Ground Vehicle Engineering Series

Driveline Systems of Ground Vehicles

Theory and Design

Alexandr F. Andreev
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Dedication

In fond and respectful memory of Dr. Anatoliy Kh. Lefarov,
engineer, scientist, and educator

My father, Anatoliy Khristoforovich Lefarov, was born on February 2, 1913, in a rural family of intellectuals. His father was an accountant and his mother was a teacher. From a young age, my father devoted his life to education and work. He graduated from a peasant's youth school in 1929 and from a technical school of agriculture in 1932. From 1934 onward, he worked as a grain-harvesting combine operator and as a driver of Caterpillar and International Harvester Farmall Tractors, first at the Evpatoria, and then at Simferopol farms that specialized in grain growing.

In recognition of the high quality of his professional work, the People's Agriculture Commissariat of the Crimean Autonomous Soviet Socialist Republic sent my father to Leningrad Industrial Institute (subsequently renamed as Leningrad Polytechnic Institute) in 1935, where he was successfully admitted.

Upon his admission to the institute, my father immediately devoted all his time and energy to his studies. His diligence, which consisted of studying 12–14 hour daily, did not fail to bear fruit. During the third year of his school, he became one of the best students in the institute and was awarded a higher-level stipend; he was later awarded the highest student honor at the time—the Stalin grantee.

Having graduated from the institute with honors in 1940, my father was assigned to work at the design department of the Gor'kiy Automotive Company (GAZ). He started his work under the guidance of one of the U.S.S.R. founders of the school of off-road equipment design—Vitaliy Andreyevich Grachev. It was precisely at this time that the direction of his scientific activity, which subsequently became his entire life purpose, germinated. Using his knowledge and through diligent work, he rapidly gained the respect of his coworkers and the leadership of the design office.

After Nazi Germany attacked the U.S.S.R. in 1941, my father was drafted and sent to the front. But he was soon recalled, as his engineering expertise was needed by the military, and he was returned to the GAZ design office. He continued working in this office donning the role of a senior designer. During the war, he designed and refined a number of mechanisms and assemblies of off-road wheeled and tracked vehicles for the military.

With the end of the war in 1945, my father, who was now a fully established expert, was transferred by the order of the Secretary of Automotive Industry of the U.S.S.R. to the Dnepropetrovsk Automobile Company (DAZ) as a deputy chief designer. Here fate again brought father together with his superior at the Gor'kiy Automotive Company, Vitaliy Andreyevich Grachev, who had been appointed as the chief designer of DAZ the previous year.

In DAZ, the following vehicles were designed under the guidance of V.A. Grachev with the direct participation of my father: the DAZ-150 4 ton self-loader, the DAZ-485 3-axle amphibian truck (LAT, large amphibian truck) for the military, the Ukraina passenger car, and other vehicles. But soon, by decree of the Soviet government the DAZ was switched to other manufacturing tasks and my father was sent to Minsk, the capital of Belarus, and appointed as the deputy chief design engineer at the Minsk Automotive Company (MAZ). In Minsk, he resumed his work with heavy-duty trucks, in particular, high-mobility trucks and tractors.

This was the time when the MAZ had just established a special design office (SKB-1 as further mentioned in the preface) for designing multiaxle rocket tractors. This office consisted of a large scientific, engineering, and manufacturing task force that subsequently not only established an entirely new direction in the design of military and civilian multi-wheel heavy-duty trucks but also became a prominent scientific and engineering school of the Soviet Union. This office was headed by Boris L'vovich Shaposhnik, a leading design engineer, and my father became his first deputy. The SKB-1 was established in 1954; the project of the base four-axis chassis MAZ-535 was already completed by 1955, and just a year later, the MAZ-537 tractor with a hydraulic gearbox, locked by a torque converter, lockable differentials, and independent suspension of all the wheels was also completed. In 1962, under the leadership and direct participation of my father, gear-type free-running differentials for heavy-duty MAZ tractors were designed. These differentials became an integral part of the driveline system of the well-known four-axle MAZ-543 chassis that went into production in 1962. The design of the differentials was so successful that they are still used on tractors that serve as carriers of various rocket launchers. Drs. Otto Ya. Zaslavskiy and Lev Kh. Gileles, who worked for many years with my father, write in their memoirs that my father had a sharp intellect, tact, and exceptional precision. This he most probably acquired from the old Russian engineering community and professors, some of whom remained in the Stalin years at Leningrad Polytechnic Institute, which was his alma mater. His colleagues made mention of the fact that he was the first one to point to the organic link between engineering developments and scientific studies, and was the first to call attention to the importance of intellectual property in modern society. He was the first staff member in the SKB-1 who was issued a certificate (Soviet equivalent of patent) for an invention. It is most likely that for these reasons my father performed experimental and analytic studies and wrote scientific works while being engaged in designing new vehicles. He authored articles and books and was the first in the SKB-1 to defend a PhD dissertation.

But father did not devote himself exclusively to military vehicles. He also worked actively on designing MAZ trucks, MAZ-501 and MAZ-509, and various modifications of the MAZ-537, for civil use. For example, he developed and put into production an original lightweight front axle for an all-wheel-drive timber carrier. The MAZ-501 automobile was the first automobile in the U.S.S.R. to employ a differential in the transfer case. This was a significant achievement for Dr. Lefarov as a designer. The locking differential designed and tested by my father under actual operating conditions started coming into use on MTZ-52 and MTZ-82 tractors of the Minsk Tractor Works, and also on the K-700 tractor of the Leningrad Tractor Company named after Kirov.

While working at the MAZ, my father, on the invitation of the administration of the College of Automobile and Tractor Engineering of Belarusian National Technical University (previously Belarusian Polytechnic Institute, BPI), Minsk, Belarus, became involved in teaching students—future experts in automobile and tractor engineering. He left his company in 1963 for the chair of the tractor engineering department of BPI.

In this institute, my father acquired students; he then established a school of study—a research group on multiwheel drive vehicles and driveline systems. His school investigated power distribution among the wheels connected with different types of driveline systems, and developed techniques of calculating the torque bias of self-locking differentials of different types. He also investigated the effect of many factors on the properties of locking and self-locking differentials.

The studies performed by father and his students were not restricted to two-axle automobiles and tractors, but concerned themselves with all multiaxle, many-wheel-drive vehicles. The main purpose of my father's school was, and still is, to find methods of optimizing the properties of systems of power distribution among the wheels and, in the final analysis, improving the overall mobility and other operational properties of wheeled vehicles operating under various road and off-road conditions.

As a result of the large volume of work performed at the SKB-1 and at the institute, my father defended his DSc dissertation (the highest degree in the U.S.S.R.) in 1976, and in 1977 he was conferred the rank of professor. For his contributions to the national machine-building industry, he was conferred the honorary title of Deserving Machine Builder of Belarus.

My father devoted all his life to work; more precisely, work was his entire life. He passed away on February 10, 1992, but left behind his scientific works, the automobiles and tractors that he designed, and, most importantly, his students who continue his work.

Dr. Victor A. Lefarov
Minsk, Belarus

Contents

Series Preface

Ground-vehicle engineering took shape as an engineering discipline in the twentieth century, and became the foundation for significant advancements and achievements, from personal transportation and agriculture machinery to lunar and planetary exploration. As we step into the twenty-first century faced with global economic challenges, there is a need to develop fundamentally novel vehicle engineering technologies, and effectively train future generations of engineers. The Ground Vehicle Engineering Series will unite high-caliber professionals from the industry and academia to produce top-quality professional/reference books and graduate-level textbooks on the engineering of various types of vehicles, including conventional and autonomous mobile machines, terrain and highway vehicles, and ground vehicles with novel concepts of motion.

The Ground Vehicle Engineering Series concentrates on conceptually new methodologies of vehicle dynamics and operation performance analysis and control, advanced vehicle and system design, experimental research and testing, and manufacturing technologies. Applications include, but are not limited to, heavy-duty multilink and pickup trucks; farm tractors and agriculture machinery; earthmoving machines; passenger cars; human-assist robotic vehicles; planetary rovers; military conventional and unmanned wheeled and track vehicles; and reconnaissance vehicles.

Dr. Vladimir V. Vantsevich
Series Editor

Preface

The dynamics and performance of a vehicle manifest themselves in the interaction of the vehicle with the surroundings and result from its properties such as energy/fuel efficiency, terrain mobility, tractive and velocity properties, vehicle turnability, stability of motion and handling, braking properties, and smoothness of ride. A distinctive feature in the design of vehicles with four or more driving wheels that is of great significance is that many of their properties depend markedly not only on the total power applied to all the driving wheels but also on the distribution of the total power among the wheels. Under given road or terrain conditions, the same vehicle with a constant total power at all the driving wheels, but with different power distributions among the driving axles and the left and right wheels of each axle, will perform differently; that is, the criteria of the above-mentioned vehicle properties will have different quantitative values. In practical engineering terms, this means that due to different power distributions among the driving wheels, a given vehicle will demonstrate variable fuel consumption, terrain mobility, and traction, and will accelerate differently and turn at different radii. Depending on the wheel power split, the vehicle can ''unexpectedly'' run into either understeering or oversteering and can sometimes become unstable, skidding in a lateral direction and eventually rolling over.

The power distribution among the driving wheels is largely determined by the vehicle's driveline system, which is generally defined as a part of the power train, located between the transmission and the driving wheels. A driveline system includes a set of mechanisms and subsystems which have been referred to in this book as power-dividing units (PDUs). Typically, a PDU has one input and two outputs. These units are employed in transfer cases, interaxle reduction gears, and driving axles. For a vehicle with one engine and with a conventional axle-type driveline system layout (which differs from the left–right side layout), the number of PDUs is equal to the number of the driving wheels less one. For example, a vehicle with four driving wheels will have three PDUs and a vehicle with eight driving wheels will have seven PDUs (see Figure 1).

It should be obvious to the reader that the number of combinations of mechanisms and subsystems that can be employed even in three PDUs of a vehicle with two driving axles is virtually limitless. In fact, a list of such mechanisms and subsystems may be compiled of open differentials and positively locked units, limited slip differentials with all kinds of torque biases, mechanically and electronically locked differentials, viscous couplings, NoSPINs, and also most current developments that are commonly referred to as torque-vectoring or torque-management systems.

This gives rise to two fundamental engineering problems: First, how to investigate the effect of different driveline systems on the properties of vehicles, their dynamics, and performance? Second, how to determine the optimal characteristics of the driveline system and its PDUs and then design them for a specific vehicle in a manner that would ensure a high level of dynamics and performance, mobility and fuel efficiency, traction and acceleration, and stability of motion and turnability?

Probably the first study of the effect of a driveline system on vehicle motion was the research paper of Prof. Nikolay E. Zhukovskiy, titled ''The theory of the instrument of engineer Romeyko-Gurko,'' published in 1903. The developments in the theory, design, and manufacture of vehicles with four or more driving wheels—all-wheel drive and multiwheel drive vehicles (see Section 1.2)—were to a large extent facilitated by

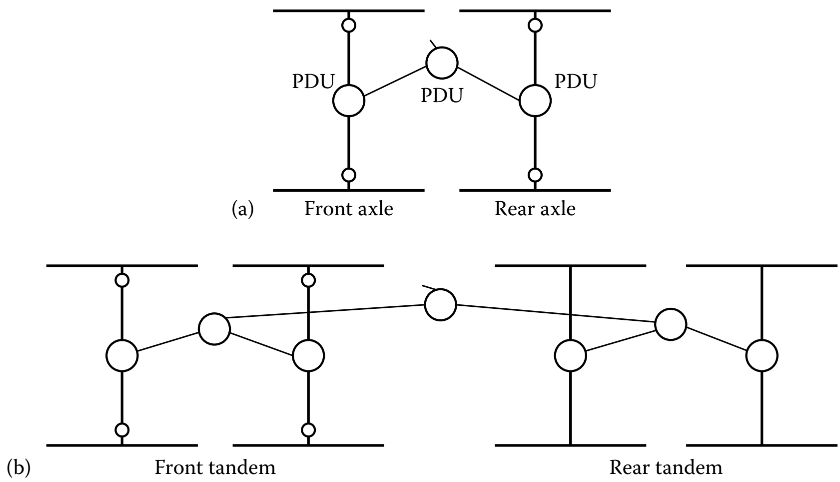

FIGURE 1
Driveline system layouts of (a) 4 × 4 and (b) 8 × 8 vehicles.

experience gathered during World War II: The shortage of arterial and high-type roads and the poor terrain mobility of vehicles determined how military applications progressed for many postwar years. After World War II, with the development of hard-surface roads, multiwheel drive vehicle layouts were actively designed and employed in the agriculture sector, and in construction, forestry, and the petroleum industry. At a later stage when space research and space flights began, planet rovers were also designed with all the wheels driven by torque. From the 1980s onward, passenger cars and sport-utility vehicles with four driving wheels became very popular. The twentieth century thus saw the emergence of multiwheel-drive vehicles of all types. In conjunction with this, the names of many engineers, researchers, and professors, who developed the theory of motion of vehicles and the practice of driveline system design, should be mentioned here. This is a very difficult task and I fully realize that I could fail to mention many of them. However, the following persons deserve special mention: Yakov S. Ageykin, Alexander S. Antonov, Dmitriy A. Antonov, Pavel V. Aksenov, Mieczyslaw G. Bekker, Nikolay F. Bocharov, G. Broulhiet, Colin Chapman, Evgeniy A. Chudakov, Keith Duckworth, J. R. Ellis, Yaroslav E. Farobin, Thomas G. Gillespie, Wunibald Kamm, Frederick W. Lanchester, Andrey S. Litvinov, William F. Milliken and Douglas L. Milliken, M. Mitschke, Tatsuro Muro, Maurice Olley, Hans B. Pacejka, Vladimir A. Petrushov, Yuliy V. Pirkovskiy, Vladimir F. Platonov, A.R. Reece, Robin Sharp, Anatoliy T. Skoybeda, Gleb A. Smirnov, Sergei B. Shukhman, Jaroslav J. Taborek, Igor S. Tsitovich, Jo Y. Wong, and Georgiy V. Zimelev.

