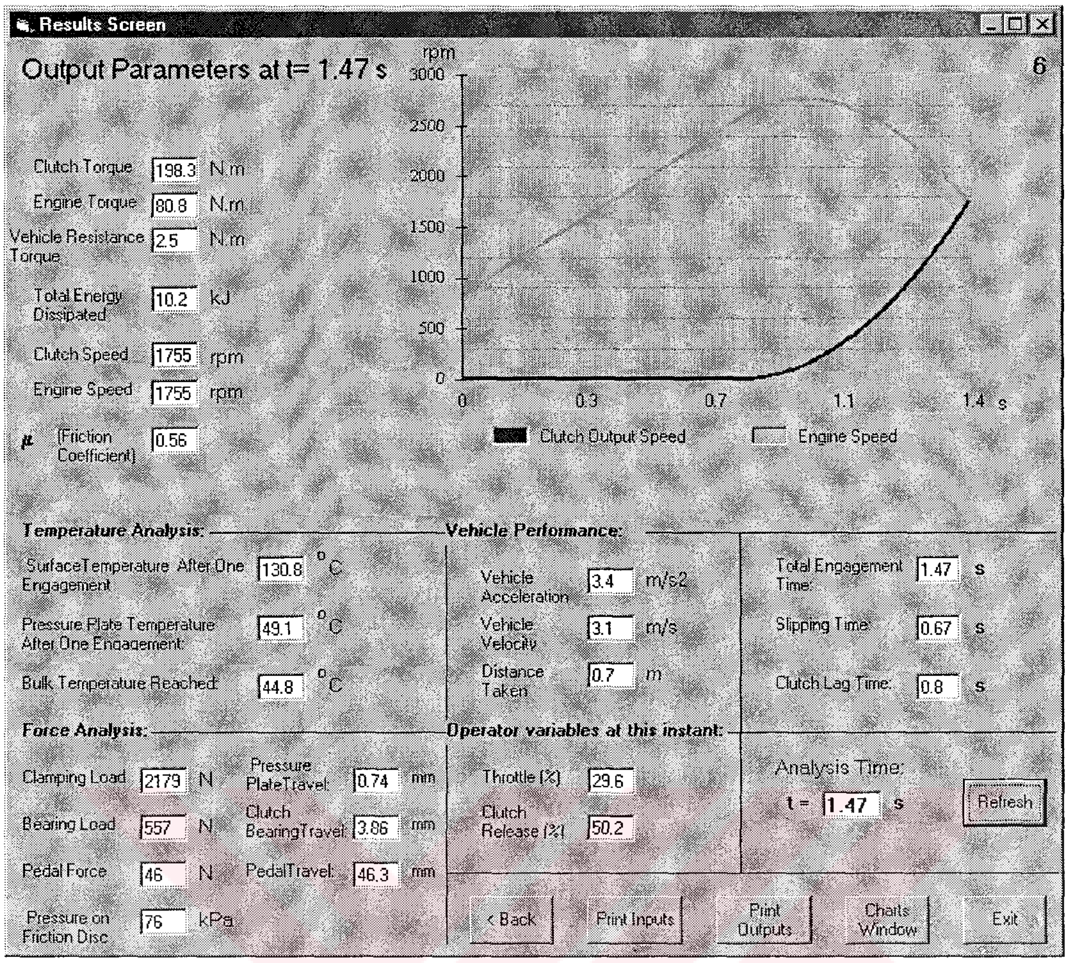


Fig. 5.22 The outputs for clutch release scenario "2"

Figure 5.22 shows the parameters for the linear clutch release motion scenario. The engagement finishes later since the clutch force increase is slower. Figure 5.23 shows the parameters for scenario "3". The clutch release starts at 1.1 seconds although it is specified as 0.8 seconds in the scenario input window. This is because of the selected clutch release motion characteristics. The engagement finishes in 1.91 seconds. Here the slipping time is about 1.11 seconds.

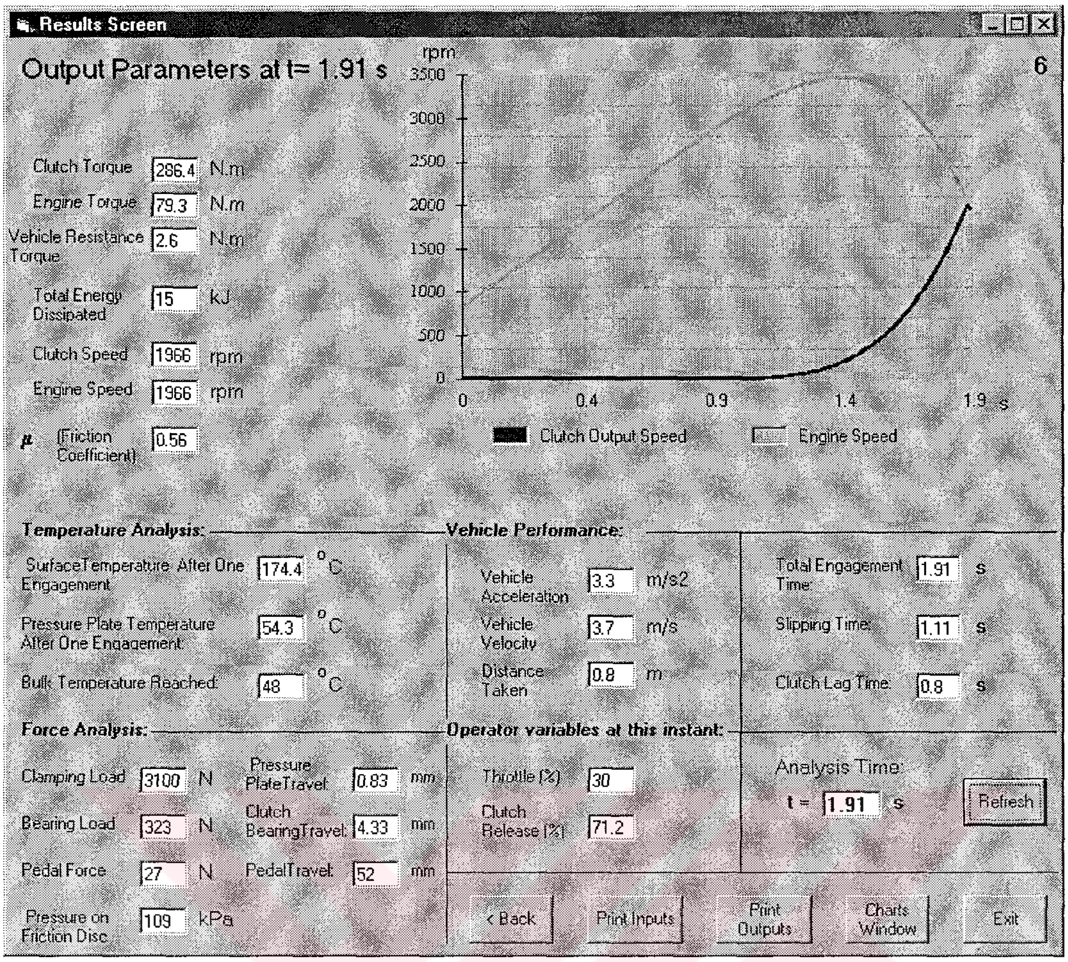


Fig. 5.23 The outputs for clutch release scenario “3”

A 2.0 s analysis is made with the scenario given in Figure 5.20 in order to compare the scenario characteristics and effect of power factors. 2.0 seconds is chosen to ensure a complete engagement in all scenarios applied. At the end of 2.0 seconds, the distance covered, vehicle velocity, average acceleration and slipping time is given in Figure 5.24 for different clutch release scenario characteristics. The vehicle average acceleration and vehicle velocity were not much effected but distance covered is slightly effected by the change of clutch release scenario. The reason is the increasing slipping time.

