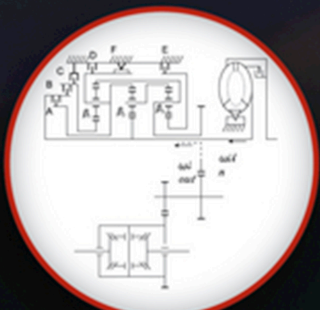
AUTOMOTIVE SERIES

AUTOMOTIVE POWER TRANSMISSION SYSTEMS

Yi Zhang
Chris Mi



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Automotive Power Transmission Systems

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Series Preface

Automotive power transmission systems are critical elements of any automobile. The ability to transmit power from the engine of a vehicle to the rest of the drive train is of primary importance. Furthermore, the design of power transmission systems is of critical importance to the overall vehicle system performance, as it affects not only performance characteristics such as torque and acceleration, but it also directly affects fuel efficiency and emissions. The power transmission system also presents one of the most complex design tasks in the overall automotive systems design and integration because it must interface with a variety of power plants such as internal combustion, electric, and hybrid plants. This is further complicated by the fact that engineers must consider a variety of transmission designs such as manual, automatic, and continuously variable systems. Furthermore, all of these elements must be condensed into the smallest, lightest package possible while functioning under significant loads over long periods of time.

Automotive Power Transmission Systems presents a thorough discussion of the various concepts that must be considered when designing a power transmission system. The book begins with an excellent discussion of how a transmission is designed by matching the engine output and the vehicle performance via proper transmission ratio selection. It then proceeds to discuss the basics of manual transmission and the analysis and design of essential transmission subsystems and components such as the gears, torque converter, and clutches. The authors then discuss more advanced transmission types such as dual clutch transmissions, continuously variable transmissions and automatic transmissions. In the final chapters, advanced control concepts for transmissions are presented, leading to the final chapters on electric and hybrid powertrains. This powerful combination of concepts results in a text that has both breadth and depth that will be valued as both a classroom text and a reference book.

The authors of *Automotive Power Transmission Systems* have done an excellent job in providing a thorough technical foundation for vehicle power transmission analysis and control. The text includes a number of clearly presented examples that are of significant use to the practicing engineer, resulting in a book that is an excellent blend of practical applications and fundamental concepts. The strength of this text is that it links a number of fundamental concepts to very pragmatic examples, providing the reader with significant insights into modern automotive power transmission technology. The authors have done a wonderful job in clearly and concisely bringing together the significant breadth of technologies necessary to successfully implement a modern power transmission system, providing a fundamentally grounded book that thoroughly explains power transmissions. It is well written, and is authored by recognized experts in a field that is critical to the automotive sector. It provides an excellent set of pragmatic and fundamental perspectives to the reader and is an excellent addition to the Automotive series.

Thomas Kurfess
January 2018

Preface

Automotive power transmission systems deliver output from the power source, which can be an internal combustion engine or an electric motor or a combination of them, to the driving wheels. There are many valuable books and monographs published for internal combustion engines (ICE), but only a few can be found in the public domain, as referenced in this book, that are specifically written for automotive transmissions. Technical publications by the Society of Automotive Engineers (SAE) in transmissions are mostly for conventional ICE vehicles and are basically collections of research papers that are aimed at readers with high expertise in transmission sub-areas. The purpose of this book is to offer interested readers, including undergraduate or graduate students and practicing engineers in the related disciplines, a systematic coverage of the design, analysis, and control of various types of automotive transmissions for conventional ICE vehicles, pure electric vehicles, and hybrid vehicles. The aim is that this book can be used either as a textbook for students in the field of vehicular engineering or as a reference book for engineers working in the automotive industry.

The authors have taught a series of courses on powertrain systems for both ICE and electric-hybrid vehicles over many years in the graduate programs of mechanical engineering, electrical engineering, and automotive systems engineering at the University of Michigan-Dearborn. The lecture notes of these courses form the framework for the book chapters, the main topics of which are highlighted below.

The book starts with automotive engine matching in Chapter 1, which covers the following technical topics: output characteristics of internal combustion engines, vehicle road loads and acceleration, driving force (or traction) and power requirements, vehicle performance dynamics and fuel economy, and transmission ratio selection for fuel economy and performance. The formulation and related analysis in Chapter 1 on road loads, performance dynamics, and powertrain kinematics are applicable to all vehicles driven by wheels and will be used throughout the book.

Chapter 2 covers manual transmissions, focusing on gear layouts, clutch design, synchronizer design, and synchronization analysis. Detailed analysis is provided on the operation principles of synchronizers and on the synchronization process during gear shifts. Example production transmissions are used as case studies to demonstrate principles and approaches that are then generally applicable.

For readers' convenience, Chapter 3 provides the basics of the theory of gearing and gear design with specific application to manual transmissions (MT). With example transmissions, the chapter details geometry design, gear load calculation, and gear strength and power ratings for standard and non-standard gears using existing equations or formulae from AGMA standards. The chapter also includes a separate section on the kinematics of planetary gear trains which are widely applied in automatic transmissions (AT). Readers are strongly recommended to read this section before reading Chapters 5 and 6.

Chapter 4 covers the structure, design, and characteristics of torque converters, focusing on torque converter operation principles, functionalities, and input–output characteristics. Methods for the determination of engine–converter joint operation states are presented in detail. The chapter also deals with the modeling of the combined operation of the entire vehicle system that consists of the engine, torque converter, automatic transmission, and the vehicle itself.