Engineering designers and their design developments are now part of history; notable engineers among them include N.A. Astrov, Marius Berliet, William Besserdich, Carl Borgward, Henry Bussing, Carlo Cavalli, John W. Christie, V.E. Chvyalev, Wesley M. Dick, G.A. Fest, V.A. Grachev, Nikolay I. Korotonoshko, A.M. Kriger, I.P. Ksenevich, Nils Magnus, Alfred Masury, A.A. Lipgart, Ralf Nash, Ferdinand Porsche, Wilfredo Ricart, Delmar B. Roos, B.L. Shaposhnik, and M.S. Vysotskij. Vitali A. Grachev, chief design engineer, who has designed many forms of multiwheel drive vehicles, became legendary in the former U.S.S.R. among experts who designed military vehicles. Boris L. Shaposhnik established the Special Design Office, known by its Russian-language acronym, SKB-1, in Minsk, Belarus, where new-generation vehicles, such as multiwheel drive missile carriers, were designed. Vladimir E. Chvyalev followed in the footsteps of B.L. Shaposhnik. He headed the design department of the Minsk Wheeled Tractors Company that grew out of SKB-1. Dr. Mikhail S. Vysotskij, academician at the National Academy of Sciences of Belarus and chief design engineer of automotive vehicles in Belarus led the design of heavy-duty vehicular trains. Even now, in spite of advancing years, he directs a scientific research institute of the National Academy of Sciences.

Another eminent individual in the field, Professor Anatoliy Kh. Lefarov, a design engineer, subsequently became a professor of Belarusian National Technical University (the current name), Minsk, Belarus; Dr. Lefarov was the deputy of chief design engineer, V.A. Grachev, and the first deputy of chief design engineer, B.L. Shaposhnik. During the 1960s, Dr. Lefarov established a research group in the field of multiwheel drive vehicles that designed various driveline systems and PDUs and performed analytic and experimental studies on the effect of driveline systems on the properties of vehicles.

We have been associated with Dr. Lefarov's research group for our entire professional lives. Dr. Kabanau was his first postgraduate student and then became the principal design engineer, who has to his credit designed many mechanisms and systems. Dr. Andreev is concerned with simulating the motion of vehicles and designing PDUs. Seventeen professionally fortunate years, which have rapidly slipped away, with our leader united me not only in a common endeavor but also in a binding friendship. Dr. Lefarov has shaped me not only as an expert but also as a human being. It so happened that after he passed away in 1992, I became the leader of our group.

The culmination of all of this has led to this book. This book comprehensively covers the subject matter from a historic overview, classification, and the nature of driveline influence on vehicle dynamics and performance (Chapter 1), through analytical fundamentals (Chapters 2 through 5) and optimization and control of wheel power distributions (Chapters 6 and 7), to mechanical and mechatronic design of advanced systems (Chapters 2 through 7) and experimental research and tests (Chapter 8). Also, I believe the readers will thoroughly enjoy the illustrations, hand-drawn by Dr. Kabanau.

In many ways, this book is unique; it is probably the only book that deals with the solution of the two fundamental engineering problems that were formulated earlier in the preface. Therefore, the reader can see that the book presents an analytical treatment of driveline systems research, design, and tests based on vehicle dynamics and performance requirements. Methodologically, this is described in two ways. First, the book introduces analytical tools for studying the driveline effects on power distribution among the driving wheels and then on the dynamics and performance of vehicles. Engineering applications of these tools, for instance, include the comparative analysis of several driveline systems with the purpose of selecting a driveline system that provides a given vehicle with better performance and also to evaluate same-class vehicles with different driveline systems. Additionally, the developed techniques adequately supplement the mathematical modeling of vehicle dynamics. Chapters 1 through 5 and Chapter 7 present the necessary material for such mathematical modeling of driveline systems that can be compiled of different types of PDUs. All analytical techniques were built based on the so-called generalized vehicle parameters, which integrate characteristics of PDUs with tire (or combined tire/soil) characteristics and, implicitly, suspension characteristics.

Second, the book develops methodologies for the synthesis of optimal characteristics of PDUs that can be applied to different types of vehicles. Thus, a researcher would not need to run a comparative analysis of hundreds of potential driveline systems to try to find a better one for the vehicle under design. Instead, optimal characteristics can be directly achieved and then optimal PDUs can be designed. Respective analytical techniques were based on the principles of driveline system designs that were developed on the inverse vehicle dynamics approach and first introduced in this book. To learn more about the inverse vehicle dynamics approach and the optimization of power distribution among the driving wheels, the reader should start with Sections 1.5 and 1.6, then go through Sections 2.9 and 3.6, and finally to Chapters 6 and 7. The mechanical design of PDUs and control development issues are covered in Chapters 2 through 5, and also in Chapter 7.

The book is also unique in the sense that it was written virtually entirely on the basis of the results of investigations by its authors. All analytical tools, and computational, design, and test methods were verified through many engineering projects; some of the projects are presented in this book as illustrative examples to prove the applicability of the developed theories. The material in this book will provide the reader with answers to intriguing engineering problems such as achieving higher energy/fuel efficiency of a vehicle by driving either all the wheels or not all the wheels, obtaining oversteering characteristics by increasing the torque at the front-steered wheels, and many other such technical problems. Engineering workers will find interesting methods of design and experimental studies of new driveline systems that provide for optimal/specified vehicle properties. The presented methodologies and results on the optimization of wheel power distributions among the driving wheels can also be of interest to engineers working on vehicles with individual wheel drives and vehicles with hybrid driveline systems.

The reader will find only some of the references in the text to the detailed bibliography at the end of the book. The bibliography reflects that, to a large degree, the history of investigations in driveline system engineering and vehicle dynamics during the twentieth century and the start of the twenty-first century was compiled by studying a large number of publications and it should be regarded as a source of additional engineering data; after all, the experience of each expert is unique. We have conscientiously investigated publications on the theory of motion of all-wheel-drive vehicles and on driveline system design for many years and, if we missed some important investigations and did not include them in the bibliography, we apologize to the authors. We have also included some of our own publications in the bibliography that reflect not only the results of investigations but also, by representing a sequence, give an idea about the development of our scientific and engineering approaches to, and methods of, solutions for engineering problems.

The volume of the scientific and engineering information and the structure of its arrangement within the book are such that it could be used both by beginning design and test engineers as well as engineers with experience in the design and experimental studies of various PDUs, driveline systems, and multiwheel drive ground vehicles as a whole. The book will also be useful to research engineers involved in simulating motion and in testing multiwheel drive vehicles, because it illuminates many aspects of the mathematical simulation of the different driveline systems and dynamics of such vehicles, something that is usually not examined in classical textbooks on vehicle dynamics. The simulation of vehicle dynamics on the basis of the inverse dynamics approach is also a topic that is examined in this book for the first time. This method is also used for working out algorithms for the control of mechatronic driveline systems.

For many years, we developed and delivered university-level courses on the theory of vehicle motion and on the design of driveline systems. This book was written with reference to these courses and can therefore also be used as a textbook on advanced vehicle dynamics and on the design of driveline systems in master of science and PhD courses. Thereby, all the mathematical formulae in the book have been derived together with the necessary detailed explanations, something that makes the material easily comprehensible and convenient both for the student and the lecturer. The analytical results are illustrated by quantitative examples (illustrative problems) and examples of developing driveline systems and PDUs. Using our experience, we devised engineering problems associated with the dynamics of vehicular motion, design, and testing of driveline systems. These problems developed for each chapter can be used in the course of studies as examination problems or homework assignments, and also by practicing engineers for better familiarization with the material in the book and for illustrating its underlying theoretical principles.

Dr. Vladimir V. Vantsevich
Southfield, Michigan

Acknowledgments

We regard it as a pleasure to express our appreciation to our colleagues whom we invited to participate and write sections of the book. More than 30 years of friendship and mutual work link me with Dr. Sergei I. Strigunov and Dr. Vladimir S. Voiteshonok. We studied together in the university and then worked with Dr. Lefarov. Dr. Strigunov presented the results of his studies and participated in writing Sections 3.1, 3.2, 3.3.1, 3.4, and 8.5. Dr. Voiteshonok contributed the results of his studies and participated in writing Sections 2.5.1, 3.6, and 8.5.

In the course of writing this book, we had the pleasure of continuing cooperation with my first PhD student and currently chief design engineer, Front Driving Axles and Wheel Systems at P/A Minsk Tractor Works, Belarus—Dr. Valeriy Yermalionak. Dr. Yermalionak participated in writing Sections 2.5.1, 2.6.1, and 2.7. He also supplied some material for Sections 1.2 and 4.4.

Dr. Siarhei V. Kharytonchyk, also my PhD student and currently the head of the computer center at the Joint Institute of Mechanical Engineering, National Academy of Sciences of Belarus, participated in the investigations that are jointly described in Section 7.6. Together with another PhD student, Dr. Gennady Valyuzhenich, we tested the differential lubrication systems, which we have described in Section 8.3.

I collaborated with Dr. Arkadij D. Zakrevskij, corresponding member of the National Academy of Sciences of Belarus, who works in the field of parallel logical control algorithms. Some of the results of his studies and their application to the design of mechatronic systems have been included in Section 7.6.2, which was jointly written.

It has been almost 15 years since I first collaborated with Dr. Gemunu S. Happawana, professor at California State University at Fresno, Fresno, California. Dr. Happawana participated in writing Sections 2.1, 6.5, and 7.6.1.

For many years, I was lucky to enjoy the friendship and professional collaboration of Dr. Yuliy V. Pirkovskiy, and, after his passing, of his successor Dr. Sergei B. Shukhman. I am glad that the book includes Sections 7.7.1 and 7.7.2, written by Dr. Shukhman and his colleague Dr. Evgenij I. Prochko.

We would like to express our gratitude to the heads of Lawrence Technological University, Southfield, Michigan—President Dr. Lewis N. Walker, Provost Dr. Maria J. Vaz, and Associate Provost Dr. Steven K. Howell—for their financial support for the translation of the manuscript.

This book was written during a transitional period of my life, when I was taking up work at Lawrence Technological University.

I wish to express my heartfelt gratitude for professional collaboration and friendship, technical and personal advice, and fruitful discussions to Dr. William Begell, Professor Eugene I. Rivin, Dr. Joseph and Mrs. Sally Wolf, Dr. Moisey and Mrs. Vera Shkolnikov, Dr. Lev Gileles and Dr. Otto Zaslavskiy, Dr. Simon and Mrs. Larisa Itskovich, Dr. Guennadi Koulechov and Svetlana Skalskaya, Robert Edmonson, Mr. John and Mrs. Carol Erickson, Mr. James and Mrs. Sandra Fisher, Mr. Patric and Mrs. Maryann Hermes, Joanne Kowalenok, Mr. Ephim and Mrs. Anna Schmidt, Mr. Volodymyr and Mrs. Lyubov Shesiuk, Dr. Jan and Mrs. Nadzia Zaprudnik, Dr. Vitaut Kipiel, Mr. Allen M. Krass, and many colleagues and friends in academia and industry, and my church.

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The publication of this book owes a great deal to the painstaking labor of Dov Lederman, BME, interpreter, to whom we express our gratitude and appreciation.

The time spent in writing the book was irreversibly taken away from family time. We are immeasurably grateful to our spouses and children for their understanding of the importance of our contribution to the profession, and for their unwavering love and support.

When I was still a student, I once asked Dr. Lefarov why he had not written his autobiography. His reply was surprising and it took me years to understand and accept it. He said that his life as a person is not of much interest—after all it is very similar to the lives of millions of people in the country where we live. Dr. Lefarov added that it would be much more interesting to read our engineering books, which reflect not only the progress made in the field of engineering, but also, frequently, give insight about the lives of the authors at a professional, social, and sometimes even personal level. He did not leave any remarks concerning his life. These were written for this book by his son, Dr. Victor A. Lefarov.

As for engineering books, our new book is now held by the readers. The authors hope that this book will be useful. As Dr. Lefarov once said, the engineer must remember that any redundant line on the blueprint of an design may require an additional machine tool and maybe even an entire automatic production line. If the publication of this book reduces the number of ''redundant lines,'' that is, mistakes, and results in the appearance of more advanced driveline systems and causes more young engineering experts to become professionally associated with multiwheel drive vehicle engineering, the authors will have the satisfaction of having achieved their goal. Any remarks and comments will be gratefully received.

Dr. Vladimir V. Vantsevich
Southfield, Michigan

Authors

Alexandr F. Andreev has been an associate professor in tractor engineering at the College of Automobile and Tractor Engineering, Belarusian National Technical University, since 1975. He received his PhD in automobile and tractor engineering in 1972. Dr. Andreev has also worked as a leading researcher in vehicle dynamics and vehicle performance analysis of the Research and Design Group on Multi-Wheel Drive Vehicles since the founding of the group in 1963. His research concentrates on motion modeling, design, and computation of ground transport and tractive vehicles, and vehicle systems. Dr. Andreev served as the principal investigator to 11 research programs and projects, and contributed to 57 more projects with his innovative analytical models of vehicles in motion, driveline systems, and tire–ground interaction models. His designs of limited slip differentials have been employed in 4×4 farm tractors and multiwheel drive terrain trucks.

Dr. Andreev has written 19 internal research reports, and has published 4 technical books and graduate level texts on vehicle driveline system design and computation, hydraulic and pneumatic control systems, and hydro-drive design and engineering. He has authorized 22 research papers published in journals and conference proceedings, and has received 8 certified inventions.

Professor Andreev developed standards and curricula for MSc programs in tractor engineering, mobile track and wheeled vehicle engineering, and urban electric vehicle engineering. He served at the Higher Education Publishing House of Belarus for 15 years, and as the executive editor of two major research journals in Belarus: *Automobile and Tractor Engineering* and *Design, Computation and Operation of Automobiles and Tractors*.