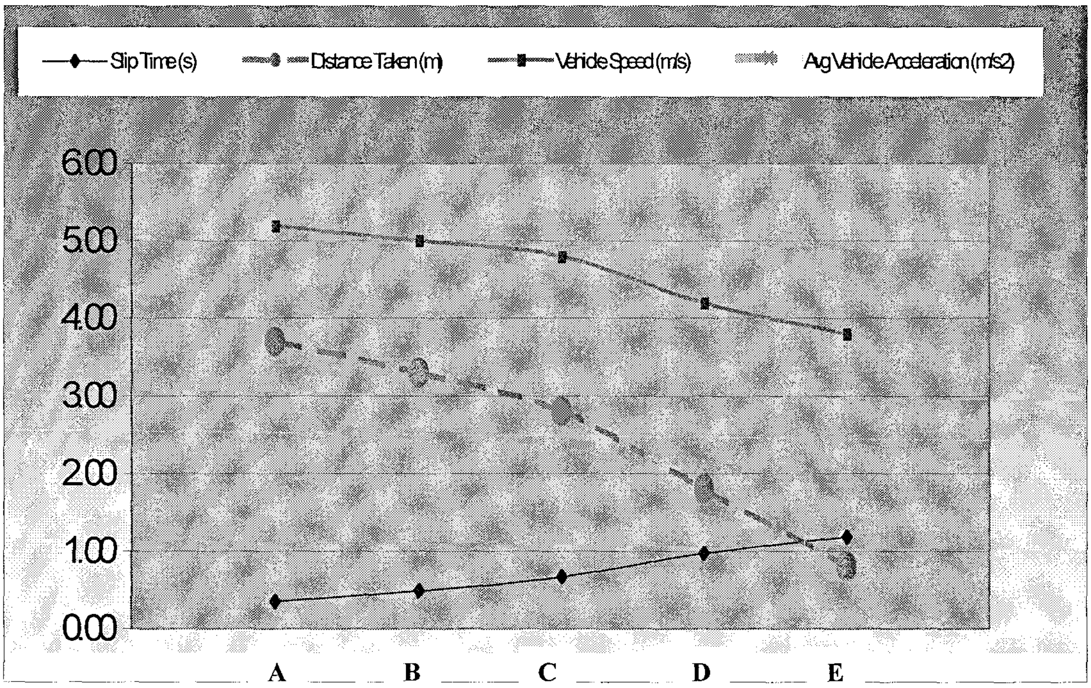


Fig. 5.24 Comparison of Clutch Release Scenario Characteristics

A: Clutch Release Scenario 3, Power Factor = 4

B: Clutch Release Scenario 3, Power Factor = 2

C: Clutch Release Scenario 2

D: Clutch Release Scenario 1, Power Factor = 2

E: Clutch Release Scenario 1, Power Factor = 4

Case Study 5: Effect of Road Gradient on Engagement

In the previous analysis, level road was considered and the vehicle was assumed to be empty. But the design analysis should be made under the worst conditions possible. So the analysis should be repeated with the vehicle mass of 1220 kg (1 driver + 4 passengers of 75 kg each). 10° grade angle will be used and 5 engagements will be done continuously with 10 s time intervals between each engagement. The scenario used for Case Study 5 is shown in Figure 5.25:

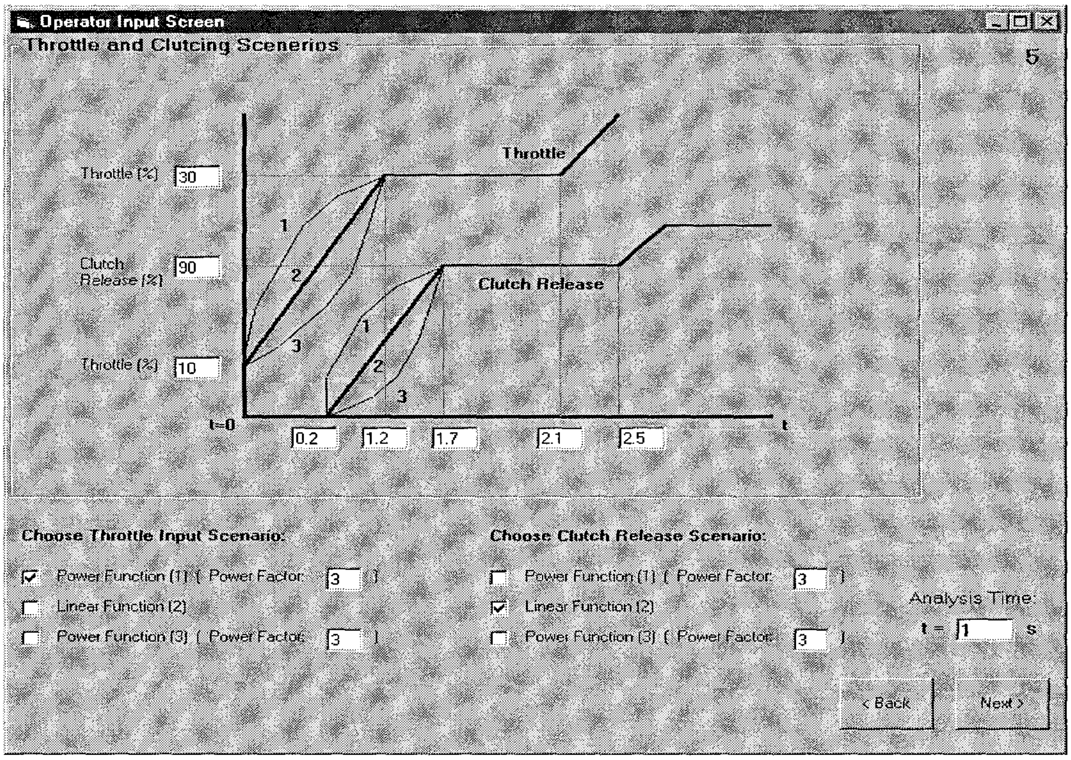


Fig 5.25 The First Driver Input Scenario for the Gradient Road Case

Initial time lag is decreased to 0.2 s to prevent the excess backward motion of the vehicle. The throttle scenario “1” is selected to accelerate the vehicle quickly to a higher speed. The engagement in this case is shown in Figure 5.26.

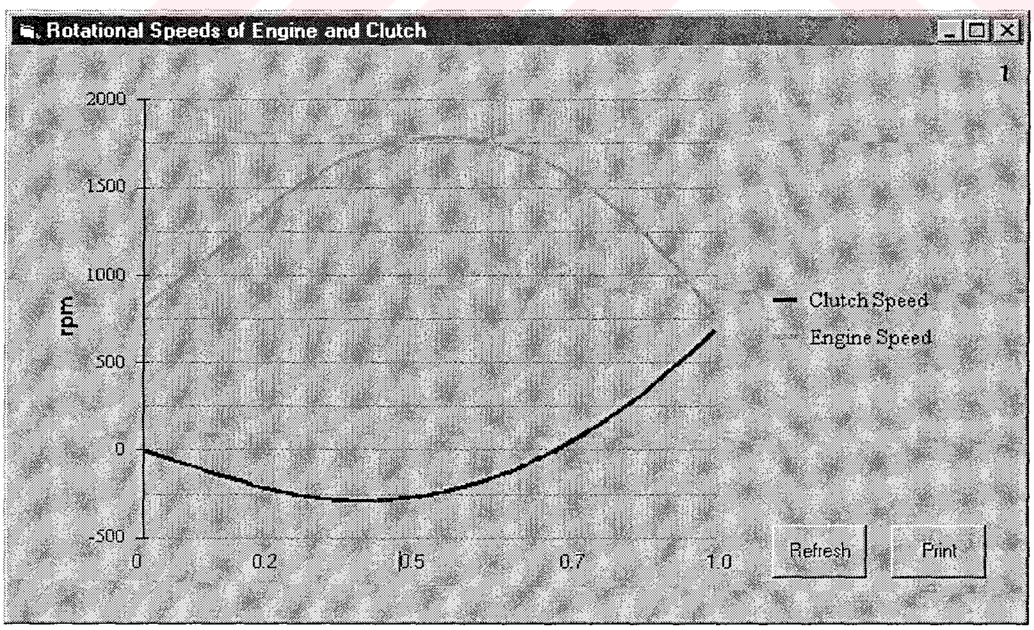


Fig 5.26 Rotational Speeds of the Engine and the Clutch in Case 5

The engine speed first increases and then decreases below 800 rpm at 1 seconds which is defined as the idle speed of the vehicle so engine stall occurs. This engagement is unsuccessful so there is no need to look at other parameters. Another try should be made by changing the scenario inputs as given in Figure 5.27.