Chapters 5 and 6 can be considered as the core of the book, as these two chapters present the design, analysis, and control of conventional automatic transmissions (AT) which are typically designed with planetary gear trains. Chapter 5 focuses on how multiple gear ratios are achieved by different combinations of clutches and planetary gear trains. A systematic method will be presented in this chapter for the design and analysis on the gear ratios and clutch torques of automatic transmissions. The chapter also gives an in-depth analysis of the dynamics of automatic transmissions during gear shifts and the general vehicle powertrain dynamics in a systematic approach, using an eight-speed production automatic transmission as the example in the case study.

Chapter 6 concentrates on hardware and software technologies of both component and system levels which are applied in the control systems for the implementation of transmission functionalities. The chapter begins with the functional descriptions of the hardware components, including hydraulic components, electronic sensors, and solenoids. The chapter then presents transmission control system configurations and the related design guidelines. Examples based on the production transmissions of previous and current generations are used to demonstrate the operation logic and functions of the control systems. A specific section is devoted to present concurrent transmission control technologies commonly applied in the automotive industry. This focuses on the accurate clutch torque control during gearshifts and torque converter clutch actuation. The chapter ends with the identification of control variables and control system calibration.

Chapter 7 mainly presents the design and control of belt type continuously variable transmissions (CVT), starting with the structural layouts of CVT systems and key components, including the basic CVT kinematics and operation principles. Topics are concentrated on force analysis during the CVT's operations, and the mechanisms for torque transmission and ratio changes. The chapter provides details of control system design and the analysis of the control of ratio changing processes. CVT system control strategies, including continuous ratio control, stepped ratio control, and system pressure control are also presented.

The design and control of dual clutch transmissions (DCT) are covered in Chapter 8. The chapter concentrates on the dynamic modeling and analysis of DCT operations, including DCT vehicle launch and shifts. DCT control system design, and shift and launch control processes are included here, and the chapter also dedicates a specific section to DCT clutch torque formulation during launch and shifts, using an electrically actuated dry DCT as the example in the case study.

Chapter 9 covers power train systems for pure electric vehicles (EV). It includes several key technical topics: design optimization and control of electric machines for EV applications, power electronics for electric power transmission and inverter design, and system control under various operation modes. The chapter also includes a section on mechanical transmissions with a fixed ratio or two ratios which are specifically designed for pure electric vehicles. Two-speed or multi-speed automated gear boxes enable EV driving motors to operate within the speed range for optimized efficiency and performance.

Finally, hybrid powertrain systems are discussed in Chapter 10, which presents various hybrid powertrain configurations including series, parallel, and complex architectures. It provides detailed analysis of the operation modes and operation control for hybrid vehicle powertrain systems. Production hybrid vehicles are used as case studies in mode analysis and operation control.

As highlighted above, each chapter of the book is dedicated to a specific transmission, and readers may choose the chapter of interest to read. If the book is used as a textbook, the course syllabus can follow the order of the chapters. If the book is used as a reference, readers with transmission expertise may just choose the chapter of interest, and those readers without broad expertise may wish to first read Chapters 1 and 4 and then read the chapter of interest.

The authors would like to express their hearty thanks for the help received from friends and colleagues in preparing the manuscript. We would like to thank especially Prof. Qiu Zhihui of Xian Jiaotong University and Prof. He Songping of Huazhong University of Science and Technology for their help in drawing the pictures for this book. We also want to thank the publisher, John Wiley & Sons, for giving us the opportunity to publish this book, and we dedicate our deep appreciation to Ms Ashmita Rajaprathapan for her invaluable contributions in editing and finalizing the book. Lastly and most importantly, the authors would like to express their thanks to engineers, scholars, and researchers who have contributed to the technologies of vehicular power transmission systems and whose work may or may not have been specifically acknowledged in the reference lists.

Yi Zhang and Chris Mi

1

Automotive Engine Matching

1.1 Introduction

Internal combustion engines have been the primary power source for automotive vehicles since the beginning of the automotive industry. Although automobiles powered by electric motors have entered the automotive market and are likely to grow in market share, the vast majority of vehicles will still be powered by internal combustion engines in the foreseeable future. This is partly due to the bottleneck in the development of key technologies for electric vehicles, such as battery energy density, durability and charging time, and the lack of infrastructure and facilities necessary for the daily use of electric vehicles. On the other hand, proven crude oil reserves can still fuel internal combustion engines for decades to come.

Modern internal combustion engines are sophisticated systems that integrate synergistically mechanical, electrical, and electronic subsystems. Engine technologies are subjects of study in great breadth and depth in the areas of combustion, heat transfer, mechanical design and manufacturing, material engineering, and electronic control [1,2,3,4]. However, this book does not cover engines themselves and is concerned only with how the engine outputs are transmitted to the driving wheels. Readers interested in engine topics are directed to the books referenced here or other related books. The engine outputs, in terms of power and torque, fuel economy, and emissions, are considered as given throughout the text of this chapter and indeed the whole book. Note that engine mapping data are highly proprietary and is usually not available in the public domain. Figures and plots pertaining to engine data in this book are mainly for illustration purposes and may not show the precise data of production engines.

The main topic of this chapter is the matching between the engine outputs and vehicle performance through the selection of transmission ratios. The chapter specifically covers: output characteristics of internal combustion engines; vehicle road loads and acceleration; driving force (or traction) and power requirements; vehicle performance dynamics and fuel economy; and transmission ratio selection. These topics are interconnected and are described in sequential order.