Dr. Viachaslau I. Kabanau has been an associate professor at the College of Automobile and Tractor Engineering, Belarusian National Technical University, ever since he received his PhD in automobile and tractor engineering in 1966. He also served as the associate dean of the college for 10 years. Dr. Kabanau has worked as a leading researcher and the principal design engineer of the Research and Design Group on Multiwheel Drive Vehicles since the founding of the group in 1963. His research and design work focuses on vehicle driveline systems and mechanism design, and vehicle experimental research. Dr. Kabanau participated in and contributed with his original designs of transfer cases, differentials with lubrication systems, locking and free-running differentials to more than 20 R&D programs and projects on terrain and highway truck and farm tractor engineering. Dr. Kabanau has authorized 6 technical books, graduate level texts, and brochures; he has published 40 reviewed papers in research journals and conference proceedings, and has received 11 certified inventions. He has also delivered a number of presentations and invited lectures to industry and universities.

Professor Kabanau actively developed innovative academic programs and courses in tractor engineering. His lecture courses on hydro dynamic drive engineering, vehicle experimental research and tests, vehicle art design, and ergonomics have been of immeasurable service to the College of Automobile and Tractor Engineering and also established him as an outstanding teacher and student project advisor.

Vladimir V. Vantsevich is a professor in mechanical engineering and the founding director of the MSc in Mechatronic Systems Engineering Program and the Laboratory of Mechatronic Systems at Lawrence Technological University, Michigan. He is also a cofounder and associate director of the Automotive Engineering Institute. Before joining Lawrence Tech, Dr. Vantsevich was a professor and the head of the Research and Design Group on Multiwheel Drive Vehicles that designed and developed a number of mechatronic and mechanical driveline systems for multipurpose vehicles in Belarus. He received his PhD and DSc (the highest degree in the former U.S.S.R.) in automobile and tractor engineering from the Belarusian National Technical University approved by the Higher Awarding Committee of the Russian Federation.

Professor Vantsevich's research area is inverse and direct dynamics of mechanical and mechatronic systems, and system modeling, design, and control. Applications include conventional and autonomous, multiwheel ground vehicles, and vehicle locomotion and driveline systems. He developed a novel research avenue—*inverse ground vehicle dynamics*—which is the basis of his optimization of power distribution among the driving wheels and control of vehicle performance including vehicle mobility, energy/fuel consumption, traction and acceleration performance, and stability of motion.

He is the author of 5 technical books and more than 100 research papers on inverse dynamics, vehicle performance and energy efficiency optimization and control, and design of driveline and autonomous wheel power management systems. Professor Vantsevich has participated in more than 110 science seminars, and has delivered lectures and technical presentations at academic institutions, professional societies, and to industry. He is a registered inventor of the U.S.S.R. and holds 30 certified inventions.

Professor Vantsevich is the founder and editor of a series of handbooks, textbooks, and references on ground vehicle engineering at Taylor & Francis Group/CRC Press. He is a member of the editorial boards of the *International Journal of Vehicle Autonomous Systems*, the *Journal of Multi-Body Dynamics* (Part K of the Proceedings of the Institution of Mechanical Engineers), and the *International Journal of Advanced Mechatronics and Robotics*. He is also an associate editor of the *International Journal of Vehicle Noise and Vibration*.

Professor Vantsevich was honored with a fellowship of the American Society of Mechanical Engineers and of the Belarusan Institute of Arts and Sciences, New Jersey. He is a member of the Association of Vehicle Autonomous Systems International, the Society of Automotive Engineers, the International Society for Terrain-Vehicle Systems, and the International Association for Vehicle System Dynamics.

List of Symbols

Symbol	Definition	First Mentioned and Explained in Section
a	Distance between the gravity center and the front axle of a vehicle with two axles	Section 2.2
$a_{\mathrm{dr}}, a_{\mathrm{n}}, a_{\mathrm{f}}, a_{\mathrm{d}}, a_{\mathrm{b}}$	Tire off-set distance, the distance between the line of the normal reaction of a wheel and the wheel axis of rotation in the driven, neutral, free, diving, and braking modes	Section 1.3
a_x	Longitudinal acceleration of the center of the wheel; also vehicle acceleration in straight line motion	Sections 1.3 and 1.4
a_y	Vehicle lateral acceleration	Section 1.5
A	Geometric parameter of limited slip differential	Section 4.2
A_{d}	Conditional cross-sectional area of a bevel gear differential	Section 2.5
b	Distance between the gravity center and the rear axle of a vehicle with two axles	Section 2.2
b	Tooth width	
B	Geometric parameter of limited slip differential	Section 4.2
B_{a}	Width of the implement's sweep	Section 1.5
C	Geometric parameter of limited slip differential	Section 4.3
C_{c}	Constant specific for a particular limited slip differential	Section 4.7
$C_{\mathrm{p}i}$	Suspension stiffness factor reduced to a wheel	Section 6.6
$C_{\mathrm{t}i}$	Tire (tire and soil for deformable surfaces) normal stiffness factor	Section 6.6
d_{a}	Diameter of the addendum circle	Section 2.3
$d_{\mathrm{go}}, d_{\mathrm{gi}}$	Outer and inner friction diameters of a bevel side gear with the case of a differential	Section 2.5
$d_{\mathrm{po}}, d_{\mathrm{pi}}$	Outer and inner friction diameters of a pinion with the case of a differential	Section 2.5
d_{sc}	Spider pin diameter in contact with the case of a differential	Section 2.5
d_{sp}	Spider pin diameter in contact with the pinion	Section 2.5
D	Geometric parameter of limited slip differential	Section 4.3
D_{a}	Air drag	Section 2.5
D_{a}	Vehicle air dynamic force (air drag)	Section 1.4
E	Shift of center of turn	Section 1.5
f	Rolling resistance coefficient	Section 1.3
f_0	Rolling resistance coefficient at the creeper speed	Section 6.6
F_0	Force in the contact between the spider and the differential's housing	Section 2.5
F_{l}	Wheel lateral reaction	Section 1.3

F_l^a	Component of the wheel lateral reaction exerted by the interaxle power dividing unit	Section 2.8
F_l^{max}	Maximal wheel lateral reaction limited by the deformation properties of the tire and the soil	Section 1.3
F_l^w	Component of the wheel lateral reaction exerted by the interwheel power dividing unit	Section 2.8
F_a	D'Alambert's force	Section 1.4
F_b	Force in the contact between the spider and the pinion	Section 2.5
F_d	Vehicle drawbar pull load	Section 1.4
F_d	Drawbar pull	Section 2.5
$F_{\text{frame}\,x}$	A force acting on a wheel from the vehicle frame in x-direction (frame force)	Section 1.3
F_k	Radial force acting between cams of a side gear and an intermedium cam bushing	Section 4.3
F_x	Wheel circumferential (tangential) force	Section 1.3
$F_{x\Sigma}$	Vehicle total circumferential force	Section 1.4
F_x^{max}	Maximum circumferential wheel force that can be attained at the contact between the wheel and the surface of motion	Section 1.3
F_w	Wheel traction force (or net tractive force)	Section 1.3
F_ψ	Total force of resistance to vehicle motion	Section 1.5
g_d	Drawbar pull–specific fuel consumption	Section 6.4
g_e	Engine-specific fuel consumption	Section 1.4
h_z	Tire normal deflection	Section 1.3
i_M	Number of friction pairs in one disk clutch of a limited slip differential	Section 4.2
I_w	Moment of inertia of a wheel about the axis of rotation	Section 1.3
k	Factor from the exponential characteristic (1.26)	Section 1.3
K	Design parameter of a planetary gear set	Section 7.4
K_a	Longitudinal stiffness coefficient of the tires of the driving axle	Section 2.9
K_{a1}, K_{a2}	Coefficients for evaluating vibrations of the total axial force acting on a bevel side gear from the pinions	Section 2.3
K_d	Torque bias (locking coefficient) corresponding to dynamic friction coefficient	Section 4.1
K_{dp}	Torque bias (locking coefficient) corresponding to static friction coefficient	Section 4.1
K_{ext}	Locking coefficient variation factor	Section 4.9
K_{max}	Locking coefficient variation factor	Section 4.9
K_{mk}	Factor of optimal torque distribution among the wheels of a drive axle	Section 6.2
K_{mo}	Factor of optimal torque distribution among the drive axles	Section 6.2
K_{opt}	Coefficient of effectiveness of distributing the power between the wheels	Section 1.4
K_{pi}	Shock-absorber resistance factor reduced to a wheel	Section 6.6

K_{pu}	Single correction factor	Section 4.2
K_q	Driveline system quality factor	Section 1.4
K_r	Relative friction torque	Section 4.1
K_{s1}, K_{s2}	Coefficients for evaluating vibrations of the total axial force acting on a bevel pinion from the side gears	Section 2.3
K_{ti}	Tire (tire and soil for deformable surfaces) damping factor	Section 6.6
K_u	Mileage wear factor of a differential	Section 2.5
K_w	Coefficient of utilization of the traction weight	Section 1.1
K_{wp}	Vehicle weight/payload ratio	Section 1.1
K_x, K_w	Longitudinal stiffness coefficients of a tire	Section 1.3
K_y	Lateral slip resistance coefficient (also called the cornering stiffness of a tire)	Section 1.3
K_{y0}	Tangent of the curve $F_l = f(\alpha)$ at the coordinate origin	Section 1.3
K_M	Force-loading factor of a differential	Section 2.5
K_N	Energy-loading factor of a differential	Section 2.5
K_X, K_Y, K_A	Indices for assessing vehicle turnability	Section 1.5
K_μ	Gripping force utilization factor	Section 1.5
i_{ab}^{H}	Gear ratio of a planetary gear set	Section 7.4
l_a	Longitudinal coordinate of the center of gravity of a multiwheel drive vehicle	Section 6.5
l_i	Distance between the front axle and the ith axle of a vehicle, $i = 2, m$	Section 2.1
l_{sc}	Spider pin length in contact with the case of a differential	Section 2.5
l_{sp}	Spider pin length in contact with the pinion	Section 2.5
L	Wheelbase of a vehicle	Section 3.3
m	Number of the driving and driven (not coupled to the driveline system) axles of a vehicle	Section 1.2
m_a	Gross (full) mass of a truck	Section 2.5
m_{dr}	Mass taken by the drive wheels of a vehicle	Section 2.5
m_r	Kinematic discrepancy factor of a 4×4 vehicle	Section 3.2
m_{rR}	Kinematic discrepancy factor of a 4×4 vehicle	Section 3.2
m_{ru}	Kinematic discrepancy factor of a 4×4 vehicle	Section 3.2
m_{ruR}	Kinematic discrepancy factor of a 4×4 vehicle	Section 3.2
m_t	Tractor mass	Section 1.1
m_{te}	Outer module a side gear and a pinion measured at the large end of the teeth	Section 2.3
m_u	Kinematic discrepancy factor of a 4×4 vehicle	Section 3.2
m_{uR}	Kinematic discrepancy factor of a 4×4 vehicle	Section 3.2
m_{Hi}	Kinematic discrepancy factor of the ith driving axle	Section 3.2
m_R	Kinematic discrepancy factor of a 4×4 vehicle	Section 3.2
M_f^c	Rolling resistance moment caused by the normal reaction shift	Section 1.3
M_B	Yaw moment due to the inequality of the left and right circumferential forces	Section 2.8

M_{Bd}	Moment of resistance to turning of a drive axle with a limited slip differential after the differential actuated	Section 4.10
n	Number of the driving axles of a vehicle	Section 1.2
n_{w}	Number of revolutions of a wheel	Section 1.3
N_{PDU}	Number of power dividing unit in a driveline system	Section 1.2
p_0	Pressure in the contact between the spider and the differential's housing	Section 2.5
p_{a}	Pressure in the thrust washer of the bevel side gear	Section 2.5
p_{b}	Pressure in the contact between the spider and the pinion	Section 2.5
p_{c}	Pressure in the thrust washer of the bevel pinion	Section 2.5
p_{w}	Tire inflation pressure	Section 1.3
P_0	Input power supplied to a power dividing unit	Section 2.5
P_{a}	Power extended on increasing the kinetic energy of the wheel in translational motion	Section 1.3
P_{ax}	Mechanical power loss in engine and power for driving auxiliary devices	Section 1.4
P_{b}	Wheel brake power	Section 7.5
P_{d}	Drawbar pull power	Section 1.4
P_{drl}	Mechanical power loss in the driveline system	Section 1.4
P_{e}	Engine power	Section 1.4
$P_{\mathrm{e}}^{\mathrm{max}}$	Maximal engine power	Section 1.1
P_{f}	Wheel rolling resistance power	Section 1.3
$P_{\mathrm{f}\Sigma}$	Vehicle rolling resistance power	Section 1.4
$P_{\mathrm{fc}\Sigma}$	Power of resistance to the rolling of the wheels caused by the vehicle curb weight	Section 1.4
P_{fg}	Wheel rolling resistance power needed for overcoming force R_{xg}	Section 1.3
$P_{\mathrm{fg}\Sigma}$	Power of resistance to the rolling of the wheels caused by the cargo being transported	Section 1.4
P_{frame}	Power transmitted from a wheel to the vehicle frame	Section 1.3
P_i	Geometric parameter of limited slip differential	Section 4.1
P_i^{in}	Engine indicator power	Section 1.4
P^{out}	Vehicle output power	Section 1.4
P_{trm}	Mechanical power loss in transmission	Section 1.4
P_{ts}	Mechanical power loss for deflecting tires and soil	Section 1.4
$P_{\mathrm{w}}^{\mathrm{in}}$	Wheel input power	Section 1.3
$P_{\mathrm{w}}^{\mathrm{out}}$	Wheel output power	Section 1.3
$P_{\mathrm{w}\Sigma}^{\mathrm{in}}$	Input power on the drive wheels	Section 1.4
p_x	Vehicle mobility indicator from the point of view of its traction	Section 1.5
P_{M}	Driveline input power	Section 1.4
P_{ε}	Power needed to increase the kinetic energy of the rotational motion of the wheel	Section 1.3
P_{δ}	Wheel slip power	Section 1.3
$P_{\delta\Sigma}$	Vehicle slip power	Section 1.4