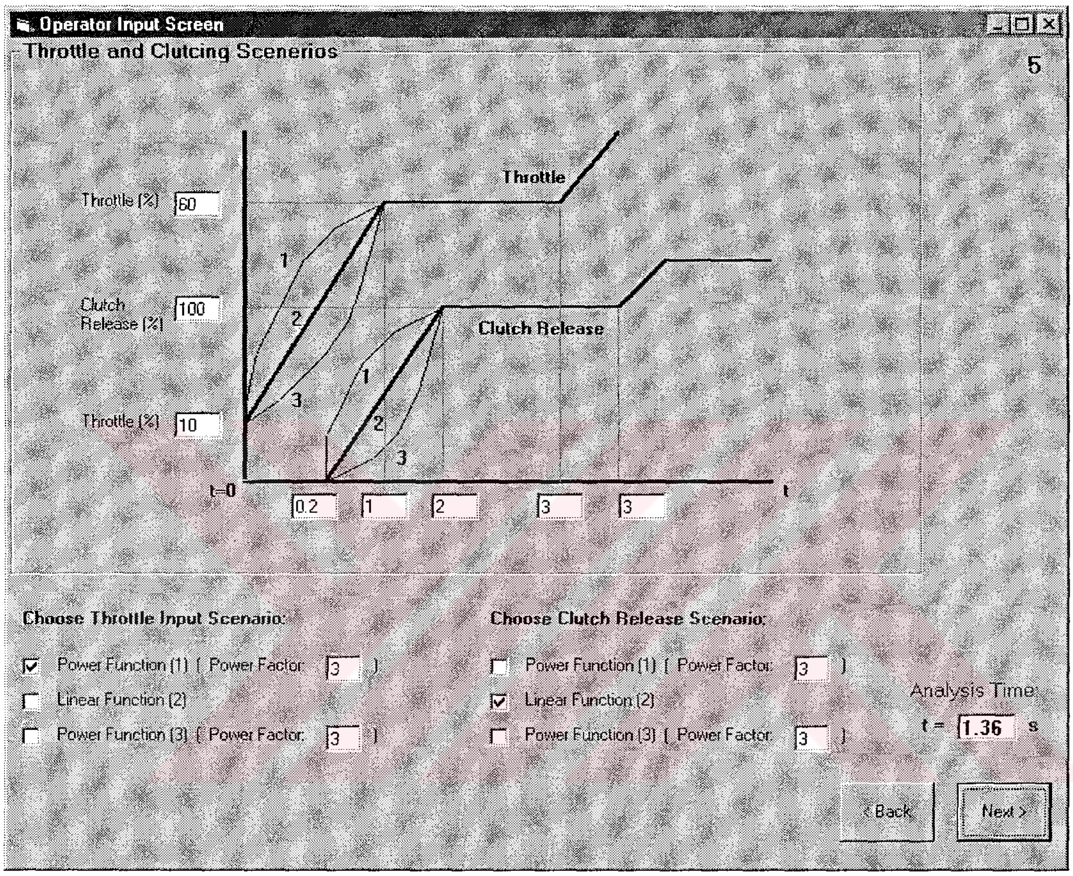


Fig 5.27 The Second Driver Input Scenario for the Gradient Road Case

Here the throttle level is increased to 60 % and its time is decreased to 1 s. Also the engine idle speed is set to 2000 rpm which means that the driver should increase the engine speed by holding the vehicle still, just before the clutch is actuated. The output screen taken by the program is given in Figure 5.28.

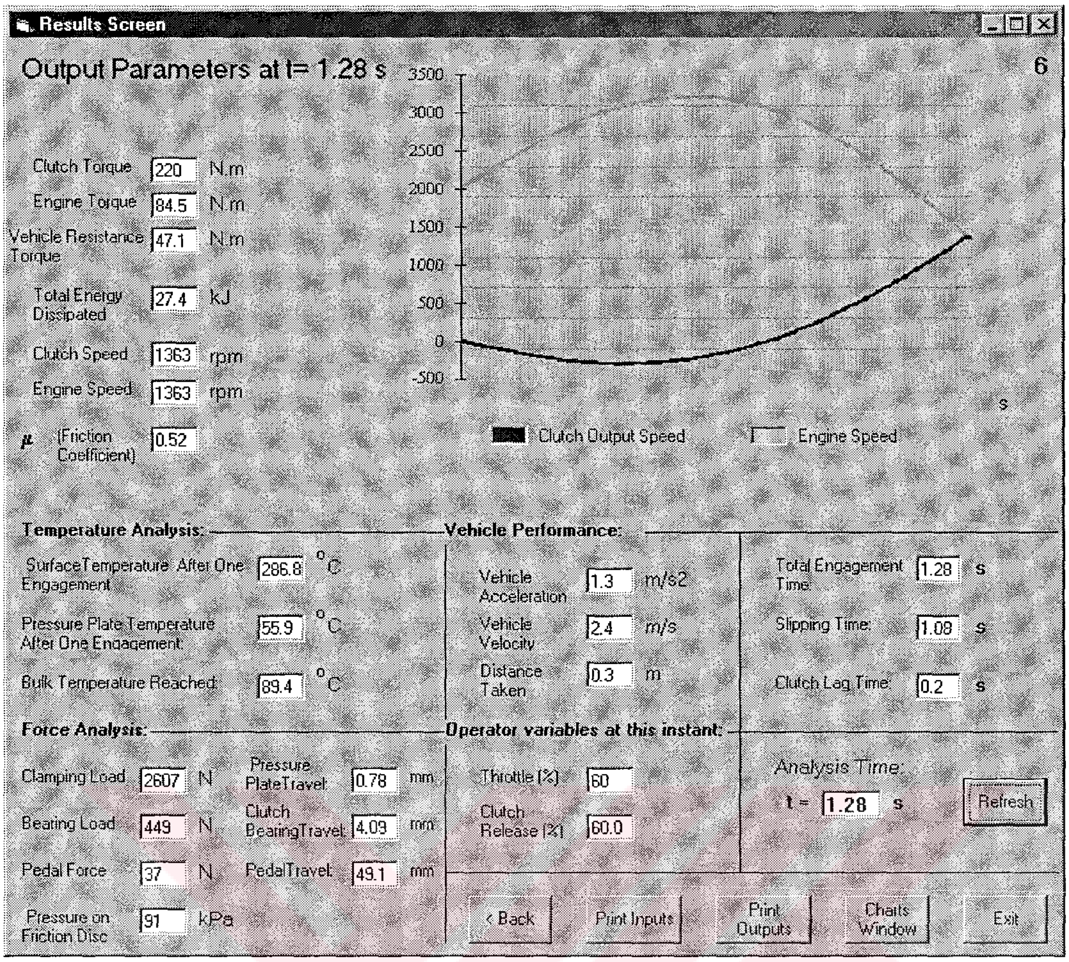


Fig 5.28 The Outputs of the Second Driver Input Scenario for the Gradient Road Case

The engagement in this case happens in 1.28 s and at 1363 rpm. The surface temperature reached is 287 °C which is considered as a high temperature for an engagement and it may cause the friction material failure in multiple engagements. The change in the surface temperature during engagement is given in the Figure 5.29. The surface temperature in this case almost tripled the values calculated on level road. The reason here is the high resistance torque on the clutch and relatively higher slipping time.

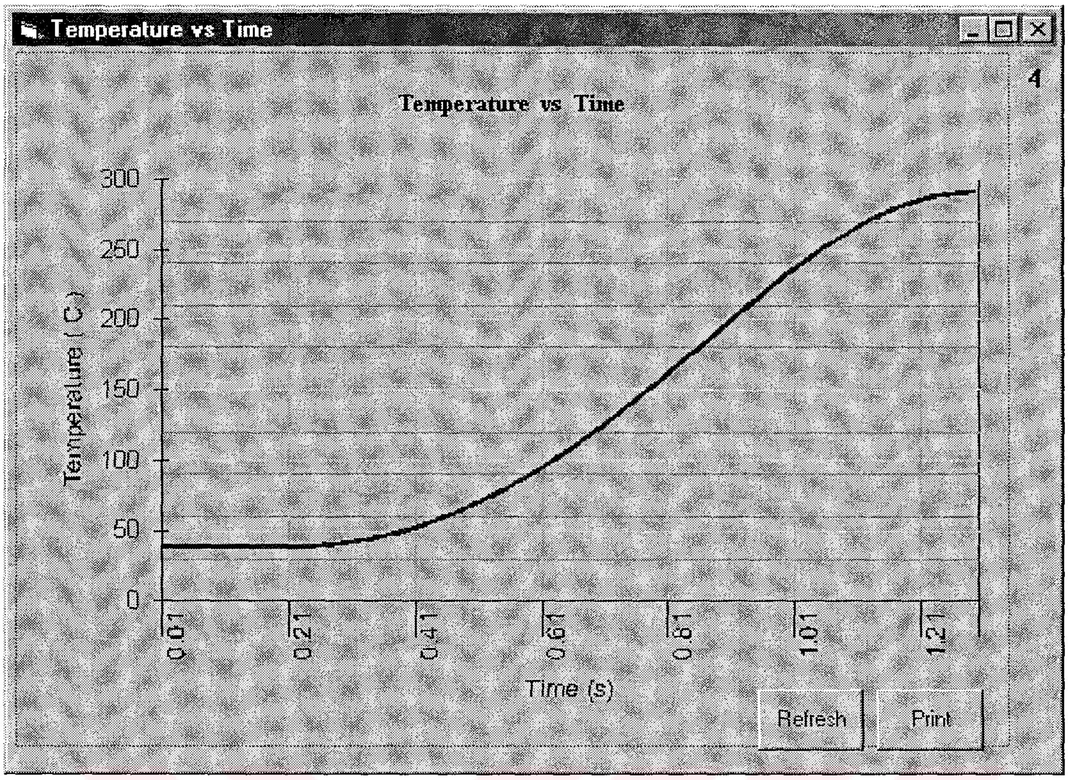


Fig 5.29 The Clutch Disc Surface Temperature Change during Engagement on 10° Gradient Road

The vehicle performance in this case is given in Figure 5.30. The vehicle first moves backwards 0.3 *m* and then starts moving forward. There is a negative acceleration during the clutch lag time by the effect of gravitational acceleration. After 0.65 *m* of forward motion the engagement is completed. This means that the full engagement occurs at 0.35 m from the vehicle initial point.