Although the chapter concerns automotive engine matching, as the title indicates, the formulation and related analysis of road loads, performance dynamics, and powertrain kinematics are applicable for all ground vehicles driven by wheels. The equations derived in this chapter will be referenced throughout the book wherever needed by the text.

Automotive Power Transmission Systems, First Edition. Yi Zhang and Chris Mi.

1.2 Output Characteristics of Internal Combustion Engines

The output of an internal combustion engine depends on its design, control, and calibration. Although computer simulation can be used to analyse engine output, engine mapping is the only experimental approach to obtain reliable engine output data. For a given production engine, its output data are provided in terms of power and torque, as well as specific fuel consumption and emissions.

1.2.1 Engine Output Power and Torque

The operation status of an internal combustion engine is defined by its crankshaft rotational speed and the output torque from its crankshaft. The output torque and power depend on the throttle opening and the engine speed, i.e. the crankshaft rotational speed in RPM. It should be noted that the output torque and output power are not independent since power is the product of torque and angular velocity. The torque map of a typical IC engine is shown in Figure 1.1, where the two horizontal axes are respectively the throttle opening as a percentage of the wide open throttle (WOT) or as a degree of throttle angle and the engine speed in RPM. The vertical axis shows the engine output torque in foot pounds in the imperial standard or in newton-meters in the international standard (SI). Without considering the engine transient behavior, the engine static output torque can be found from Figure 1.1, usually by numerical interpolation, for a given set of engine speed and throttle opening. This is the torque as a load at which the engine reaches dynamic equilibrium at the specified engine speed and the throttle opening.

In practice, the engine output torque is often plotted as a curve against the engine speed for specific throttle openings as shown in Figure 1.2, where the throttle opening for each torque curve is represented as a percentage of the wide open throttle angle. Clearly, the engine output torque is a function of engine RPM for a given throttle opening and there is a torque vs RPM curve for each throttle opening. Figures 1.1 and 1.2 provide the same output torque data and are just drawn for convenience of reference.

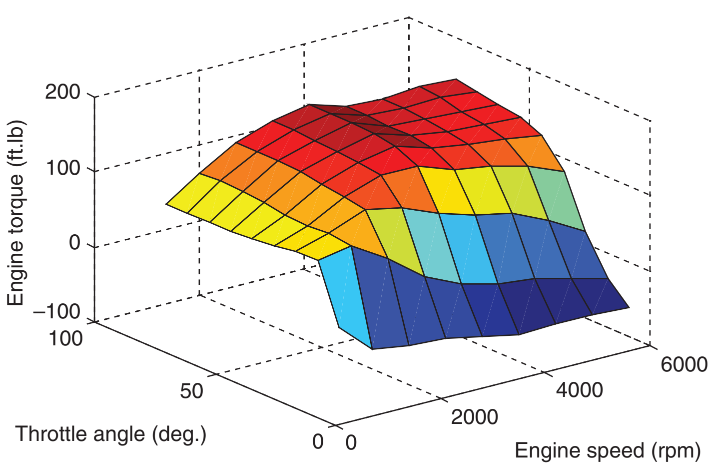

Figure 1.1 Engine output torque map.

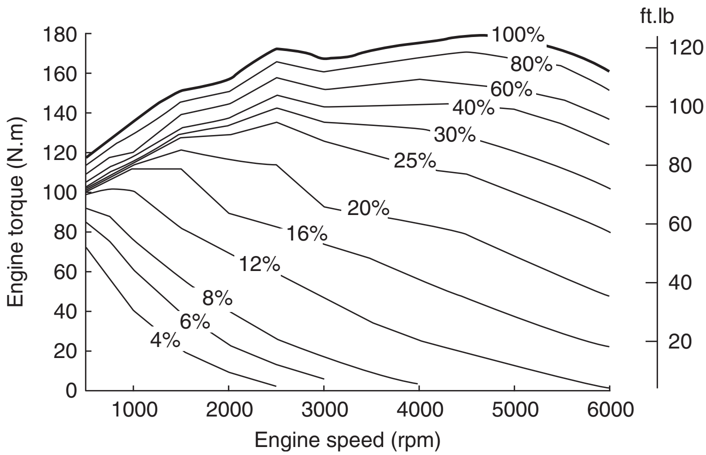

Figure 1.2 Engine torque curves for various throttle openings.