p_μ	Vehicle mobility indicator from the point of view of its grip with the surface of motion	Section 1.5
q	Coefficient of power distribution to a drive axle	Section 1.4
q_d	Pressure on the plate disks of a limited slip differential	Section 4.9
q_k	Pressure in the contact of the pinion pin and the groove of the differential's case	Section 4.9
q_R	Vehicle turning radius variation factor	Section 1.5
$q_{\alpha T}$	Correction factor for reflecting the effect of traction force F_w on function $F_l = f(\alpha)$	Section 1.3
Q_a	Total axial force acting on a bevel side gear from the differential's pinions	Section 2.3
$Q_a^{'}$	Axial component of the resultant force on a bevel side gear from one pinion	Section 2.3
Q_{avg}	Average fuel consumption per 100 km of travel	Section 1.5
Q_h	Per hour fuel consumption	Section 1.4
Q_k	Axial (thrust) force acting between cams of a side gear and an intermedium cam bushing	Section 4.3
Q_m	Lubricant flow rate	Section 8.3
Q_s	Fuel consumption referred to the distance traveled by the vehicle	Section 1.4
Q_s	Total axial force acting on a bevel pinion from the differential's side gears	Section 2.3
Q_{sp}	Spring force	Section 4.7
r	Radius of an unloaded wheel (also, radius of a rigid wheel)	Section 1.3
r_a^0	Generalized rolling radius of a vehicle in the driven mode	Section 2.8
r_{ai}^0	Generalized rolling radius of the ith driving axle in the driven mode	Section 2.8
r_c	Pitch (average) radius of a pinion	Section 2.5
r_{cf}	Average friction radius of a bevel pinion	Section 2.5
r_{cr}	Radius of the carrier	Section 2.1
r_d	Dynamic loaded radius of a wheel	Section 1.3
r_g	Pitch (average) radius of a side gear	Section 2.1
r_{gb}	Medium friction radius at the end of the pressure cup and the pinion's shoulder	Section 4.4
r_{gf}	Average friction radius of a bevel side gear	Section 2.5
r_k	Medium radius of the end cams in a limited slip differential	Section 4.3
r_{kp}	Medium radius of friction between the spider and a side gear	Section 4.3
r_0	Radius (arm) of the force F_0	Section 2.5
r_s	Static loaded radius of a wheel	Section 1.3
r_{sp}	Spider pin radius in the contact with pinion	Section 4.2
r_w	Effective rolling radius of a wheel in the driving and braking modes	Section 1.3
r_{wp}	Friction radius of the pinion at the differential's case	Section 4.2
r_w^f	Effective rolling radius of a wheel in the free mode	Section 1.3

r_{w}^{0}	Effective rolling radius of a wheel in the driven mode	Section 1.3
r_{M}	Medium friction radius of disk plates in limited slip differential	Section 4.2
R_{a}	Vehicle actual radius of turn	Section 1.5
R_{t}	Vehicle theoretical radius of turn	Section 1.3
R_{td}	Turn radius of a drive axle with a limited slip differential after differential actuated	Section 4.10
R_{x}	Rolling resistance force of a wheel	Section 1.3
$R_{x\mathrm{c}}$	Wheel rolling resistance caused by the curb weight W_{wc}	Section 1.3
$R_{x\mathrm{g}}$	Wheel rolling resistance caused by the payload weight W_{wg}	Section 1.3
$R_{x\mathrm{g}\Sigma}$	Vehicle total rolling resistance caused by W_{g}	Section 1.4
R_{z}	Ground normal reaction onto a wheel	Section 1.3
s_{δ}	Tire slip ratio (slippage)	Section 1.3
$s_{\delta a}$	Generalized slippage of a vehicle	Section 2.8
$s_{\delta \mathrm{a} i}$	Generalized slippage of the ith driving axle	Section 2.8
S_{e}	Handling sensitivity	Section 1.5
$S_{\mathrm{e}}^{\mathrm{dn}}$	Unsteady-state handling sensitivity	Section 1.5
$S_{\mathrm{e}}^{\mathrm{st}}$	Steady-state handling sensitivity	Section 1.5
S_{t}	Tooth thickness	Section 2.3
S_{w}	Wheel's travel	Section 1.3
t_{b}	Wheel tread (a distance between the wheels of an axle)	Section 2.2
t_{p}	Pure time of work of a farm tractor	Section 1.5
t_{Σ}	Total time of work of a farm tractor	Section 1.5
t_{R}	Time delay in vehicle's reaction to a quick turn of the steering wheel	Section 1.5
T	Output torque on one output shaft of a power dividing unit	Section 2.1
T_{b}	Brake torque	Section 1.3
T_{gb}	Torque of friction at the pressure cup	Section 4.4
T_{i}	Torque of the ith drive axle	Section 1.3
T_{int}	Elastic internal torque in a differential	Section 2.1
T_{lock}	Locking torque of a clutch	Section 2.7
T_{0}	Input torque of a power dividing unit	Section 2.1
T_{r}	Dynamic friction torque in limited slip differential	Section 4.1
T_{rp}	Torque of the primary friction in the clutches of a limited slip differential	Section 4.2
T_{rs}	Static friction torque in a limited slip differential	Section 4.1
T_{sb}	Torque of friction between the bushing and the differential's case (or the spider)	Section 4.3
T_{sp}	Friction torque between the pinion and the carrier pin	Section 4.2
T_{w}	Wheel drive torque	Section 1.3
T_{wp}	Friction torque between the pinion and the case of a differential	Section 4.2
u_{a}	Driving axle gear ratio	Section 2.5

u_d	Internal gear ratio of a differential	Section 2.1
u_{dd}	Internal gear ratio of a double differential	Section 7.4
u_d^*	Optimal internal gear ratio of a differential	Section 2.9
u_f	Final drive gear ratio	Section 2.1
u_i	Gear ratio from the central power dividing unit's (i.e., transfer case) input shaft to the wheels of the *i*th drive axle or to the interaxle differential of the *i*th drive axle (see a note on p. 302)	Section 2.8
u_k	Wheel–hub reduction gear ratio	Section 2.1
u_p	Torque distribution factor to link the differential gear ratio and the kinematic discrepancy factor	Section 3.3
u_{trm}	Vehicle transmission gear ratio	Section 6.1
u_w	Weight distributing factor	Section 2.8
V_a	Theoretical travel velocity of a vehicle	Section 3.2
V_{avg}	Vehicle average actual velocity	Section 1.5
V_c	Linear velocity of the pinion and the spider (on the friction surface)	Section 2.5
V_{cr}	Vehicle critical speed	Section 1.5
V_{mid}	Vehicle average design velocity	Section 1.1
V_t	Theoretical linear velocity of the wheel center (no slip occurs)	Section 1.3
V_x	Actual linear velocity of the wheel center (slip occurs). Also, vehicle actual velocity in straight line motion or along the longitudinal axis	Sections 1.3 and 1.4
V_δ	Tire slip velocity	Section 1.3
W_a	Vehicle gross (total) weight	Section 1.1
W_c	Vehicle curb weight	Section 1.1
W_{dr}	Adhesion weight, i.e., the weight devolving upon the driving wheels	Section 1.1
W_g	Vehicle payload	Section 1.1
W_w	Normal load of a wheel (wheel weight)	Section 1.3
W_{wc}	Wheel normal load without the useful load (curb weight)	Section 1.3
W_{wg}	Wheel normal load caused by the useful load (payload weight)	Section 1.3
x	Coefficient of initial contour displacement	Section 2.3
y_Δ	Perceptible displacement coefficient	Section 2.3
Y_w	Wheel lateral force	Section 1.3
z_c	Gear tooth number of a pinion of a differential	Section 2.1
z_c	Tooth number of a pinion of a differential	Section 2.3
z_{cr}	Tooth number of a spur gear equivalent to a bevel pinion	Section 2.3
z_g	Gear tooth number of a side gear of a differential	Section 2.1
z_g	Tooth number of a side gear of a differential	Section 2.3
z_{gr}	Tooth number of a spur gear equivalent to a bevel side gear	Section 2.3

Greeks

α	Wheel slip angle	Section 1.3
α_k	Pressure angle corresponding to a tooth point of contact and to the rotation center of a spur gear equivalent to a bevel side gear	Section 2.3
α_{kc}	Pressure angle corresponding to a point of tooth contact and to the rotation center of a spur gear equivalent to the pinion	Section 2.3
α_{w}	Pressure angle corresponding to the pitch point	Section 2.3
δ	Wheel steering angle	Section 1.3
δ_{c}	Pitch pinion angle	Section 2.3
δ_{r}	Vehicle mass factor	Section 6.3
ε_{w}	Wheel angular acceleration	Section 1.3
η_{d}	Mechanical efficiency in a reduction gear	Section 4.1
η_{f}	Vehicle rolling resistance efficiency	Section 1.4
η_{fw}	Wheel rolling resistance efficiency	Section 1.3
$\eta_{\mathrm{fw}}^{\mathrm{tr}}$	Wheel rolling resistance efficiency for evaluating the resistance exerted by the payload	Section 1.3
η_{h}	Fuel consumption variation factor	Section 3.4
η_{M}	Overall mechanical efficiency of the driveline system	Section 1.4
$\eta_{\mathrm{M}i}$	Mechanical efficiency of *i*th branch of the driveline system	Section 1.4
η_{n}	Mechanical efficiency of a differential	Section 2.5
η_{trm}	Mechanical efficiency of vehicle transmission	Section 6.1
η_{t}	Vehicle tractive efficiency	Section 1.4
$\eta_{x\Sigma}$	Vehicle running gear total efficiency	Section 1.4
η_x^{t}	Vehicle running gear tractive efficiency	Section 1.4
η_{v}	Vehicle velocity variation factor	Section 3.5
$\eta_{\mathrm{w}}^{\mathrm{t}}$	Wheel tractive efficiency	Section 1.3
$\eta_{\mathrm{w}}^{\mathrm{tr}}$	Wheel transportation efficiency	Section 1.3
η_x^{tr}	Vehicle running gear transportation efficiency	Section 1.4
η_δ	Vehicle slip efficiency	Section 1.4
$\eta_{\delta\mathrm{w}}$	Wheel slip efficiency	Section 1.3
$\eta_{\Sigma\mathrm{w}}$	Wheel total efficiency	Section 1.3
γ	Lead angle of the worm thread	Section 4.5
λ	Lagrange factor	
λ_i	Normal loading factor	Section 2.8
λ_{w}, γ_{w}	Longitudinal elasticity coefficient of a tire	Section 1.3
μ	Dynamic friction coefficient	Section 2.3
μ_{l}	Peak friction coefficient in lateral direction of a tire	Section 1.3
$\mu_{\mathrm{p}x}$, μ_{p}	Peak friction coefficients in longitudinal direction of a tire (also grip coefficients)	Section 1.3
μ_{s}	Static friction coefficient	Section 4.1
μ_x	Current friction of grip coefficient (friction coefficient ''in use'')	Section 1.3
ν_{c}	Angle of pinion rotation	Section 2.3

$(\pi m - S_t)$	Width of space	Section 2.3
φ_k	Angle of the working edges of the cams. Slope of the V-shaped groove in the differential's case	Sections 4.3 and 4.4
θ	Wheel angle of rotation	Section 1.3
θ_n	Surface longitudinal angle of inclination	Section 1.4
ρ	Friction angle and dynamic friction coefficient	Section 2.3
ω_a	Angular velocity of a vehicle about the vertical axis	Section 2.8
ω_c	Angular velocity of a pinion about the carrier	Section 2.1
ω_{rel}	Relative angular velocity of a side-gear and the case of a differential	Section 2.1
ω_w	Wheel angular velocity	Section 1.3
ω_0	Angular velocity of the input element of a power dividing unit	Section 2.1
ω_{01}, ω_{02}	Angular velocities of the output shafts of an interaxle power dividing unit	Section 1.3
ω', ω''	Angular velocities of the output shafts of an interwheel power dividing unit	Section 2.1

Subscripts

r, l	right, left	Section 2.3
′ and ″	Indices relating to the left/right or right/left wheels	

Note: This list contains symbols that are used through the whole book. Symbols, which are only used in a particular section, are not listed here.

1

Driveline Systems and Vehicle Performance

1.1 Brief Review of Driveline Systems History

1.1.1 First Wheeled Carriages

The first motorized wheeled carriage capable of moving under its own power was a steam-driven vehicle designed by a French army engineer, N. J. Cugnot (Figure 1.1). This vehicle was constructed in 1769.

With all its shortcomings, the steam-driven vehicle—Cugnot's chariot—had a rational design. It had the most simple driveline system combined with a most simple steering mechanism. All this stemmed from the fact that the force cylinder was most successfully located above the forward drive and, simultaneously, the steering wheel. The vehicle's weak spot consisted of its using a ratchet and pawl transmission mechanism that made it impossible for the vehicle to move uniformly. In addition, because of its imperfect steam engine, Cugnot's chariot stopped every 10 m to allow steam to accumulate in the boiler and to rise to the required level.

In 1801, an enterprising English inventor, Richard Trevithick, constructed the first steam-driven passenger stagecoach and organized, for the first time in history, mass construction of such coaches in the insular part of Great Britain. The rear driving axle of the coach with wheels rigidly fastened to it was driven by a pair of gears from an intermediate transmission shaft. The design of the wheels, taken from horse-driven carriages and their rigid coupling with the driving axle, was the weak link of the first such coaches. The driving wheels, with their smooth and narrow rim, had poor traction with the road and frequently skidded. This was remedied by equipping the driving wheels with an additional device, consisting of a set of pushers and detents—''claws''. Figure 1.2 presents another vehicle of a similar design of the claws. Performing reciprocating motion were a pair of claws hinged on rods that became alternately coupled with the road surface and assisted in rotating the driving wheels of the carriage without perceptible skidding.