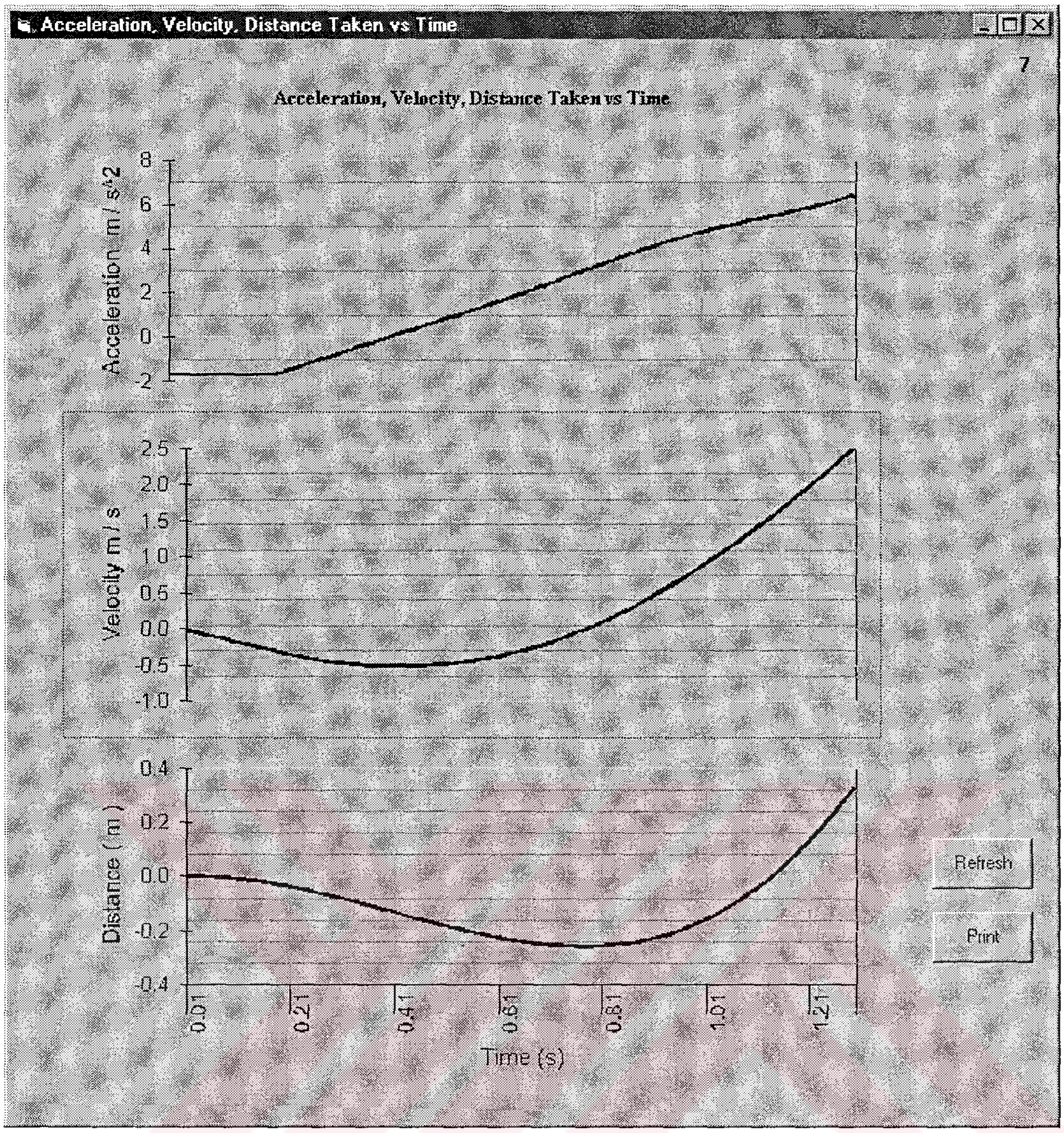


Fig 5.30 Vehicle Acceleration, Velocity, Distance Graphs of the Gradient Road Case

This clutch can be used for this vehicle in most conditions but when the extreme cases (ie. steep gradient, repeated engagements) are taken into consideration there may be failures in the friction material because of high temperatures. Generally for clutch and brake pads the failure occurs after 400 °C. The wear also accelerates exponentially after 250-300 °C [17, 18]. So for the extreme cases, the analysis can be repeated with a larger diameter clutch pair. If the same vehicle is analysed under the same conditions (ie. 10 ° gradient, fully loaded vehicle, 10 repeated engagements) with Valeo ϕ 200 R32 QX/490 friction disk and a suitable pressure plate of Valeo 200 CPO 3500 diaphragm spring, the engagement time and temperatures may be reduced. The outputs taken from the program with this friction disc and pressure plate is shown in Figure 5.31.

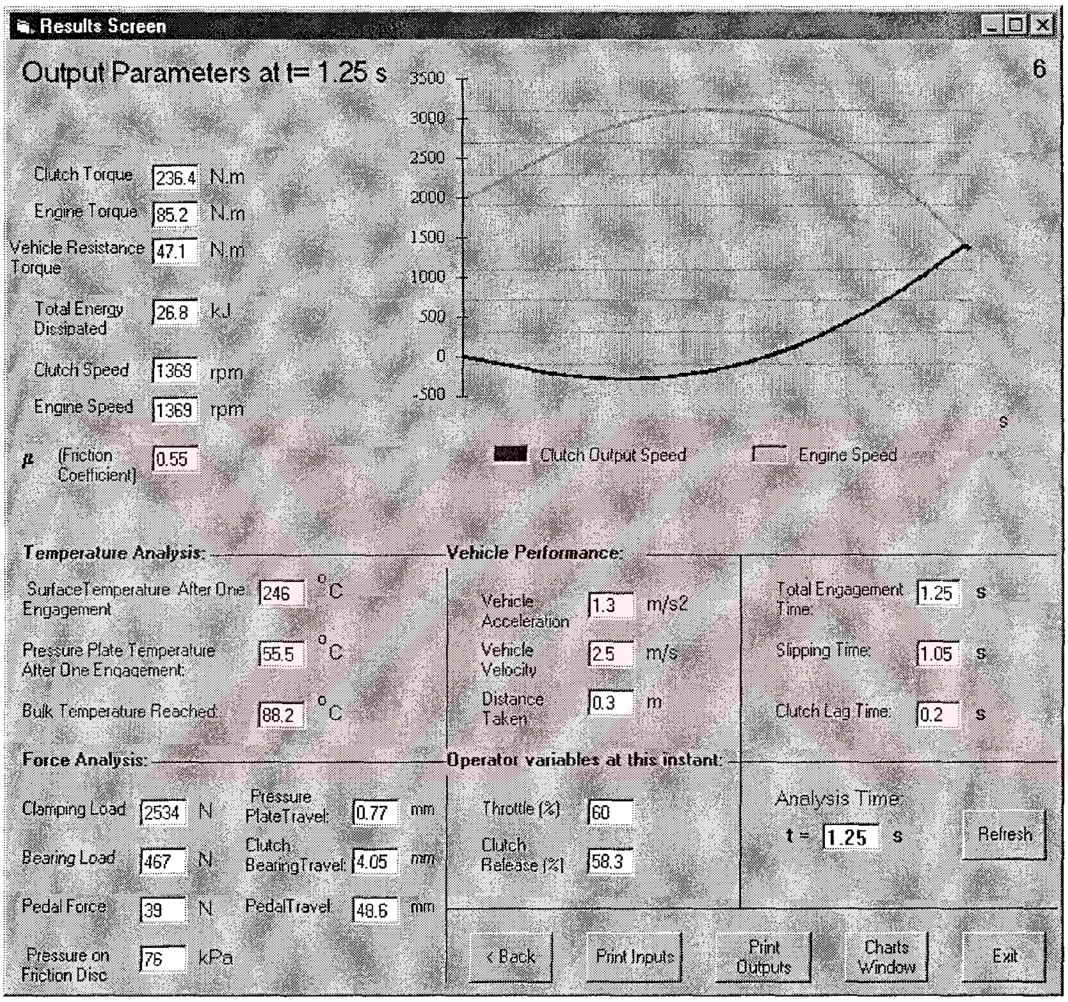


Fig 5.31 The Outputs for the 10° Gradient Road Case with 200 mm Diameter Clutch Disc

With the same driver throttle and clutch release scenario, a lower temperature is reached both at the surface and the clutch assembly. Practically the engine speed (1370 rpm) at the end of engagement may seem low but the shortest and most efficient engagements can be achieved by decreasing the engine speed to the lowest value possible. Higher starting speed or higher throttle values can be used in the scenario to decrease the probability of engine stall during engagement.

CHAPTER 6

DISCUSSION OF RESULTS

The clutch size, spring properties, engine torque map, vehicle parameters and the road conditions are the important parameters in the engagement process and clutch selection. The case studies are performed to show the effect of these variables on the engagement process and the vehicle performance.

The results taken from the case studies show that, the engagement process depends mainly on the throttle level increase and clutch release. In the first two case studies the effect of slipping time is performed on the level road with empty vehicle. The first case study is given to analyse a high clutch release rate and the second case study is for low clutch release rate.

Comparing vehicle performance characteristics of these two scenarios shows that in the high clutch release rate scenario (Case study 1), the vehicle takes a longer distance and the vehicle speed is higher in the same time period. The reason here is due to the low release rate of the clutch pedal.

As it is expected in the high release rate situation, the engagement is quick and the heat dissipated by the clutch is low. So both the clutch disc interface temperature and the bulk temperature of the clutch assembly after the engagement is low. In low release rate situation, it is seen that the energy dissipated by the clutch and thus the temperatures reached are increased. The engine is accelerated to 2400 rpm and then comes down to 2000 rpm. The high angular velocity difference between the engine and the clutch causes high energy dissipation. Also the engagement time is much longer compared with the high release rate situation. This example shows the idea of low throttle engagements causes shorter friction time and less heat dissipation [1, 2].