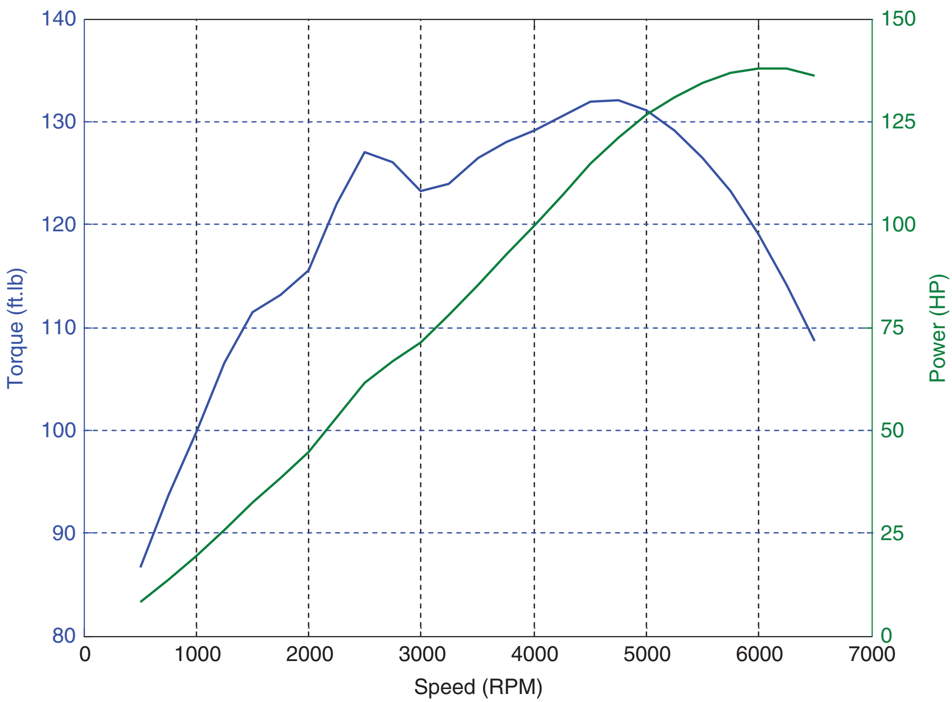

Figure 1.3 Engine torque and power at wide open throttle.

Apparently, the engine capacity torque output or the power output is achieved at the wide open throttle (WOT), as shown in Figure 1.3. It should be noted that the maximum engine torque and the maximum engine power occur only at two separate RPM values on the WOT torque and power curves. The two popularly referred engine performance

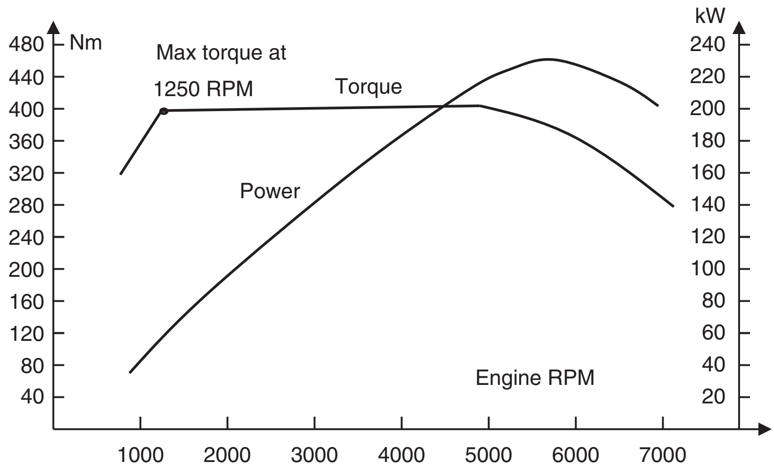

Figure 1.4 Typical torque curve of turbo engines.

specifications, engine power and engine torque, are actually the peak values for the power and torque on the WOT output plot. As can be observed in Figure 1.3, internal combustion engines provide stable power output within a range of engine rotational speed, defined by the so-called idle RPM and redline RPM. Below the idle, the engine does not run stably but stalls, without being able to provide any usable output. On the other side, running the engine beyond the redline speed may cause excessive damage to the engine.

The shape of the torque curve in the operation range defined by the idle and redline is characteristic of the IC engine, depending on its design, fuel injection method, control, and calibration. As an example, the torque curve in Figure 1.3 has a local peak at around 2500 RPM and the maximum engine torque occurs around 4800 RPM. The engine power increases from the idle point almost linearly up to a peak at around 6100 RPM.

In general, the torque curves for naturally aspirated engines can be categorized as rising and buffalo shaped [5], while for turbo-charged engines, the torque curves are flat from a certain low RPM up to a relatively high RPM, as shown in Figure 1.4. Using turbo technology, the maximum output torque can be increased by more than 50% for the same engine displacement. To make things better, this maximum torque becomes available at a much lower RPM in comparison to naturally aspirated engines and stays flat up to a high RPM. This provides the vehicle with much better acceleration performance, especially at low vehicle speed. Turbo engines with small displacements provide outputs in torque and power equivalent to those of naturally aspirated engines of much larger displacements but consumes less fuel. Because of these advantages, vehicles powered by turbo engines are increasingly popular and represent the trend in the automotive industry.

1.2.2 Engine Fuel Map

Engine fuel efficiency is a top performance specification in today's automotive industry. The fuel consumption data of internal combustion engines are indispensable for the design, operation and control of vehicle powertrain systems. These data are experimentally obtained by intensive engine mapping and are usually provided for a production

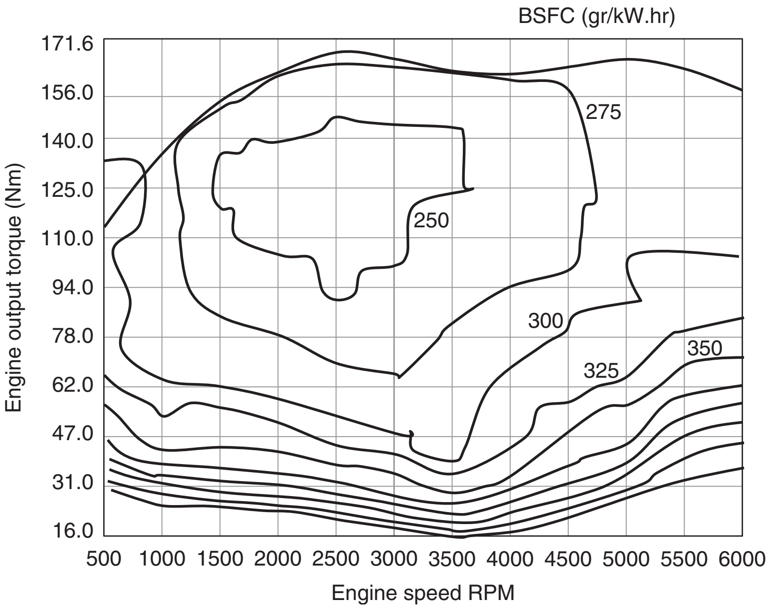

Figure 1.5 Engine specific fuel consumption map.