Subsequently, in 1813, somebody by the name of Brunton invented pushers and used them as the principal and sole propulsion device on the primitive locomotive constructed by him (Figure 1.3a). Without exception, all the wheels in Brunton's locomotive were driven (no torque applied), and functioned as a support structure. It should be noted that at that time few believed in the traction ability of carriage wheels. Even such an experienced mechanic as Trevithick did not fully trust wheels with smooth rims. In particular, he did not trust the wheels of the locomotive that were supposed to roll on smooth rails. For this reason, he placed forged nails on the rims past the flanges of the driving wheels of his two first locomotives constructed in 1803 (Figure 1.3b). The nails

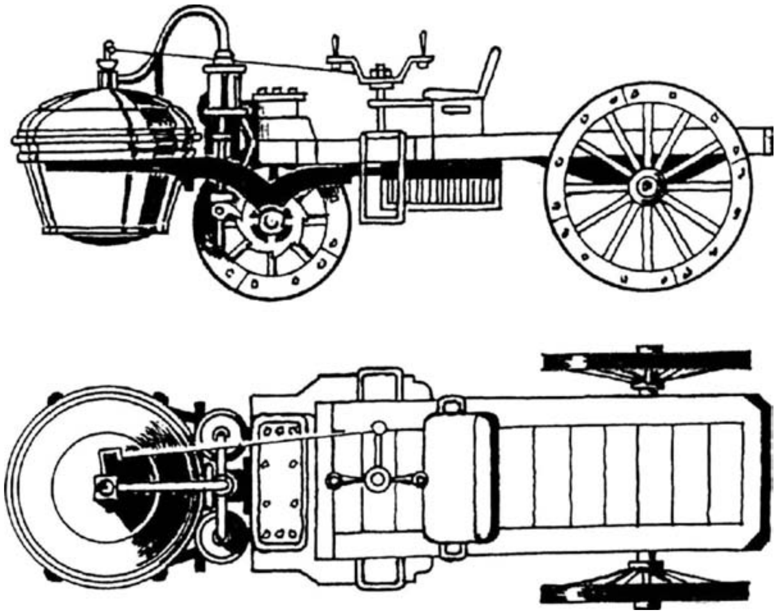


FIGURE 1.1
First self-propelled steam automobile designed by French army engineer N. J. Cugnot.

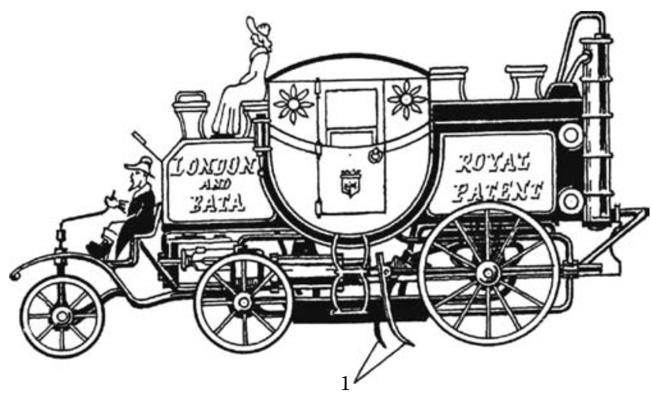


FIGURE 1.2
H. Herney's ''steamer'': 1, Coupling ''claw.''

stuck into wooden beams placed along the rails, thus significantly improving the coupling between the wheels and the additional support surface and, hence, improved their traction.

One of the nonrail carriages constructed with a drive similar to that of Brunton's locomotive was the carriage of D. Gordon. It was constructed in 1824, and consisted of a three-wheeled machine—a steam-driven stagecoach (Figure 1.4). All the wheels in this stagecoach were supporting driven wheels, whereas the propulsion was provided by a ''pusher-leg'' system with a complicated lever-type drive. In this manner, by placing additional devices in the form of ''pushers'' and ''claws,'' the designers of steam-driven carriages gradually and surely approached the invention of a wheel with a tread. It is

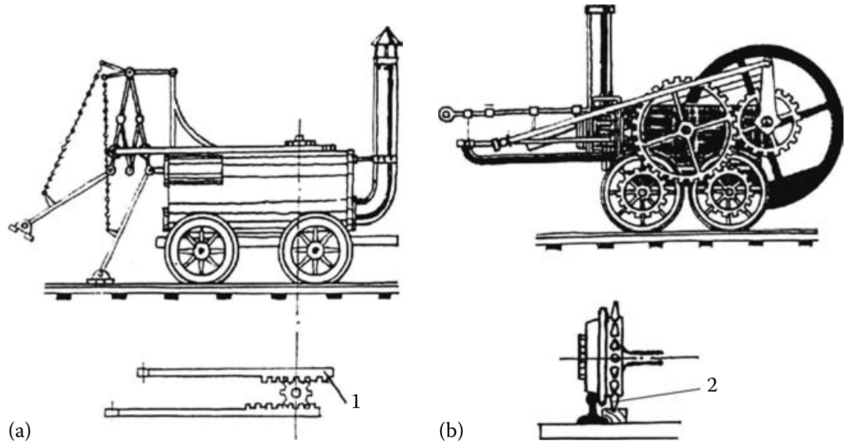


FIGURE 1.3
Locomotives: (a) Brunton's locomotive (1, drive of pusher ''legs''); (b) Trevithick's locomotive (2, wheel pair with nails on the rims).

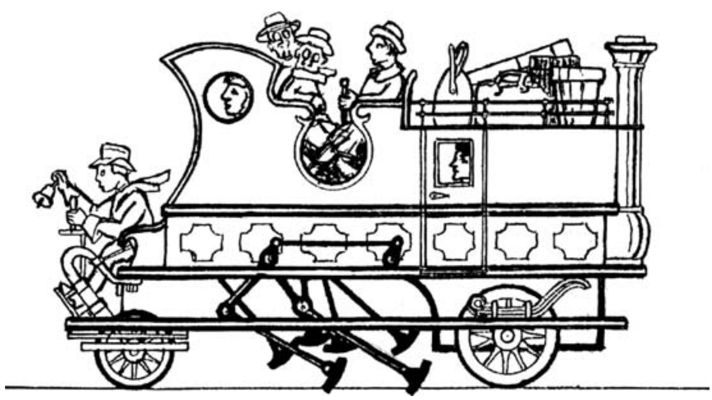


FIGURE 1.4
Gordon's three-wheeled stagecoach with a system of pusher ''legs.''

precisely such a wheel that, in the majority of cases, is capable of ensuring reliable traction between itself and the road surface.

In 1805, an American mechanic, Oliver Evans, constructed a suction dredge for cleaning up the Philadelphia harbor waterfront. According to the inventor, the dredge was supposed to deliver itself to its place of work. For this purpose, Evans equipped it, together with an aft propulsion screw, which served for floating, five wheels (Figure 1.5) that would allow it to move on land. The leading wheel together with the wheels of the front turning axle allowed the dredge to take turns, whereas the wheels of the rear driving axle provided the traction. The rear driving axle of the dredge was driven by a steam engine by means of a belt drive. The *Orukter Amphibolas* (which was the name of the dredge) was the first motorized amphibious vehicle.

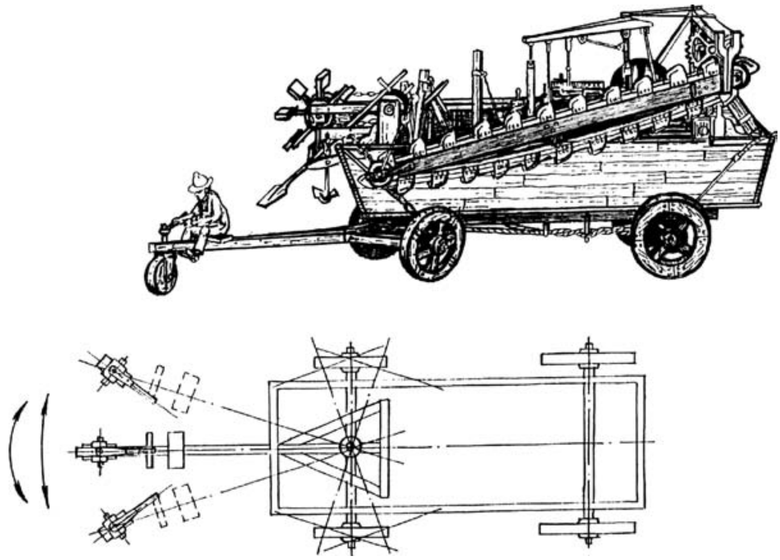


FIGURE 1.5
Oructur Amphiblus—dredge designed and constructed by Oliver Evans.

1.1.2 First Differential Transmissions

One of the problems that were encountered was that the driving wheels of the first steam-driven vehicles were rigidly coupled to their driving axles. Because of this, each of the driving wheels could not roll along its own path that were formed by irregularities in the road ahead each of them. Two wheels on a single rigid axle in translational motion traversed different paths over the same time interval. For this reason, one of the wheels skidded while the other slipped, which was detrimental to the machine's traction. Moreover, the rigid fastening of the wheels did not allow the driving wheels to roll at different angular velocities (over different paths) when the vehicle made turns. This situation brought about the invention of the differential transmission (henceforth referred to as differential).

The differential was invented by a French watchmaker, Onesiphore Pecquer, in 1827 and was installed by him in 1828 in his private steam-driven vehicle. From this time on, the differential started appearing in other steam-driven vehicles and, subsequently, in the drivelines of vehicles with engines driven by petroleum-derived fuels, i.e., in driveline systems of automobiles. The differential separated the driving axles of the machine into two half axles with driving wheels, allowing each of them to roll along its own path. This reduced the loss of power incurred in the skidding of wheels when their traction with the road surface was sufficient.

The first differentials were extremely simple. They divided into half the torque that became converted into the traction of the vehicle between the driving wheels of the axle. As a result of this, the torque to both driving wheels was that utilized by the wheel that had the poorer contact with the road surface. This is a negative property of the simple differential. It causes the vehicle to come to a full stop when one of its driving wheels spins on a slippery part of the road, i.e., it brings about complete loss of mobility by the vehicle. Such differentials are now known as open, or free differentials.

The mobility of a wheeled vehicle and, in particular, of an automobile refers to its ability to travel over poor roads and rough terrain while transporting loads, delivering the maximum possible work output. Automobile utilization practice puts precisely such requirements to the vehicles. In order to improve the vehicle's ability to travel over difficult surfaces, it is necessary to force the differential to divide the torque supplied by the engine among the driving wheels in such a manner that the wheels would exert a torque in accordance with the conditions of their gripping with the surface of motion. The solution to this problem began when automobile manufacturers started inventing various devices that lock the half-shafts of the driving axle, i.e., lock the differential itself. In this manner, the driveline systems acquired locking differentials of various design. This is discussed further down in this chapter. The properties of current differentials are presented in Chapters 2, 4, 5, and 7.

In 1878, a French engineer, Amede Bollee Sr., together with his son, also named Amede (Amede Bollee Jr.), constructed a steam-driven vehicle that they named *La Manselle* (Figure 1.6), which had an original packaging and a very simple layout. The driveline system of its driving wheels had, in addition to the previously mentioned gear and belt transmissions, a universal-joint drive coupled with a chain drive. Unfortunately, the drive

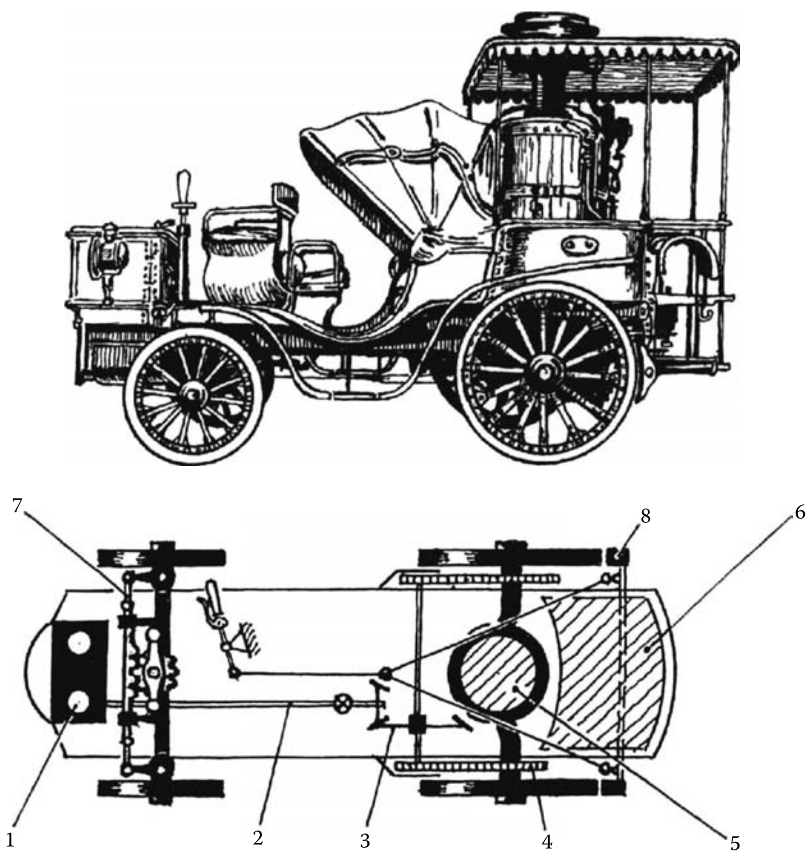


FIGURE 1.6
La Manselle constructed by Bollee Sr. and Bollee Jr.: 1, engine; 2, cardan propeller shaft; 3, final drive; 4, chain drive; 5, boiler; 6, water tank; 7, hinged steering-gear parallelogram; 8, shoe brakes.