Lower engine speeds at the end of the engagement means less heat is generated. The moment of inertia of the engine is lower than the total moment of inertia of the transmission parts. So lowering the engine speed is easier than increasing the clutch speed. Theoretically engaging at the minimum engine speed (idle speed) is the most advantageous case considering the temperature increase, wear and thus the clutch life. But practically for the drivers, it is not easy to engage in this way because the engine stall can occur. Also the low torque characteristics of the engines at low rotational speeds makes it hard to engage at low speeds. It can also be harmful for the engine to run at low speeds under load around its idle speed.

The effect of road gradient on clutching performance is analysed under the full vehicle load with 10° grade angle. The fast pedal release caused engine stall because of the sudden drop of the engine speed under idle speed. The reason for the sudden drop is low engine speed and fast clutching action. So in the second scenario an initially higher throttle level increase in a smaller time used. Also the clutch is applied slowly letting the clutch speed increase while the engine speed decreases slowly. The engagement is completed in 1.28 s. It is a short engagement interval and at the end of the engagement period, the engine speed is 1360 rpm which is a low value. Although the engagement is quick, the temperatures reached both on the interface and the clutch assembly are quite high. The reason is the high resistance torque on the clutch because of the gradient and the increased weight of the vehicle.

In order to see the effect of the friction area of the friction disc on engagement, 200 mm diameter Valeo® clutch disc is assumed instead of 190 mm diameter one in the next analysis. This caused a quicker engagement. With the same scenario, the friction disc surface temperature and the bulk temperature of the clutch disc assembly decreased. This shows the importance of the friction disc surface area on the engagement parameters.

From the case studies given, it is observed that the temperatures reached are directly related to the slipping period. Short slipping periods cause lower temperatures and thus longer clutch life. Short slipping periods means less difference between the engine speed and clutch speed.

Higher kinetic energy of the engine gives user the option of applying a higher clutch release rate without stalling the engine. Even the heat dissipation of the clutch is higher in this case, the transmitted energy is higher in the same time period. So if the performance is the prior consideration, the engagement should start with a high engine speed and a high clutch release rate should be applied.

The time necessary to increase the engine speed should also be considered during the analysis.

The decision of selecting the clutch size can be made by considering the operational conditions. For example, if the vehicle to be analysed is a sports car then the analysis should be made with repeated and high engine speed engagements. After several tests under different simulation conditions, it will be possible to select or design the most suitable clutch sizes for a specified vehicle.

CHAPTER 7

CONCLUSION AND FUTURE RECOMMENDATIONS

This study's main objective is to develop a design and analysis software for friction clutches. This study is performed on a realistic basis trying to avoid common assumptions in the literature. From the flywheel to the vehicle wheels, a complete analysis has been made to predict the engagement parameters in a real clutch engagement.

Although the computer program includes all the variables needed to be analysed, there are some modifications that can be made on it. These are:

- Friction material list can be improved. Also the friction model can be modified to give more realistic results. The effect of pressure can be added in the friction model.
- Clutch disc and diaphragm spring libraries can be extended to cover larger sizes for heavy vehicles.
- In the thermal analysis, the thermal properties of all the friction materials are assumed to be the same because of the lack of data. This assumption can be modified by using different k, ρ, c values for different friction materials.
- A vehicle library can be formed. So that the user can select different vehicle-clutch pairs easily and see the effect on performance of the vehicle.

- Different types of actuation systems can be optional for the user to select. So the clutch actuation system can be analysed in detail.
- Friction disc wear and clutch life analysis can be added in the program.

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APPENDIX

A. FLOWCHART OF THE COMPUTER PROGRAM

B. FRICTION MATERIAL LABARATORY TEST RESULTS

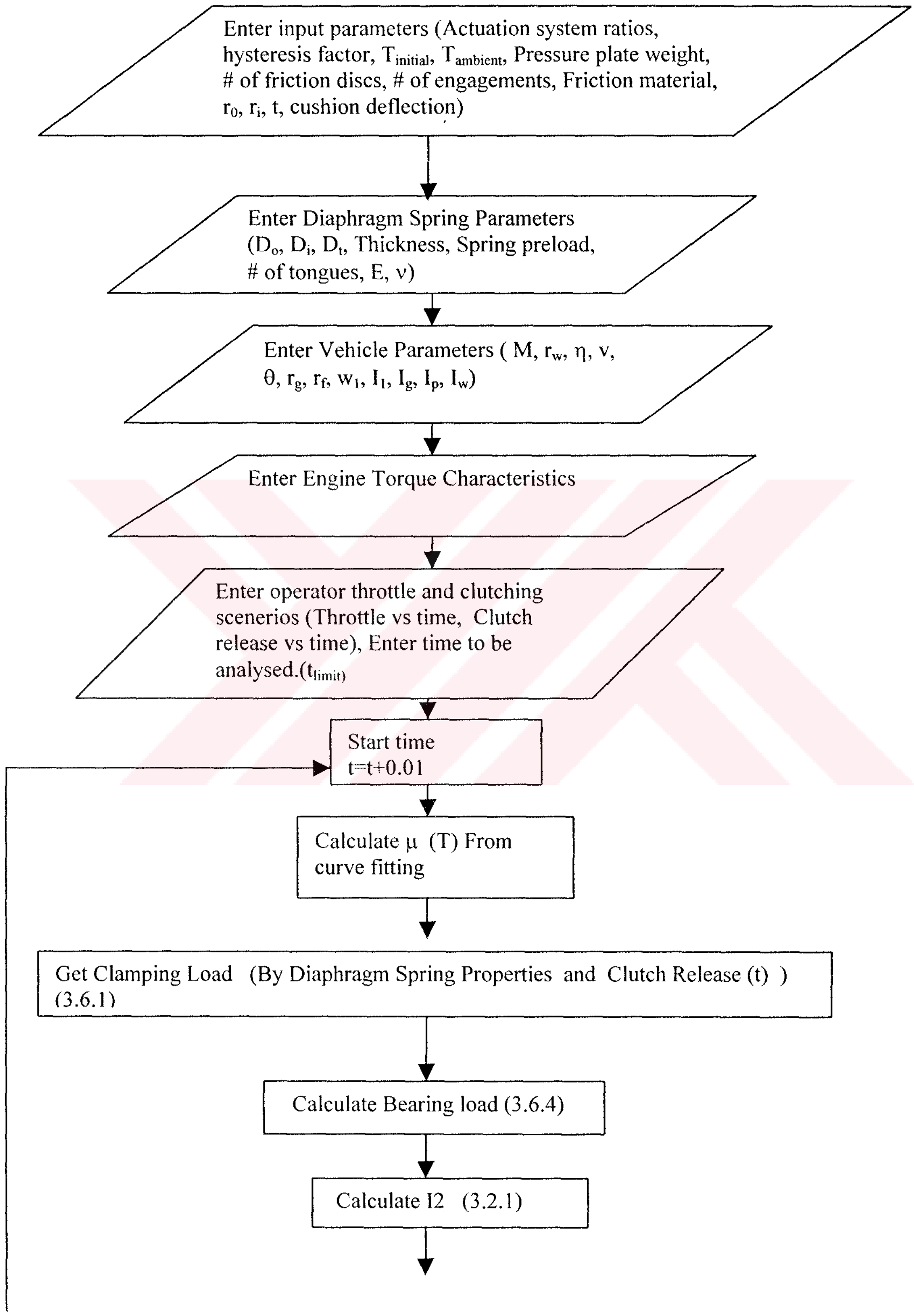
C. COEFFICIENT OF FRICTION VS. TEMPERATURE GRAPHS FOR DIFFERENT FRICTION MATERIALS