engine as a fuel map, which indicates in contour plots the specific fuel consumption of the engine at a given operation status, as shown in Figure 1.5. The specific fuel consumption is the amount of fuel that the engine needs to burn in order to do one horsepower hour of work, either in litres, grams, or pounds of weight.

The horizontal and vertical axes in Figure 1.5 are respectively the engine RPM and the engine output torque that define the engine operation status. The numbers by the contours are the specific fuel consumption in gram per kilowatt-hour (grams of fuel the engine consumes for it to do one kilowatt-hour of work). For example, if the engine runs stably at 4000 RPM with an output torque of 94 Nm, the specific fuel consumption is 275 gr/kW.hr. At the operation status defined by the RPM and the torque, the engine power is 39.37 kW. If the engine runs at the status continuously for one hour, the fuel consumed by the engine will then be 10.82 kg. The engine fuel consumption along a contour is the same even though the operation status is different, so the hourly fuel consumption of the engine is also 10.82 kg if it runs stably at 4740 RPM with an output torque of 125 Nm. Apparently, the engine will be more fuel efficient if it runs along the contour with 250 gr/kW.hr. It can also be observed that the most fuel efficient operation status is near the point with 2500 RPM and 125 Nm. When a vehicle is driven on a road at constant speed, the engine operation status depends on the road load, the vehicle speed, and the transmission gear ratio, the last determining the engine RPM and torque at a given vehicle speed and thus largely affects the vehicle fuel economy.

1.2.3 Engine Emission Map

When internal combustion engines generate power to propel ground vehicles, unwanted pollutants, harmful to the environment and to human health, are emitted in the process of combustion. These pollutants include CO, CO_2, NO_X, and other harmful gasses or particulate matter. The standard on emission control is increasingly stringent in the automotive industry due to environmental and human health concerns. Engine technologies, especially the technologies for combustion control and after-combustion treatment, are the key to minimizing the emission of pollutants. Transmissions also contribute to lowering vehicle emission levels by keeping the engine running in more efficient and less polluting operating ranges.

Engine emission maps are even more difficult to obtain than fuel maps because the quantity of a pollutant under various operation conditions is hard to measure. Computer simulation can be used to analyse engine emissions, but reliable emission data can only be obtained experimentally through extensive tests. Engine emission maps are provided for a given engine in formats similar to engine fuel maps. The specific quantity of a particular pollutant emitted by the engine is interpolated from the emission contour for a given engine operation status. Using the emission maps of the engine, the amount of emission of a pollutant can be simulated for a specified drive range.

1.3 Road Load, Driving Force, and Acceleration

Various forces are applied to a vehicle when it travels on a road surface. These forces include gravity, wheel–road contacting forces, road load, and driving force, which is also termed traction. Road load is against the motion of the vehicle, while traction force, or driving force, propels the motion of the vehicle. The driving force of a vehicle originates from the engine via the transmission and is fundamentally limited by the road traction limit. The total road load is the resultant of three separate road loads: rolling resistance, grade load, and air resistance. Figure 1.6 is the free body diagram of a vehicle of weight W that is being accelerated uphill.

In the free body diagram, v and a are the vehicle speed and acceleration respectively. R_A is the air drag or air resistance. The air drag is a distributed load, but for simplicity, it is assumed to be a point load acting at height h_A. R_F and R_R are the rolling resistance from

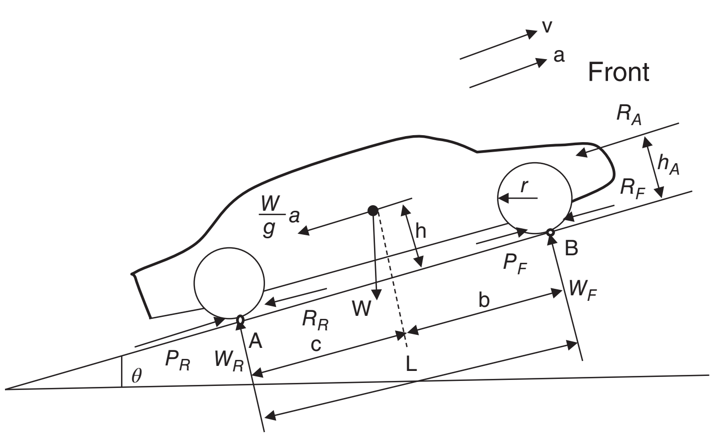

Figure 1.6 Free body diagram of a vehicle accelerated uphill.