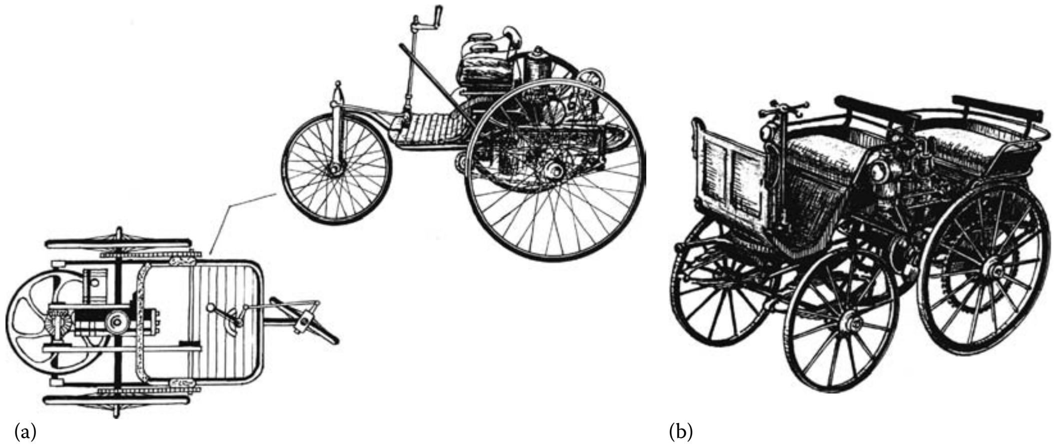


FIGURE 1.7
The first automobiles: (a) by Benz; (b) by Daimler.

of Bollee's steam-driven vehicle lacked the simplest of differentials that was already available at that time.

The first world-acclaimed automobiles manufactured by Carl Benz and Gottlieb Daimler also lacked differentials in the drives of their driving wheels. The driving wheels of the Benz automobile were driven by a chain transmission, whereas the Daimler car was equipped with a gear transmission (Figure 1.7).

The differential in the drives of automobiles built by these carmakers appeared somewhat later. Starting with the Panhard–Levassor automobiles produced in 1891 (Figure 1.8), differentials have gradually come into use in the drives of all vehicles, both on the intermediate axles and, directly, on the driving axles (Figure 1.9).

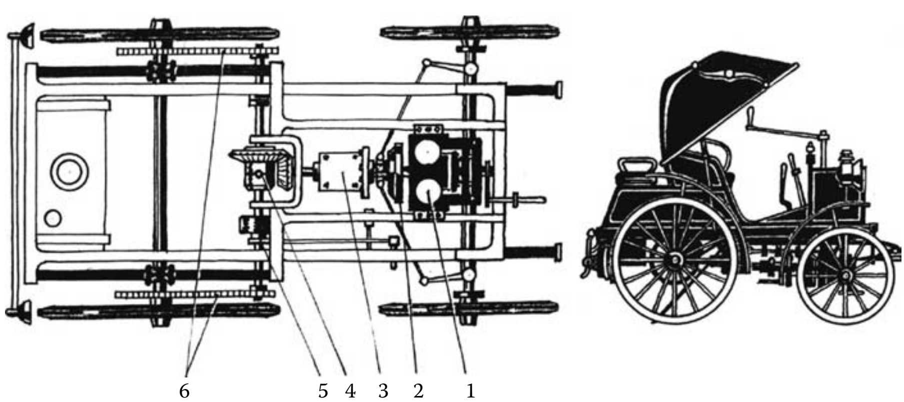


FIGURE 1.8
Schematic diagram and driveline of the Panhard–Levassor automobile (designed by Emile Levassor); 1, engine; 2, main clutch; 3, gearbox; 4, differential with final drive; 5, transmission brake; 6, chain drive.

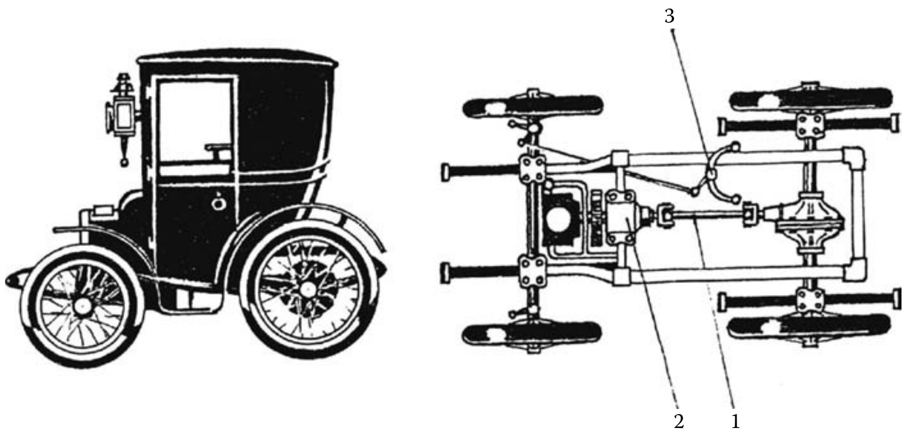


FIGURE 1.9
View and design of a Renault automobile: 1, cardan propeller shaft; 2, gearbox; 3, bicycle-type steering gear.

1.1.3 First Hybrid Cars and All-Wheel Drive Vehicles

In 1900, at the request of E. W. Hart, a resident of Luton, Austria, Ferdinand Porsche, an engineer employed by the Joined Lohner company, constructed a four-wheel drive automobile driven by electric motors built into the wheels. These motors were supplied with electricity from storage batteries that weighed 1800 kg. It turned out somewhat later that the weight of these batteries was an insurmountable obstacle to the making of such vehicles. For this reason, Porsche replaced the storage batteries by an internal combustion engine with an electric generator and, thus, obtained a ''hybrid drive'' automobile. In 1903, he was issued a number of patents on this automobile and started manufacturing such vehicles.

The first hybrid Porsche automobiles had only two driving wheels and, even if they were a great improvement over the four-wheel drive vehicle with storage batteries, their speed was insufficient for participating in races. And then Porsche, who was a devoted enthusiast of car races, redesigned his hybrid automobile as a four-wheel drive vehicle. This happened in 1903. In the same year, Porsche participated with this car in races that took place not too far from Vienna. It is quite possible that the Porsche–Lohner hybrid-drive automobile was the first 4×4 formula all-wheel drive car.

1.1.4 Front-Wheel Drive Designs

The first front-wheel drive (FWD) automobile appeared in 1903. This was a unique vehicle designed by an American, John Walter Christie (Figure 1.10). Its uniqueness consisted in the fact that the beam cross-member of the front nondrive axle consisted of the engine, which was located transversely to the longitudinal axis of the vehicle, with a crankshaft that was coupled through reduction gearing to the driving wheels. The rear axle of the vehicle was driven and steered.

Christie's FWD vehicle demonstrated the many advantages of this kind of drive over the rear-wheel drive that became standard at that time. It was found to impart good course stability to the car. Inspired by this, Christie established the Christie Direct Action Motor Car Company. This company produced only a limited number of cars with Christie's

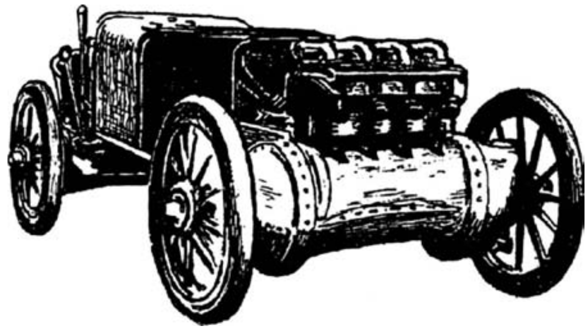


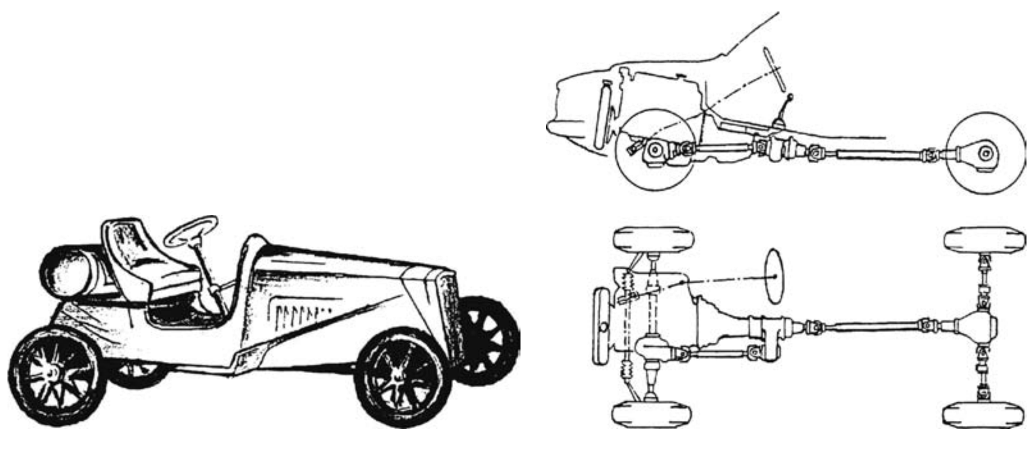
FIGURE 1.10
J. W. Christie's FWD automobile.

FWD, because they were not in great demand. For this reason, he had to switch to the manufacture of high-power two-wheel (single axle) tractors that served for pulling steam-driven fire pumps.

The development of FWD vehicles started in earnest during the 1920s and 1930s. Harry Miller and Leo Goosen constructed a pair of radical front-drive cars late in 1924. Miller managed to solve the problem of getting enough weight on the front driving wheels, given his rather long, north-south-mounted engine. Miller created an extremely compact transmission and differential in one case, the first transaxle.

1.1.5 First Mechanical Four-Wheel Drives

The first all-wheel drive automobile with a 4×4 formula and a purely mechanical drive appeared in 1903. It was exhibited at the Paris Automobile Salon by the Dutch Spyker company (Figure 1.11a). The drive of this car's wheels was designed in a manner that rapidly became classic (see Figure 1.11b). It had an additional gearbox and, in it, a locking



(a) (b)

FIGURE 1.11
All-wheel drive automobiles: (a) 4×4 automobile—the Dutch Spyker; (b) classical design of a 4×4 automobile in which the front wheels were engaged by the driver by means of a tooth-type clutch.

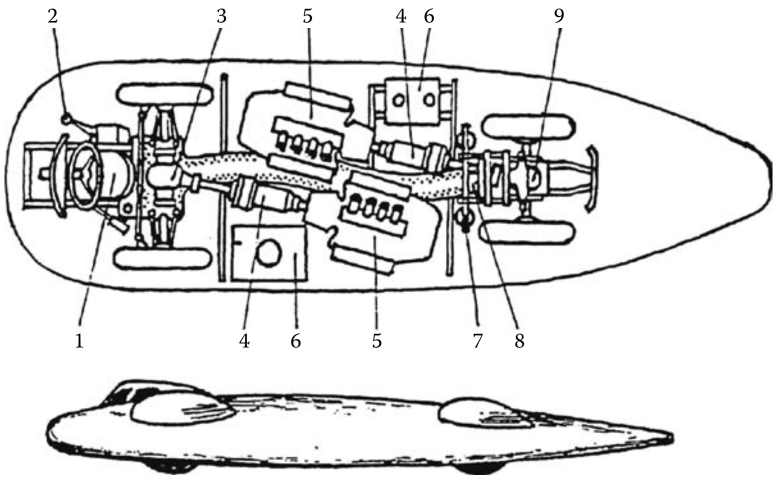


FIGURE 1.12
Railton's 1947 4×4 racecar. 1, driver's seat; 2, gearbox lever; 3, front axle; 4, gearbox and transmission brake; 5, engine; 6, fuel and oil tanks; 7, pneumatically controlled brake cylinder; 8, chassis with the central tube of the body skeleton; 9, rear axle.

dog clutch that could be used for coupling the engine to the front axle when it became necessary to increase the car's traction. The power to the steered and at the same time driving wheels of the front axle was transmitted by means of a universal-joint drive and single Hooke's joints with an improved spider.

It is interesting to note that, at approximately the same time, in 1906, the world's first armored car with two driving axles was constructed by Paul Daimler, the engineering director of the Austro-Daimler company.

In 1905, J. W. Christie constructed a 4×4 car in which the wheels of each of the axles were set into motion by a separate engine. This car was used by Christie personally to participate in the Vanderbilt Cup and other American auto races. The machine did not live up to expectations. The races were along an elliptical path with a large radii of curvature. Nevertheless, Christie's four-wheel drive vehicle took poorly even such turns because of the excessively large kinematic discrepancy between the rotation of the front and the rear wheels, each of which was powered by a separate engine, and also because of the use of single Hooke's joints in the drives of the front wheels.

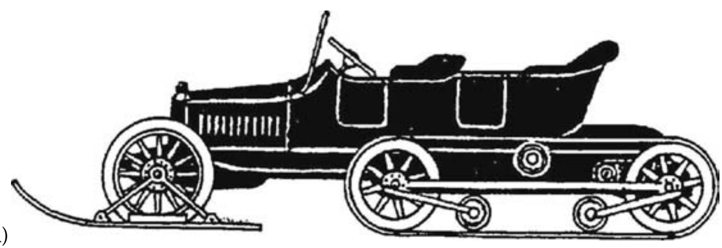
A second attempt to construct a 4×4 car with two engines was undertaken in 1947 by an Englishman, Railton Reid. He used the same design for constructing a racing car (Figure 1.12). When driven by racing-car driver John Cobbs along a straight path, it attained a speed of 634.26 km/h. This was a new speed record. As to handling turns, this car suffered from the same shortcomings as Christie's vehicle.