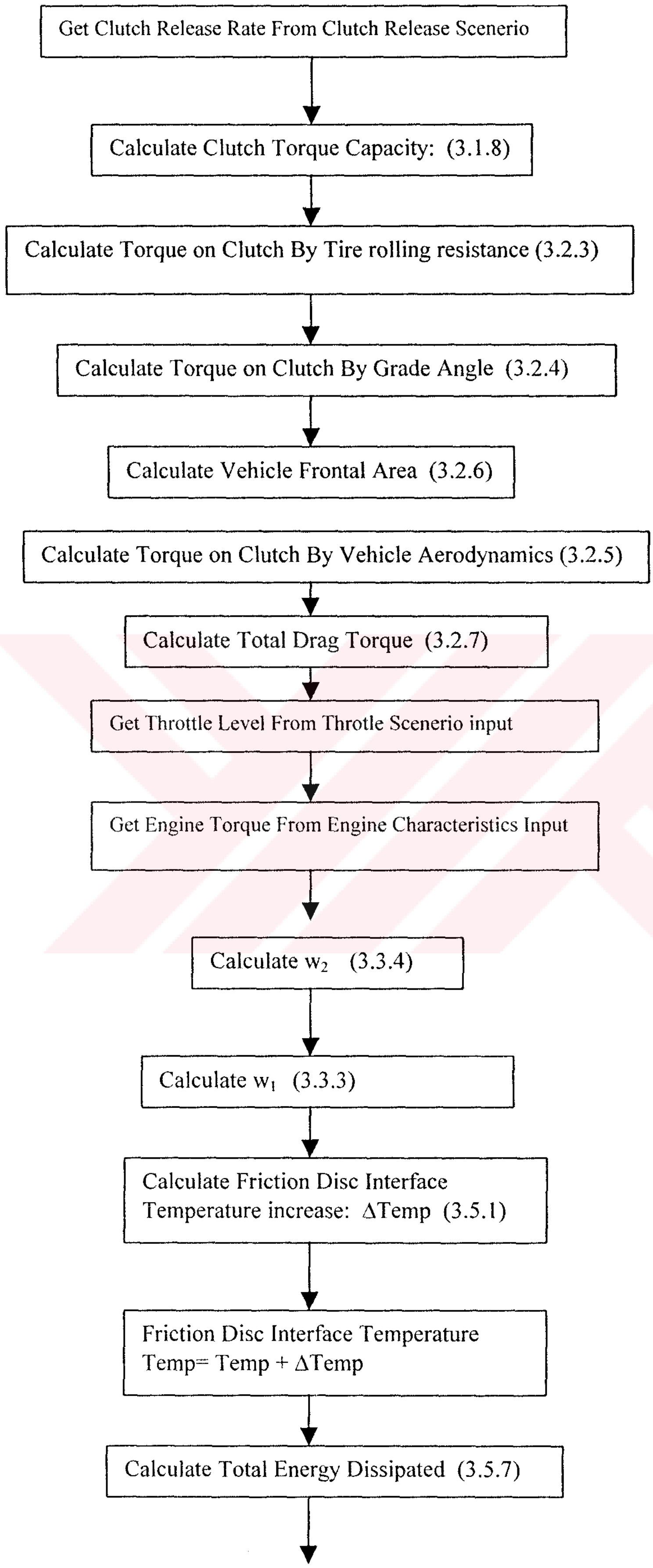
D. PROPERTIES OF CLUTCH LININGS

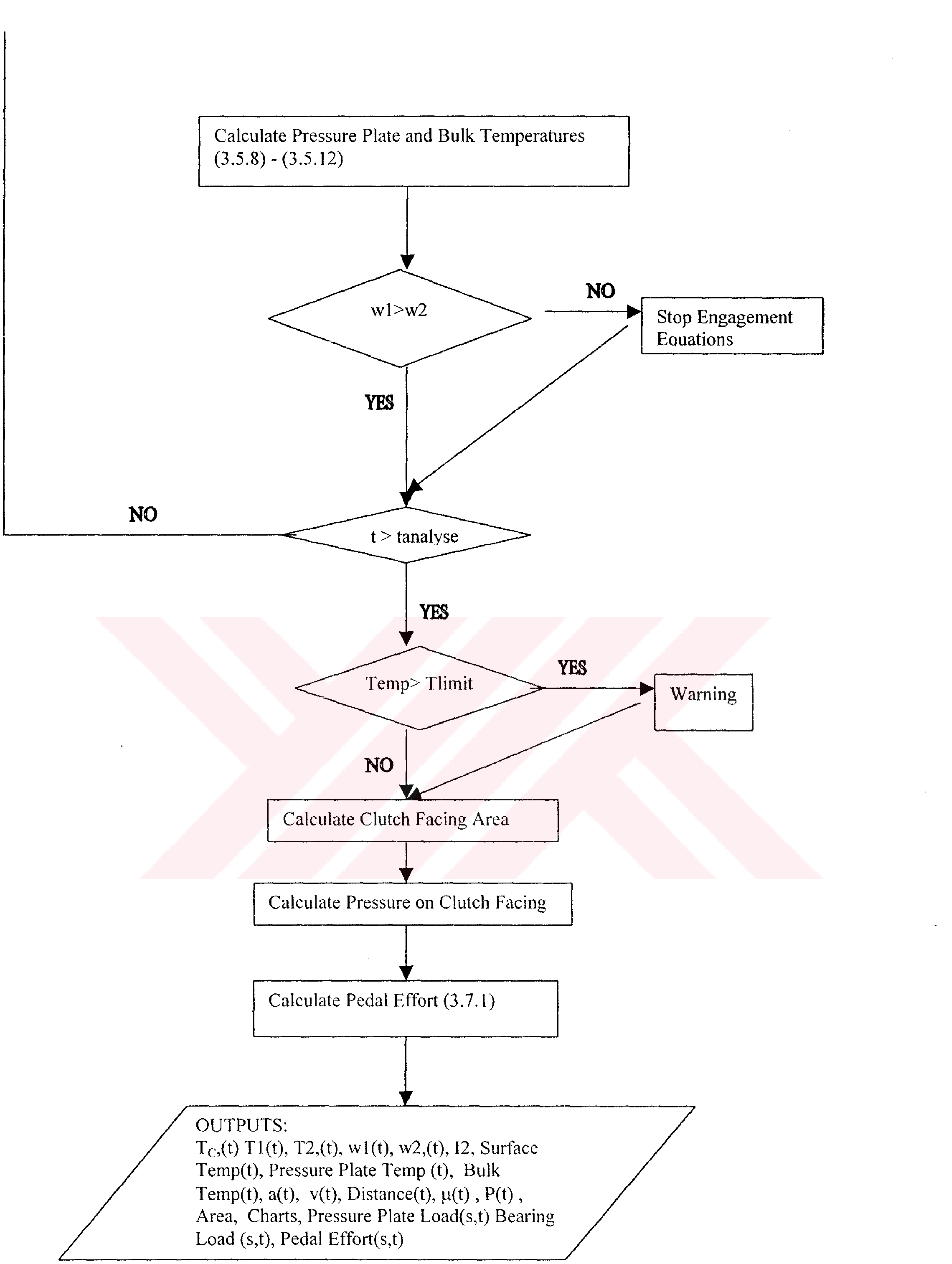
E. VALEO® FRICTION DISCS AND DIAPHRAGM SPRINGS PARAMETERS

F. ENGINE TORQUE CHARACTERISTICS OF JEEP CHEROKEE 4.0 L GASOLINE ENGINE

A. FLOWCHART OF THE COMPUTER PROGRAM




Get Clutch Release Rate From Clutch Release Scenerio
Calculate Clutch Torque Capacity: (3.1.8)
Calculate Torque on Clutch By Tire rolling resistance (3.2.3)
Calculate Torque on Clutch By Grade Angle (3.2.4)
Calculate Vehicle Frontal Area (3.2.6)
Calculate Torque on Clutch By Vehicle Aerodynamics (3.2.5)
Calculate Total Drag Torque (3.2.7)
Get Throttle Level From Throtle Scenerio input
Get Engine Torque From Engine Characteristics Input
Calculate w_2 (3.3.4)
Calculate w_1 (3.3.3)
Calculate Friction Disc Interface Temperature increase: ΔTemp (3.5.1)
Friction Disc Interface Temperature Temp= Temp + ΔTemp
Calculate Total Energy Dissipated (3.5.7)


Calculate Pressure Plate and Bulk Temperatures
(3.5.8) - (3.5.12)
w1>w2
NO
Stop Engagement Equations
YES
NO
t > tanalyse
YES
Temp> Tlimit
YES
Warning
NO
Calculate Clutch Facing Area
Calculate Pressure on Clutch Facing
Calculate Pedal Effort (3.7.1)
OUTPUTS:
Tc,(t) T1(t), T2,(t), w1(t), w2,(t), I2, Surface Temp(t), Pressure Plate Temp (t), Bulk Temp(t), a(t), v(t), Distance(t), μ(t) , P(t) , Area, Charts, Pressure Plate Load(s,t) Bearing Load (s,t), Pedal Effort(s,t)

B. FRICTION MATERIAL LABARATORY TEST RESULTS

Table B.1 Friction Material Labaratory Test Resul

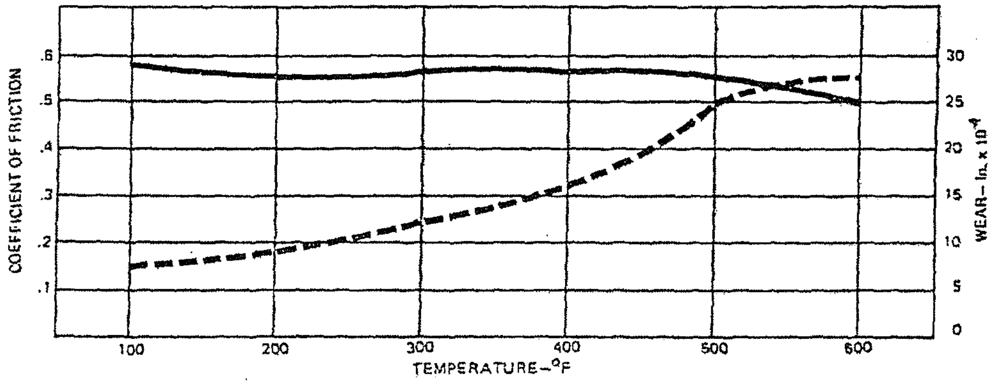
TYPICAL CLUTCH DYNAMOMETER DATA

Summary of six manufacturers' clutch dynamometer lab data on one popularly approved molded glass/acrylic fibered and phenolic matrixed friction material.