the front and rear wheels respectively. P_F and P_R are the driving force from the front and rear wheels respectively. W_F and W_R are the axle loads, which are respectively the contact force between the front wheels and the road surface and between the rear wheels and the road surface. A and B denote the points of contact between the wheels and the road surface. θ is the grade angle of the slope and r is the rolling radius of the tire. The height of the center of gravity and the height of the air resistance are respectively denoted as h and h_A. For passenger cars, these two heights are assumed to be the same. The vehicle wheelbase is L and the longitudinal position of the gravity center is determined by b and c, which is the distance from the gravity center to the front axle and rear axle respectively. Unless otherwise stated, the US customary unit system will be used in the equations, where forces are in pounds, linear dimensions are in feet, speed is in ft/s, and acceleration is in ft/s^2. It should be noted that the inertia force $\gamma^{W}/_{g}\, a$ in the free body diagram is in the opposite direction to the acceleration, based on the D'Alembert's principle. γ is the equivalent mass factor that is introduced to account for the mass moments of inertia of all rotational components in the powertrain, including transmission input and output shafts, gears in the power flow path, drive shaft, differential, wheels, etc. The value of γ can be accurately determined based on the total vehicle equivalent kinetic energy as follows,

$$\frac{1}{2}\gamma\frac{W}{g}v^2 = \frac{1}{2}\frac{W}{g}v^2 + \sum_{i=1}^{n}\frac{1}{2}J_i{\omega_i}^2 \tag{1.1}$$

In the equation above, J_i is the mass moment of inertia of each rotational component and n is the total number of rotational components in the powertrain. The equivalent mass factor is then determined as,

$$\gamma = 1 + \sum_{i=1}^{n}\frac{gJ_i}{W}\left(\frac{\omega_i}{v}\right)^2 \tag{1.2}$$

For a given vehicle, the ratio $\frac{\omega_i}{v}$ is a constant for each rotational component that depends on the transmission gear ratios. Empirical formula and tables are available for the approximation of the equivalent mass factor [6]. For passenger cars, the value of γ is small and can be considered to be equal to one for vehicle acceleration analysis and transmission ratio selections.

1.3.1 Axle Loads

The forces in the free body diagram (Figure 1.6) form a system of equilibrium, and three scalar equations can be written based on the condition of equilibrium. As shown below, the first two equations are based on the conditions that the sum of moments made by all forces about point A and point B must be equal to zero. The third equation is that the sum of all forces, including the inertia force, must be equal to zero in the direction of vehicle motion.

$$\begin{aligned} \sum M_A &= 0 \\ \sum M_B &= 0 \\ \sum F &= 0 \end{aligned} \tag{1.3}$$

These equations can be arranged to express the axle loads and the inertia force as follows:

$$W_F = \frac{1}{L}\left(Wc\cos\theta - R_A h_A - \gamma\frac{W}{g}ah - Wh\sin\theta \right) \quad (1.4)$$

$$W_R = \frac{1}{L}\left(Wb\cos\theta + R_A h_A + \gamma\frac{W}{g}ah + Wh\sin\theta \right) \quad (1.5)$$

$$\gamma\frac{W}{g}a = P_F + P_R - R_F - R_R - R_A - W\sin\theta \quad (1.6)$$

The first two equations determine the dynamic axle weights for the vehicle. During acceleration, there is a weight transfer equal to the magnitude of the inertia force from the front axle to the rear axle, as shown in Eqs (1.4) and (1.5). The static axle weights on level ground are obtained from the equations by making the slope angle θ, the air drag R_A, and the acceleration a equal to zero. It should be noted that tractions are available from both front and rear wheels only for a four wheel drive vehicle. P_R is zero for front wheel drive and P_F is equal to zero for rear wheel drive. Total driving force and rolling resistance from both front wheels and rear wheels are:

$$\begin{aligned} P &= P_F + P_R \\ R &= R_F + R_R \end{aligned} \quad (1.7)$$

The rolling resistance depends on many factors, such as tire material, texture, tread, inflation, speed, etc. Accurate calculation of the rolling resistance is very difficult, indeed impractical. For simplicity, it is common practice in the automotive industry to calculate the rolling resistance by:

$$R = fW\cos\theta \approx fW \quad (1.8)$$

where f is the rolling resistance coefficient and is approximately equal to 0.02. By rearranging Eqs (1.4–1.7) with the assumption that $h \approx h_A$, the axle loads can be solved in the following form:

$$P - fW = \gamma\frac{W}{g}a + R_A + W\sin\theta \quad (1.9)$$

$$W_F = \frac{Wc}{L} - \frac{h}{L}(P - fW) \quad (1.10)$$

$$W_R = \frac{Wb}{L} + \frac{h}{L}(P - fW) \quad (1.11)$$

where $\frac{c}{L}$ and $\frac{b}{L}$ are the weight distribution factors. The term $\frac{h}{L}(P - fW)$ is the dynamic weight transfer. Eqs (1.10) and (1.11) represent the dynamic axle weights in terms of the static axle weights and the weight transfer. The dynamic axle weight on the driving axle determines the maximum traction available for the vehicle under a given road condition.

1.3.2 Road Loads

There are three kinds of road loads that are against vehicle motion when the vehicle travels on a road surface: rolling resistance, air drag, and grade load, as shown in Figure 1.6.

The rolling resistance is calculated by Eq. (1.8). The grade load is the component of gravity on the slope direction and is equal to $W\sin\theta$. At level ground, only rolling resistance and air drag exist. At high vehicle speed, the air drag becomes more significant than the rolling resistance.