1.1.6 Invention of Pneumatic Tires and Design Measures for Improving the Poor-Terrain Mobility of Vehicles

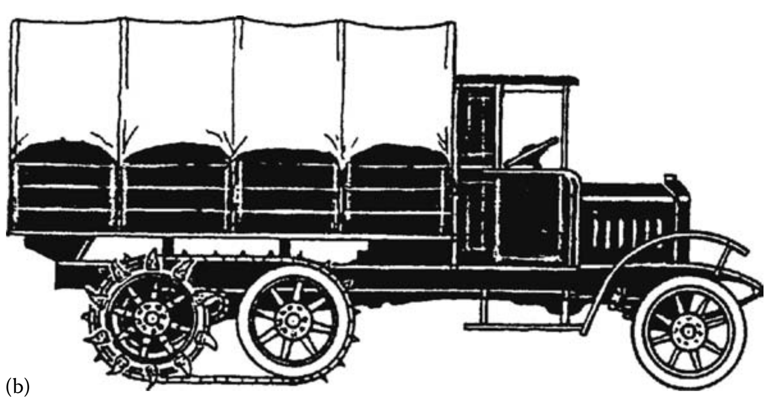
The invention of the pneumatic tire was an epoch making event for the development of the car and of wheel-drive designs. In 1888, John Boyd Dunlop, a veterinary surgeon, was issued the English Queen's patent for the bicycle pneumatic tire invented by him. Eleven years later, in 1899, the French brothers Edoard Michelin and Andre Michelin were the first to ''shoe'' an electric racecar with pneumatic tires. As of that time, the two brothers, who established the Michelin Pneumatic Tire Company, subdivided the tire into a chamber and a cover so that it could be easily changeable. The cover had shallow grooves on the tire tread that, according to the inventors, should have ensured adhesion with the road surface. All this was indeed true under good road conditions. However, when a car equipped with tires having shallow grooves was called upon to travel under poor terrain conditions (sandy, swampy terrain), over wet dirt roads, and also over snow and ice encrusted roads, its wheels started spinning and the car completely lost its ability to travel. This forced car manufacturers and tire engineers to find a way to improve the mobility of their products. The first addressed this problem by looking for methods of locking the differentials and inventing self-locking differentials; this included inventing such means for increasing the gripping of the supporting surface by the driving wheels as chains wound on individual wheel rims and caterpillar tracks (Figure 1.13). The latter were termed half-track drives. The others—tire engineers—focused on improving the grooving pattern and the tire tread design.

Adolphe Kegresse, a Frenchman, employed as a mechanic at the Russian Imperial garage, was one of the first persons to invent and use the half-track drive for automobiles of the Russian Emperor. His invention was intended for improving the mobility of vehicles on snow-covered roads and unpaved stretches of land. Kegresse patented this invention in 1912.

Following Kegresse's caterpillar drive design, a composite rubber-metal caterpillar track, that was nicknamed ''overall,'' was designed in the United States. It was simply slipped



(a)



(b)

FIGURE 1.13
Half-track running gears: (a) by Adolphe Kegresse on the Russo-Balt automobile; (b) with ''overall'' caterpillar tracks and grousers made by the Henschel Company on the rear wheels of a truck.

FIGURE 1.14
Snowmobile, U.S.S.R., 1961.

over the pair of rear wheels of army and commercial six-wheeled trucks and also of armored vehicles. From this time, the half-track drive with an overall type caterpillar track came into use also in other countries, primarily on trucks. For example, in 1934, the Henschel company (Germany) used overall-type rubber-metal tracks, coupled with special metal grousers, on its trucks (see Figure 1.13b). German armored carriers equipped with half-track drives did not exhibit sufficient mobility, particularly in snow and mud. Attempts were made to design vehicles with half-track drives and skies instead of leading wheels (see Figure 1.14). It may apparently be stated that the half-track drive did not succeed in becoming an alternative to four-wheel drive vehicles with various types of driveline systems.

However, developments in tire design promoted the development of four-wheel drive vehicles. Their mobility was highly improved by the invention, during the 1920s, of low-pressure tires. These tires have a reduced rolling resistance and larger than ordinary contact patch between the soft tire and the surface, which improves the vehicle's mobility.

It can be claimed that tire designers started, during the 1940s, to actively improve the grooving patterns and design of the tire tread, which should highly improve the mobility of vehicles. As a result, there appeared high-mobility tires with developed tread (Figure 1.15) for use primarily in off-road vehicles. Intensive work was also done to design tires that would provide improved mobility to both off-road vehicles and those intended for paved-road travel. Here is an example.

A tire with a wide annular groove over the center of its tread was designed. The groove served for removing water from the pavement-tire contact patch. For this purpose, the groove actually divided the patch into two distinct contact spots. Its cone-shaped transverse grooves together with the longitudinal annular groove rapidly remove water from both contact spots. Both hard halves of the tread provide the tire with good directional stability.

There are also other approaches to improving tires for enhancing the mobility of vehicles. One such example designed in the 1990s has the wheel rim equipped with wedge-shaped flanges for preventing the beads of the tire from sliding into the rim. It has a device that

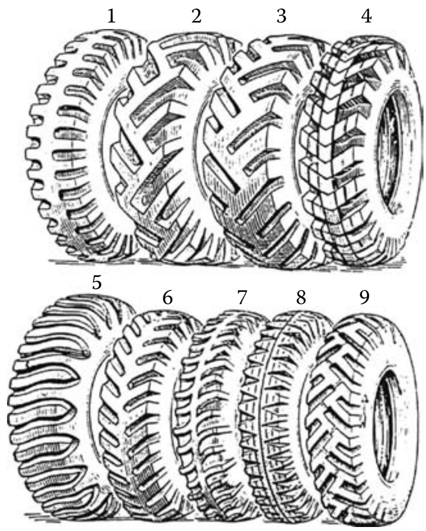


FIGURE 1.15
Drawings of high-mobility (all-terrain) tire treads: 1, straight herringbone; 2 and 3, skewed herringbone (''ribbed''); 4, split herringbone; 5, spiral (asymmetric about the longitudinal axis of the wheel); 6, herringbone; 7, semispiral (''skeleton''); 8, drawing with longitudinal slots; 9, split skewed herringbone.

signals a drop in the tire air pressure. The tire has two detents that take up the weight of the vehicle upon a drop in the inner-tube air pressure. The shape of the wedge-like connecting strips of the rim assists in holding the sidewall of the tire in it even when the air pressure drops to the atmospheric pressure. This tire-wheel arrangement allows driving a car with a fully or partially flat tire for 80–90 km. This arrangement is intended primarily for ensuring passenger safety when a tire goes flat. However, it can be easily seen that this system also protects the car from total mobility loss. Recently, in 2008, an active self-inflating tire system was introduced. The technical idea is based on peristaltic pump principles. To inflate the tire, the normal load and motion of the vehicle are used.

It may be concluded that improvement of tire design was one of the most significant conditions for the intensive development of multiaxle drive vehicles.

1.1.7 Constant-Velocity Joints

Another factor that delayed the appearance of not only front-drive but also all-wheel drive vehicles, in which the engine's power is utilized to a greater extent by being transformed into their traction capacities and acceleration performance, was the lack of constant-velocity joints.

The problem was that single Hooke's joints could not transmit uniform rotation to the steered wheels of the vehicle, particularly when taking a turn. For this reason, the invention of constant-velocity joints became one of the first priorities for vehicle designers from the very start of the twentieth century.

In 1925, a German design engineer and scientist, Richard Bussien, linked two single Hooke's joints, and thus transformed a nonconstant velocity joint into a new constant-velocity mechanism. He used his invention on a front-drive passenger car constructed by him and called it VORAN (Vorderrad Antrieb—FWD). It turned out that the rather bulky dual Hooke's joint that caused the steered wheel to have a too-large overhang

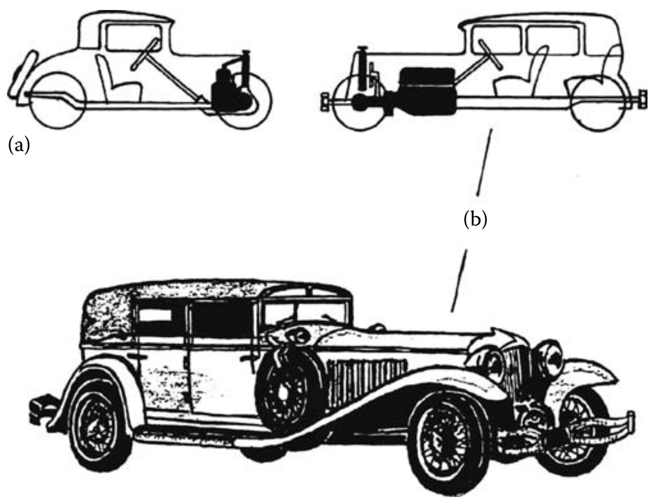


FIGURE 1.16
Front-wheel drive automobiles: (a) schematic of the Trakta vehicle (1926) designed by Jean-Albert Gregoire; (b) schematic and view of the Cord-L-26 automobile (1929) designed by E. L. Cord.

(kingpin offset) is not suitable for passenger cars. A steered wheel with a large kingpin offset exerts a greater resistance to turns and causes the machine difficult to steer. As a result, very few VORAN passenger cars were sold and their manufacture ceased.

Jean-Albert Gregoire, an engineering designer and entrepreneur, started to design in 1925 and constructed in 1926 a FWD vehicle that he called *Trakta* (Figure 1.16a). This was the first FWD vehicle with satisfactory handling. Satisfactory handling was achieved by a constant-velocity joint designed and constructed by Gregoire's companion, Pierre Fenay, a passionate devotee of the four-wheel driveline. Since Fenay's joint was used for the first time on the Trakta automobile, it has automatically acquired the car's name and it was patented in 1926 under the name Trakta.

The Trakta joint was much more compact than the constant-velocity dual Hooke's joint (Figure 1.17a). As was already said, shortly before this, Bussien attempted using the dual Hooke's joint for the steered driving wheels of a passenger car. This was not successful. Gregoire's Trakta avoided the fate of Bussien's automobile by using Fenay's joint.

The Fenay constant-velocity joint was followed by a similar joint designed in 1927 by a Czech engineer, Rzeppa, that was patented under the name of the Rzeppa joint and others (see Figure 1.17). Thus, between 1923 and 1930, there was a boom in inventing constant-velocity joints.

Toward the end of the 1930s and the beginning of the 1940s, the Trakta constant-velocity joint was used in the first of Erret Loban Cord's FWD vehicles. Cord's cars had an unusual appearance because of the very long engine hood (see Figure 1.16b). This was caused by the fact that the clutch mechanism as well as the gearbox were located in front of the engine, rather than at the back of it, as in the classical arrangement. This new driveline system turned out to be rather poor. It reduced the load on the front, both steered and driving axle, which impaired the vehicle's traction and its stability in taking turns. For this reason, front drive Cords did not last too long. But still, in spite of their short life, they left behind a rather happy memory. They are remembered as America's first cars with a fully shrouded radiator and having the longest engine hood.

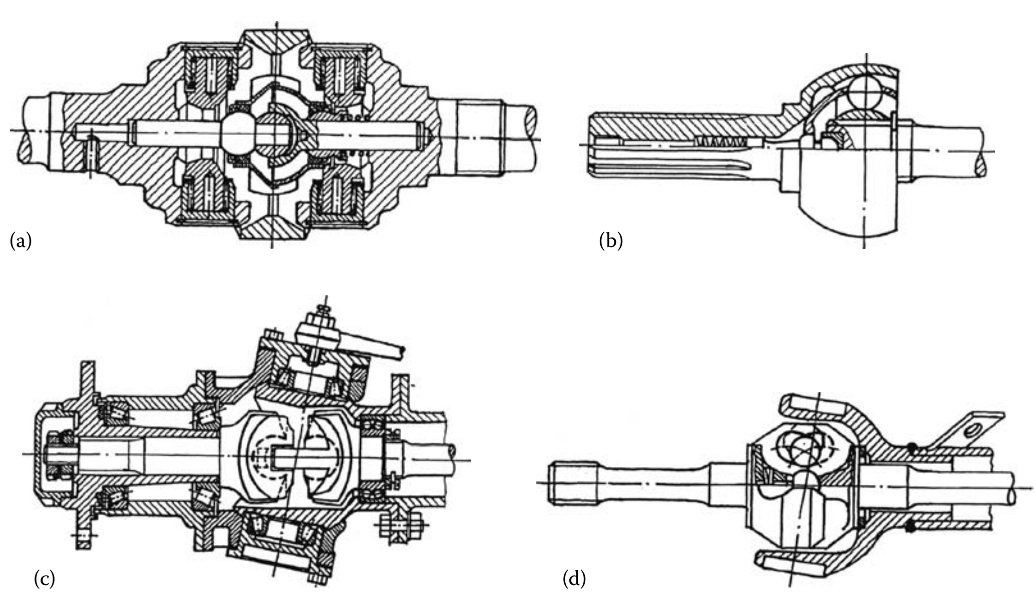


FIGURE 1.17
Equal-velocity joints: (a) Hooke's double joint; (b) Rzeppa's joint; (c) Bendix–Trakta joint; (d) Bendix–Weiss joint.

1.1.8 All-Wheel and Multiwheel Drive Trucks and Passenger Cars

As early as 1911, all-wheel drive automobiles designated as FWD started rolling out from the gates of the factory owned by Otto Zachow and William Besserdich. The FWD automobiles utilized the previously mentioned Dutch Spyker vehicle drive, but they differed by utilizing new ball joints in that part of the drive that provided for transmission of the engine power from the differential to the front, steered wheels of the car. These were by now constant-velocity joints. They were invented by one of the designers of the FWD vehicle—Otto Zachow. His joints, as opposed to the familiar single Hooke's joints, and the first Clarence Spyker joints, transmitted to the steered wheels of the car uniform rotation, this means that they were those first constant-velocity joints, which were badly needed by car manufacturers for setting up mass production of all-wheel drive vehicles. For this reason, starting with the FWD car, the U.S. automobile fleet started rapidly filling up with all-wheel drive vehicles.

In 1922, the French company Renault designed and started manufacturing the first in the world three-axle truck, the Renault MN. It was the first to use doubled wheels on all the axles, including the front steering axle.

The results of mobility tests of the Renault MN under difficult road conditions, even when using low-power engines (from 10 to 25 HP) that this machine used, exceeded all expectations. Three-axle trucks with doubled wheels were in no way inferior to the Citroen-Kegresse all-terrain (half-track) vehicles on poor roads and even exceeded them with respect to many performance indicators.