TEST SOURCE	INERTIA		RPM	DISC SIZE		TOTAL AREA (2 facings)		Control Temp.	Radius of Gyration (R.)		Cycles/ Test	Cycles/ Min.	K.E./CYCLE		Power Density		Average Coefficient	Total Wear	TEST SOURCE
	SI•Ft.^{2}	Kg. f.•M^2		Inches	mm.	In.^{2}	Cm.^{2}	°C	Ft.	Cm.			Lb.•Ft	Joules	H.P./In^2/cycle	Watts/cm^2/cycle	μ_d	mm	
LAB 1.	2.12	2.88	1500	7.87X5.28	200X134	53.68	346.34	N.A.	.279	8.51	3000	1.5	26151	35526	.021	2.44	.28	.17	LAB.1
LAB 2.	1.11	1.50	1750	11.81X7.48	300X190	131.14	846.12	100, 200, 300	.412	12.55	5000	variable[1]	16118	21920	NA	NA	NA	NA	
	.86	1.16	1650	7.87X5.12	200X130	56.80	366.48	100 200 250 300	.276	8.43	1000 0		14329	19487	NA	NA	.42	NA	LAB 2.
LAB 3.	7.18	9.73	3000-2930	7.87X5.28	200X134	26.84	173.17	N.A.	.279	8.51	40	.05	16340	22052	.005	0.58	NA	NA	LAB 3.
			3277-2773								25		130846	176588	.037	4.30			
			2070-								1		168671	227635	.048	5.58			
			3277-2723								25		130846	176588	.037	4.30			
			3387-2613								–		182807	246713	.052	6.04			
LAB 4.	2.36	3.2	1450	7.87X5.28	200X134	53.68	346.34	N.A.	.279	8.51	6000	1.5	27203	36886	.022	2.56	.41	.20	LAB 4.
LAB 5.	5.37	7.28	2230	11.00X6.50	279X165	120.96	780.44	N.A.	.376	11.46	1000	1.13	146406	199112	.039	4.53	.37	.33	LAB 5.
	2.01	2.72	1650	8.00X6.00	203X152.4	43.96	284.00	N.A.	.294	8.96	6000	3.50	30001	40559	.075	8.71	.34	.42	
LAB 6.	.64	0.87	3000	7.87X5.12	200X130	56.80	366.48	200	.276	8.43	500 1000 1000 500	variable[1]	31579 31579 45888 60197	43076 43076 62418 81809	NA	NA	.44 .44 .42 .39	.03 .05 .13 .12	LAB 6.
	.64	0.87																	
	.93	1.26																	
	1.22	1.65																	

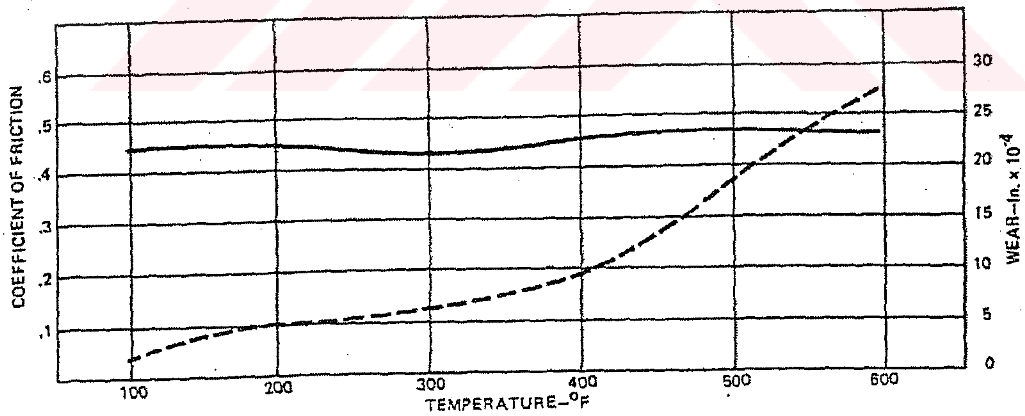
1. Cycles/min. varied to maintain 200° C surface temperature.

C. COEFFICIENT OF FRICTION VS. TEMPERATURE GRAPHS FOR DIFFERENT FRICTION MATERIALS



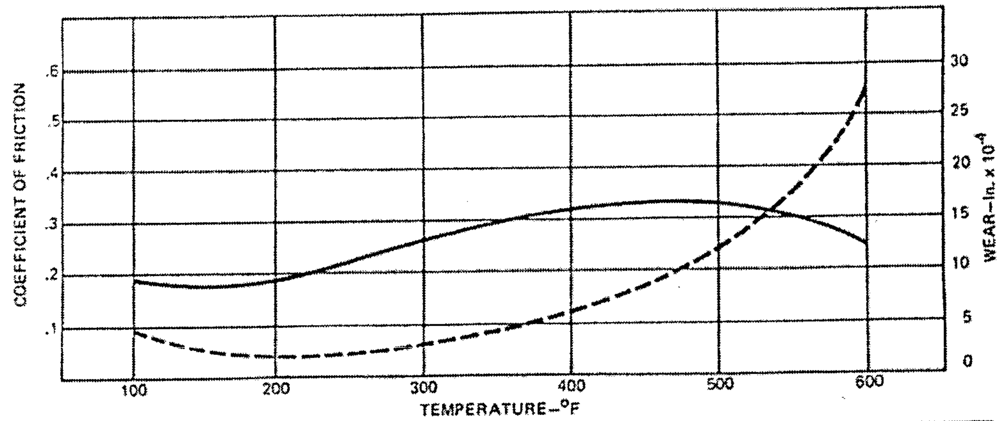
Average Friction	Average Wear	Overall Wear
100°-600°: .464 100°-300°: .460 400°-600°: .447	100°-600°: .00115 in. 100°-300°: .00048 in. 400°-600°: .00183 in.	.00697 in.

Fig. C.1 Rigid Molded High Friction Material



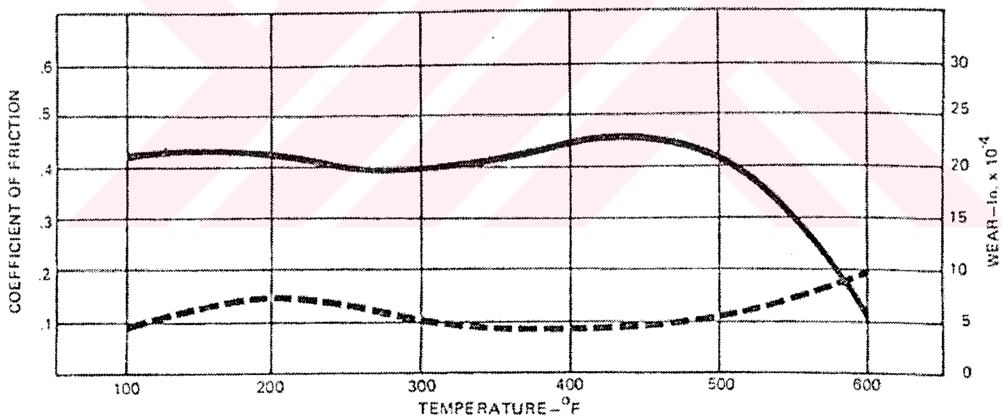
Average Friction	Average Wear	Overall Wear
100°-600°: .449 100°-300°: .439 400°-600°: .460	100°-600°: .00112 in. 100°-300°: .00043 in. 400°-600°: .00182 in.	.00674 in.

Fig. C.2 High Friction Material for Industrial Brakes and Off-road Equipment



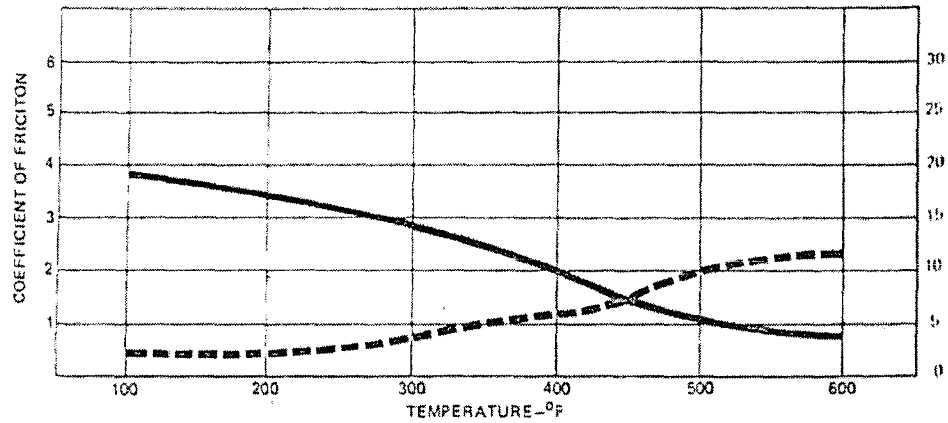
Average Friction	Average Wear	Overall Wear
100°-600°: .259 100°-300°: .213 400°-600°: .306	100°-600°: .00092 in. 100°-300°: .00031 in. 400°-600°: .00153 in.	.00554 in.

Fig. C.3 Rigid-molded asbestos material for industrial brakes



Average Friction	Average Wear	Overall Wear
100°-600°: .371 100°-300°: .416 400°-600°: .326	100°-600°: .06116 in. 100°-300°: .05700 in. 400°-600°: .06533 in.	.00367 in.