There are two causes for the generation of air resistance: friction between the air and the vehicle body surface; and air turbulence formed around the vehicle body [6]. The latter is the main cause of air drag for ground vehicles. Factors affecting the magnitude of the air drag include the shape and finish of vehicle body, the vehicle frontal projected area, air density and atmospheric condition, and most importantly, the vehicle's speed. It is very challenging to exactly determine the air drag by analytical means. In the standard of the Society of Automotive Engineers (SAE), the air resistance or air drag is calculated by the following formulation [6]:

$$R_A = 0.26 C_D A \left(\frac{v}{10}\right)^2 \tag{1.12}$$

where C_D is the unit less air drag coefficient that mainly depends on vehicle body shape and body surface smoothness. The air drag coefficient can be determined with high accuracy by wind tunnel testing. Modern passenger cars with streamlined body can have an air drag coefficient as low as 0.26. A is the vehicle frontal projected area in ft^2 that mainly depends on the vehicle size. This is the area of the vehicle body that confronts the air flow in the direction perpendicular to vehicle motion. To determine this area, a flat board can be held perpendicular to the road surface behind the parked vehicle and a flashlight is then used to beam the vehicle body horizontally in front of the vehicle. The area of the shadow casted on the board is the frontal projected area. As shown in the formulation, the air drag is proportional to the square of the vehicle speed v relative to the wind. With the speed v in mph, the formulation determines the air drag as a force in pounds. For the analysis and calculations of vehicle dynamics, the vehicle speed and acceleration are often in ft/s and ft/s^2, then the formulation for air drag will be used in the following form:

$$R_A = 0.00118 C_D A v^2 \tag{1.13}$$

The equation above is directly transformed from Eq. (1.12) by considering that one mph is equal to 1.467 ft/s. The resultant road is the sum of the rolling resistance, grade load and air drag, as expressed below,

$$R = 0.00118 C_D A v^2 + fW + W\sin\theta \tag{1.14}$$

1.3.3 Powertrain Kinematics and Traction

There are various layouts for vehicle powertrain systems. In this section, the powertrain of a rear wheel drive vehicle with a manual transmission (MT), in the layout shown in Figure 1.7, is used as the example for demonstration. The analysis on powertrain kinematics and related equations derived in the example are applicable to all other powertrain layouts.

The engine output torque and output angular velocity are denoted as T_e and ω_e respectively, the transmission output torque and angular velocity are denoted as T_t and ω_t, the

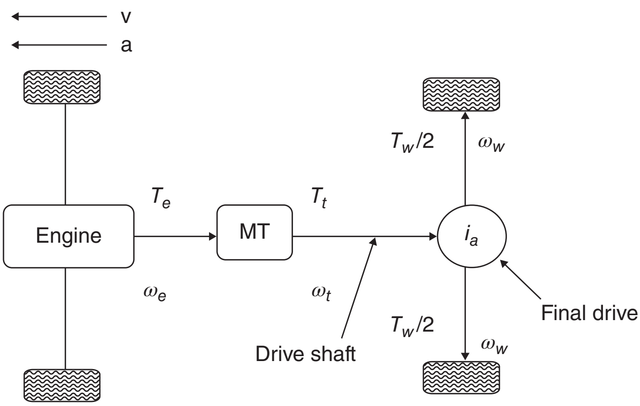

Figure 1.7 Layout of RWD manual transmission powertrain.

angular velocity of the driving wheel is denoted as ω_w, and the torque on each of two wheels on the driving axle is denoted as $\frac{T_w}{2}$. As shown in Figure 1.7, the engine output is transmitted through the transmission to the drive shaft, which then transmits the transmission output to the final drive. The final drive assembly contains a pair of spiral bevel or hypoid gears that multiplies the transmission output torque by the final drive ratio i_a and transmits the rotation of the drive shaft to the wheels between the two perpendicular axes. The outputs of the engine and the transmission are related by the following equation:

$$T_t = \eta_t i_t T_e \tag{1.15}$$

$$\omega_t = \frac{\omega_e}{i_t} \tag{1.16}$$

where i_t is the transmission ratio defined as the division of the input angular velocity by the output angular velocity and η_t is the transmission efficiency. The transmission ratio is a stepped variable for manual transmissions. For a five speed transmission, the ratio varies between the highest value in the first gear and the lowest value in the fifth gear. Note that the accurate determination of the transmission efficiency is experimental by nature, and the efficiency is assumed to be a constant value throughout the text of this book.

The transmission output is further transmitted by the final drive to the driving wheels. The torque on the two driving wheels and the angular velocity of the driving wheels are related to the engine output torque and engine angular velocity by the following equations:

$$T_w = \eta_a i_a T_t = \eta_a \eta_t i_a i_t T_e \tag{1.17}$$

$$\omega_w = \frac{\omega_t}{i_a} = \frac{\omega_e}{i_a i_t} \tag{1.18}$$

where i_a is the final drive ratio and η_a is the final drive efficiency. When a vehicle travels on road, there is always a small amount of slippage between the tire and the road surface. The amount of tire spillage is determined by the tire slip rate defined as:

$$\delta = \frac{\omega r - v}{\omega r} \tag{1.19}$$

Under normal driving conditions, the slip rate is very small ($\delta \le 0.02$) and can be neglected for the selection of gear ratios in engine–transmission matching. If the slippage is not considered, then vehicle motion and driving wheel rotation are related as,

$$\omega_w = \frac{v}{r}$$
$$\alpha_w = \frac{a}{r} \tag{1.20}$$

where α_w is the angular acceleration of the driving wheels. Based on Eqs (1.17), (1.18), and (1.20), the engine output torque, the engine angular velocity, the torque on the driving wheels, and the vehicle speed are related by the following equations,

$$\begin{cases} T_w = \eta i_a i_t T_e \\ \omega_e = i_a i_t \dfrac{v}{r} \end{cases} \tag{1.21}$$

where η is the overall powertrain efficiency. The second of these equations relates the engine angular velocity and the vehicle speed without considering tire slippage. Knowing the torque on the driving wheels, the driving force that originates from the engine and propels the vehicle is then determined by the following equation for the rear wheel drive (RWD) vehicle in the example:

$$P = P_R = \frac{\eta i_a i_t T_e}{r} \tag{1.22}$$

The maximum value of the driving force for ground vehicles driven by wheels is limited by road–tire contact conditions and is commonly termed as the traction limit. For the RWD vehicle in question, the traction is limited by the following inequality:

$$P = P_R \le \mu W_R \tag{1.23}$$

where μ is the so-called traction coefficient, which depends on the road condition and the tire properties. On a standard highway surface (road surface with skid number 81), the value of the traction coefficient is equal to 0.81. The traction force or driving force available from the powertrain cannot exceed the traction limit, or the driving wheels will slip excessively. W_R is the dynamic rear axle load, which is determined by Eq. (1.11). It is emphasized here that Eqs (1.15–1.23) are derived for the RWD vehicle in the example, and are applicable for all other types of powertrain systems as mentioned previously. With the driving force determined by Eq. (1.22), the vehicle acceleration can then be determined from Eq. (1.16) by:

$$a = \frac{P - (fW + W\sin\theta + 0.00118 C_D A v^2)}{\gamma \frac{W}{g}} \tag{1.24}$$

Eq. (1.9), or Eq. (1.24), is actually the equation of motion of the vehicle when it runs on a straight path. The vehicle acceleration can be calculated by Eq. (1.24) at any given vehicle speed if the engine throttle opening and the road condition are provided. The equivalent mass factor γ in Eq. (1.24) is approximately equal to one for passenger vehicles. The following example demonstrates how the equation series is used for the calculation of vehicle acceleration and fuel economy.

Example 1.1 A manual transmission used for a vehicle with data given below has six forward speeds and the gear ratios are: 1st gear (3.72), 2nd gear (2.31), 3rd gear (1.51), 4th gear (1.07), 5th gear (0.81), 6th gear (0.63). The engine WOT output is as given in Figure 1.3 and the fuel map is as given in Figure 1.5. The vehicle has the following data:

Front axle weight: 1750 lb	Rear axle weight: 1550 lb
Center of gravity height: 17 in.	Wheelbase: 104 in.
Air drag coefficient: 0.32	Frontal projected area: 22.90 sq.ft
Tire radius: 11.70 in.	Roll resistance coefficient: 0.018
Powertrain efficiency: 0.94	Traction coefficient: 1.0
Max. power @6000 RPM: 138 HP	Max. torque @4500 RPM: 132 ft.lb

a) The engine RPM drops by 662 (RPM) when a 4–5 upshift is made at a vehicle speed of 45 mph. Determine the final drive ratio of the vehicle.
b) Determine the engine torque and the engine power when the vehicle is cruising at a constant speed of 65 mph on level ground in the 6th and 5th gears respectively.
c) Determine the fuel economy in mpg or litres per 100 km for the conditions in (b).
d) Determine the maximum acceleration that the vehicle can achieve in 4th gear at a speed of 65 mph on a 2% slope.

Solution:

a) $$v = 45\,\text{mph} = 1.46(45) = 65.7\,\text{ft/s}$$

$$\omega = \frac{v}{r} = \frac{65.7}{11.70/12} = 67.38\,\text{rad/s} = 643.5\,\text{RPM}$$

The engine RPM drops by 662 in a 4–5 upshift, so $i_4 i_a \omega_w - i_5 i_a \omega_w = 662$, and:

$$i_a = \frac{662}{\omega_w(i_4 - i_5)} = \frac{662}{643.5(1.07 - 0.81)} = 3.96$$

b) $$v = 65\,\text{mph} = 65(1.46) = 94.9\,\text{ft/s} = 104\,\text{km/h}$$

$$R = fW + 0.00118 c_D A v^2 = 0.018(1750 + 1550) + 0.00118(.32)(22.9)(94.9)^2 = 137.3\,\text{lb}$$

$$\omega_w = \frac{v}{r} = \frac{94.9}{11.7/12} = 97.33\,\text{rad/s}$$

Since the vehicle speed is constant, engine power is the same for both gears and is equal to:

$$\frac{Rv}{\eta} = \frac{137.3(94.9)}{.94} = 13861.6\,\text{ft.lb/s} = \frac{13861.6}{550} = 25.2\,\text{HP} = 18.8\,\text{kW}$$

Engine torque depends on the gear ratio and is calculated respectively for the 6th and 5th gears,

$$\text{6th gear}: T_e^{(6)} = \frac{Rr}{\eta i_6 i_a} = \frac{137.3\left(11.7/12\right)}{.94(.63)(3.96)} = 57.1\,\text{ft.lb} = 76.5\,\text{Nm}$$

$$\text{5th gear}: T_e^{(5)} = \frac{i_6}{i_5} T_e^{(6)} = \frac{.63}{.81}(57.1) = 44.4\,\text{ft.lb} = 59.5\,\text{Nm}$$