Fig. C.4 Woven Material of Asbestos Yarn With Brass Wire Inserts for Face, or Multiple-disc brakes and Clutches



A Average Friction	Average Wear	Overall Wear
100°-600°: .242 100°-300°: .343 400°-600°: .140	100°-600°: .00621 in. 100°-300°: .00310 in. 400°-600°: .00933 in.	.00373 in.

Fig. C.5 Low-Friction, Rigid Molded Material from Phenolic Resin, Friction Modifiers and Asbestos Fibers

D. PROPERTIES OF CLUTCH LININGS

Table D.1 Properties of Clutch Linings

Name of Property	Woven Lining	Molded Lining	Rigid Block
Comp. strength, MPa	70-100	10-18	10-15
Tens. strength, MPa	17-21	27-35	21-27
Max. temperature, °C	200-260	260	400
Max. speed, m/s	38	25	38
Max. pressure, kPa	340-690	690	1000
Frictional coeff., mean	0.45	0.47	0.40-0.45

E. VALEO® FRICTION DISCS AND DIAPHRAGM SPRINGS PARAMETERS

Table. E.1 Friction Disk Properties of Some Valeo Clutche

	FRICTION DISKS									
		Inner	Deflection	Max.	Max. Def.	Thickness at		Moment		
VEHICLE	CODE	Diameter	Start	Deflection	Load	This Load	Mass	of Inertia	Spring Torque/Angle	Spring Torque/Angle
		(mm)	(N)	(mm)	(N)	(mm)	(Kg)	(Kg.m^2)	Positive (Nm / Deg)	Negative (Nm/Deg)
Renault 12	ϕ 170 K22 AX/490	120	50	06 - 08	3350	7.4 +0.5-0.3			150 / 6	70 / -6
Murat 131	ϕ 181.5 K22 AX/490	127	70	0.7 - 0.9	3300	7.7 +0.3-0.3	0.8	0.00211	146 / 6	88.5 / -6
Lada	ϕ 190 K22 AX/408	134	200	0.8 - 1.0	3750	7.7 +0.3-0.3			150 / 6	80 / -6
Tofaş	ϕ 190 X110Q-F490	134	200	0.8 - 1.0	4000	7.0 +0.3-0.3			150 / 14	108 / -10
Renault	ϕ 200 R32 QX/490	137	70	0.6 - 0.9	4000	7.6 +0.2-0.2	0.8	0.0024	191 / 6	127 / -6
Murat 131-Lada	ϕ 200 R33 AX/490	137	70	0.6 - 0.9	4100	7.7 +0.3-0.3			191 / 6	127 / -6
Ford Transit	ϕ 215 M(D93) 32AX/408	145	50	0.75 - 1.1	5000	7.7 +0.3-0.3			237 / 17	190 / -13

Table E.2 Diaphragm Spring Parameters of Some Valeo Clutcl

			DIAPHRAGM SPRING				
VEHICLE	CODE	Outer Diameter	Inner Diameter	Mass	Moment of Inertia	Release Load/Dist	Charge Load/Dist
		(mm)	(mm)	(kg)	(kg m^2)	(N / mm)	(N / mm)
Murat 124	180 CPO 3100	180.000	130.000	2.730	0.018	1100-1650 / 8.5	3100 / 1.5
	200 CPO 3500	200.000	148.000	3.540	0.027	1300-1500 / 8.5	3500 / 1.5
Tofas Sahin, Dogan	190 CP 4500	190.000	132.000	2.980	0.021	1450-1850 / 8.5	4250-4500 / 1
Tempra, Kartal	190 CPR 4000	190.000	132.000	2.520	0.019	1250-1800 / 8	4000 / 1,3
Renault	200 CPO 4000	200.000	148.000	3.230	0.026	1200-1600 / 8	4000 / 1
12 / 19 / 21	180 CP 3300	180.000	130.000	2.730	0.018	1200-1480 / 8	3300 / 1,7
	200 CPO 3500	200.000	148.000	3.540	0.027	1050-1500 / 8	3500 / 1,4
	170 DB 3000	170.000	120.000			1250-1420 / 7,5	4000 / 1,5
LADA	190 CP 4000	191.500	126.000			1450-1650 / 8	4000 / 1.5
FORD TRANSIT	215 DBR 4850	215.000	154.000	4.300	0.041	1200-2000 / 8	4850 / 1,5

F. ENGINE TORQUE CHARACTERISTICS OF JEEP CHEROKEE 4.0 L GASOLINE ENGINE

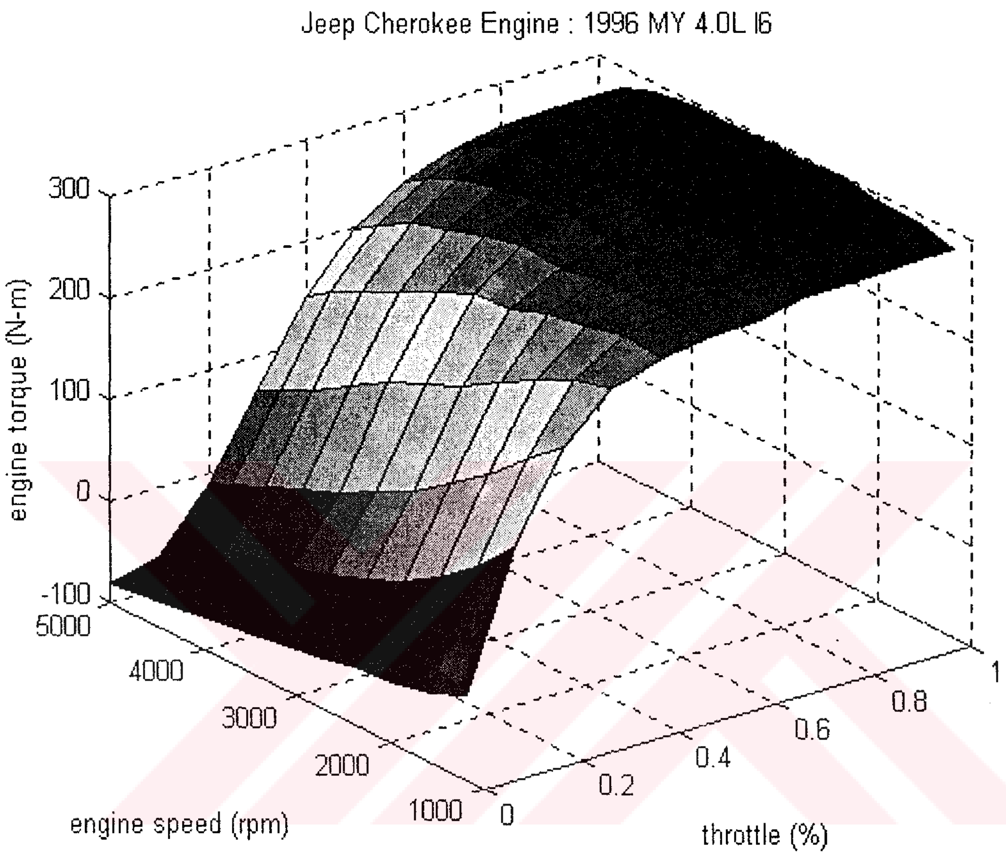


Fig. F.1 Engine Torque Characteristics of Jeep Cherokee 4.0 L Gasoline Engine

Table F.1 Numerical Torque Values of Jeep Cherokee 4.0 L Gasoline Enç

Torque Characteristics of Jeep Cherokee 4.0L

Throttle Level(%)	800	1200	1600	2000	2400	2800	3200	3600	4000	4400	4750	5000 RPM
10	153.13	114.9142	76.7044	48.9389	26.7346	9.0608	0	0	0	0	0	0
20	233.83	201.2355	168.6412	136.5217	108.7290	83.5678	64.6460	44.1779	28.8913	12.4518	0	0
30	254.45	244.1520	233.8569	215.5591	193.9516	171.7202	156.9626	139.4379	117.5863	95.6669	78.9967	66.6128
40	255.32	259.4522	263.5892	253.9452	248.1534	234.5351	229.8284	215.1250	195.5522	178.4751	161.4116	146.4912
50	259.15	265.7323	272.3109	268.9741	267.8483	261.1070	266.4241	256.5359	242.8363	231.5510	213.9857	198.6719
60	262.03	268.7842	275.5391	273.9657	276.9362	272.5686	281.6700	277.1532	270.0185	261.8801	246.8784	230.6423
70	268.12	277.7772	287.4347	289.4015	294.0540	276.1630	290.0797	286.7837	283.9352	277.9806	263.8198	248.5060
80	268.53	278.4554	288.3842	289.3744	296.1157	292.4941	291.8159	293.1994	289.9305	285.3052	272.5279	257.5532
90	271.71	282.1719	292.6297	293.0095	298.7064	295.1391	296.5633	295.7495	293.6606	290.3103	276.5700	261.1070
100	274.65	283.3384	292.0329	292.1550	299.5067	295.7359	297.2686	295.9936	295.4239	290.8122	278.8487	261.7852

** Torque Values Are Given in Nm.*