The only serious shortcoming of the Renault MN was its poor turnability. This stemmed from the fact that the forward, steering axle also had doubled wheels. Nevertheless, triple-axle trucks with doubled wheels enjoyed a very good reputation in spite of the doubled wheels on the steering axle. Their reputation improved even further when the Renault MN trucks completed a trip over African desert sands. The large supporting surface of the doubled wheels prevented the machines from sinking and digging deeply into the sand.

Up to 1931, it was up to the driver to decide whether to engage the front wheels. However, in the case of high-speed cars the advantage of all-wheel drive should be utilized permanently and independently of the driver. This can be achieved simplest by inserting into the drive between the driving axles of the car an open (also called free), symmetrical bevel-geared differential that divides the torque into half, i.e., by introducing a permanent drive between the two axles. This is the conclusion that was drawn by Ettore Bugatti, an Italian design engineer who lived in Elzas, France. Having ascertained from experience that the two driving wheels of a four-wheel automobile are not capable of fully utilizing the torque produced by a large-volume engine, he constructed three first 4×4 racing cars with a permanent interaxle drive of all four wheels, located between the two axles. His machines were known as *Bugatti-Type53*. The torque to the front steered wheels of these machines was transmitted by a double Hooke's joint. The Bugatti-Type53 machines had a reputation of being difficult to drive, as it became evident during their participation in races, because of the use of the double Hooke's joint. In connection with this, Bugatti's automobiles were used primarily for races on straight tracks that passed over rough terrain.

Starting with 1932, the English companies, Grey, Leyland, and Armstrong Siddeley, started constructing multiaxle, multiwheel drive and all-wheel drive trucks with complex arrangements for driving the axles and the driving bogies (or tandems). At that time, a vehicle similar to those mentioned above was constructed at the Yaroslavl Automotive Vehicles Plant in the U.S.S.R. (Figure 1.18). This was an all-wheel drive truck with an 8×8 wheel formula with steered and driving wheels in the front tandem and two driving nonsteered axles in the rear tandem. The steered wheels of each of the axles of the front tandem are coupled, as shown in Figure 1.18, to the differentials of their axles by means of drive shafts with Hooke's single universal joints. This stems from the fact that both axles of the front tandem are of the De Dion design. The drive of the ЯГ-12 vehicle, including the drive of the rear tandem, uses a total of 9 drive shafts with 18 Hooke's single universal joints. The second axle of the front tandem, as the first axle of the rear tandem, is of the drive-through type. They employ an original drive-through bevel-gear final drive shown in the figure. It should be noted that the ЯГ-12 machine, which was the name given to the above Yaroslavl plant product, left a good impression. It exhibited high mobility. It handled relatively easily up to 1.5 m wide trenches, fords, slopes of up to 30°, confidently moved over mud and deep snow; was capable of attaining speeds of 40–45 km/h on roads with minimum power consumption, at that time for this class of vehicles—52 L per 100 km of travel.

Ferdinand Porsche designed, in 1938, the folk passenger car the Volkswagen Beetle which was produced during World War II as a reconnaissance amphibian vehicle with a distinctive body. It was known as the ''pail-car'' (*Kubelwagen* in German) (Figure 1.19a). This was an all-wheel drive vehicle. It was the first car in which the drive of all the four wheels differed highly from the classical arrangement. The in-line four-cylinder engine of this vehicle was located in the rear part of the body. The engine, transmitted power to the rear driving wheels through a gearbox located past the engine—first to the rear axle and then from that axle to the front wheels—via a cardan drive. This 4×4 driveline system turned out to be simpler than the classical one. It still employed a dog clutch, that allowed the driver to couple also the front wheels to the engine, but it dispensed with a separate transfer case. This driveline system is called ''pure.'' It has attained final purity in the 1980s and 1990s. At that time, it will be put to use in the Audi Quattro car, where it was rotated through 180°, since the engine in the Audi Quattro is located ahead of the leading axle. Subsequently, Porsche himself continued using this driveline system in his all-wheel drive vehicles.

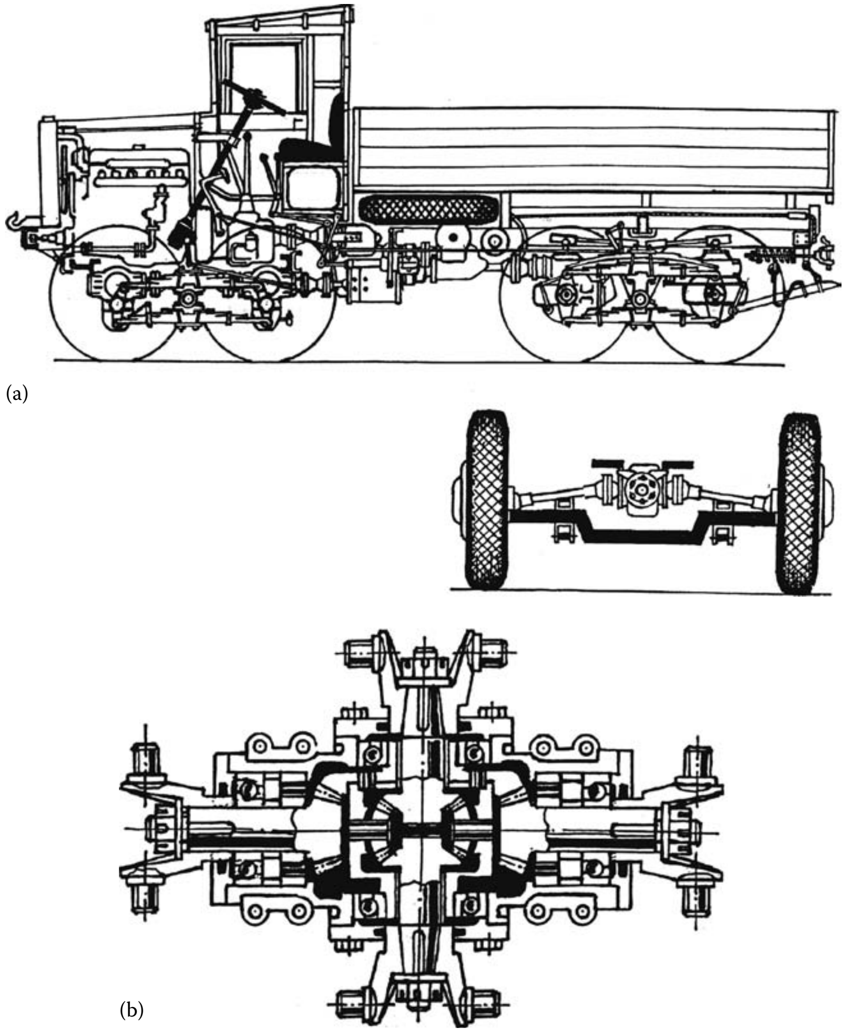


FIGURE 1.18
Design of the first Soviet-made four-axle, 8×8 formula truck, ЯГ-12, with two front-steered axles: (a) De-Dion-type steered and driving axle; (b) final drive of the drive-through axle.

While Porsche was working on his new 4×4 wheel driveline system, the U.S. industry continued producing such machines with the classical 4×4 driveline system. Thus, Karl Probst, an engineer working for the American Bantam company developed a design of a classical all-wheel driveline system for an army vehicle with a highly simple, open body (Figure 1.19b). The machine was manufactured and became a part of the U.S. army vehicle fleet. At first, it was called the ''Mobile Bantam,'' then ''Willys MB,'' and, finally, ''Ford GPV'' (Ford General Purpose Vehicle). During World War II, it was used, under the last name, as a light artillery towing tractor and staff car. Light artillery all-wheel drive vehicles with the classical driveline similar to the American ones were also produced in the Soviet Union under the brand name GAZ-67 (Figure 1.19c)

In 1947, an engineer, Spen King, employed by the British Rover Company designed a gas-turbine passenger car, the T3 Rover (Figure 1.20). This was a four-wheel drive

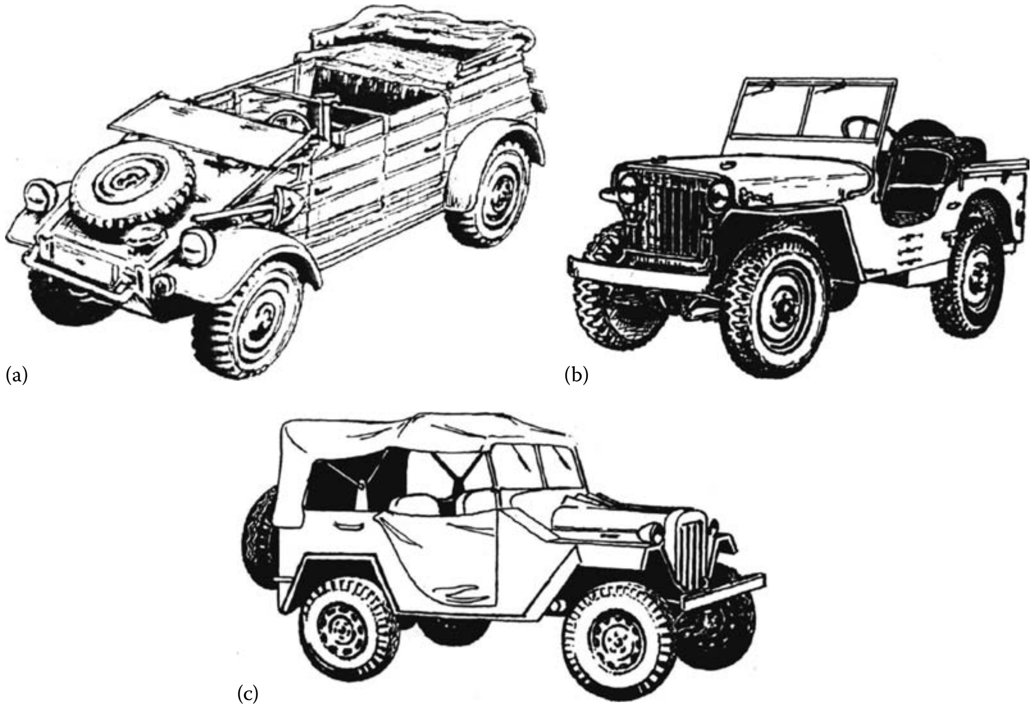


FIGURE 1.19
4×4 automobiles: (a) Kubelwagen with ''pure'' drive; (b) Ford GPV with classic drive; (c) GAZ-67 with classic drive.

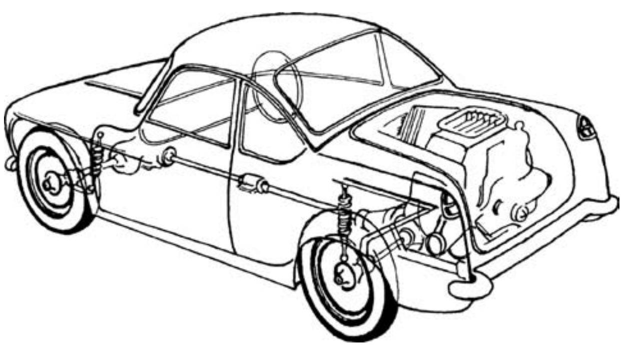


FIGURE 1.20
First gas-turbine 4×4 formula highway automobile—the T3 Rover (1947).

highway car. Its gas-turbine engine was located in the back. For this reason, the power from it was transmitted to the front axle from the rear axle. The two axles were coupled by an overrunning clutch. The kinematic discrepancy between the axles built into the driveline (analytic material on kinematic discrepancy is presented in Chapter 3), caused the clutch to disengage the front axle of the car when it took a turn. For this reason, the leading wheels could rotate when taking a turn at a higher speed than the rear wheels. At that time, it was a new progressive technical solution.

The driveline of the T3 Rover was very similar to that of the Kubelwagen. They differed from one another by the fact that the driveline of the former employed an overrunning clutch whereas of the latter—a dog clutch.

A system similar to that of the T3 Rover driveline, except that rotated through 180° because of its being located at the head of the engine and with an interaxle differential instead of the overrunning clutch, was used at the end of the 1970s the Audi-200 Turbo Quattro. It was noted by a large number of automobile manufacturers, impressed them, and has advanced it toward further development of a ''pure driveline.''

Starting with 1948, Rover started producing an all-wheel automobile with a permanent drive to the wheels under the name ''Landrover'' which, unlike the T3 Rover, had an ordinary internal combustion engine.

In the 1940s and 1950s, permanent four-wheel driveline systems of medium- and heavy-duty trucks came into extensive use. Dr. A. Kh. Lefarov headed a team during this time that designed a new 4×4 timber truck with an interaxle differential and redundant locking capacity, the first in the former Soviet Union (Figure 1.21).

Owing to consumer demand, the manufacture of the Buick 70 automobile began, starting in 1947, installing the Dynaflow hydrodynamic transmission, invented and constructed by G. W. Simpson. Starting with the next 1948 model, this transmission became a standard on all the Buick 70 cars. The Dynaflow transmission consisted of a torque converter and a mechanical planetary gearbox that provided one lower-speed gear and operation in reverse. The transmission was disconnected under ordinary driving conditions and was put into operation only under difficult road conditions and when using the engine for braking the automobile on descents to assist the brake operation. The car's motion under ordinary road conditions started with the torque converter converting the torque followed by its functioning as hydraulic clutch after the car attained the desired acceleration. This means that under ordinary conditions the transmission operated without shifting of gears, i.e., fully automatically. The Dynaflow transmission was followed by other similar devices. These were used not only in the drives of expensive passenger cars, but also in buses and in heavy-duty trucks.

FIGURE 1.21
MAZ-501, Belarus.

The construction of gas and oil mains, railroads, and highways, open-pit mining, mechanization of the timber industry, agriculture, and military needs call for the availability of high-performance all-terrain wheeled vehicles with high mobility and traction performance.

Such equipment can be designed by improving mobility of existing conventional vehicle designs as well as by special high mobility vehicles with original design solution of overall arrangement, units, and systems. High mobility vehicles are more suitable for the aforementioned tasks. This conclusion is to some extent supported by data in Table 1.1 that lists weight/payload ratios calculated from the expression

$$K_{\mathrm{wp}} = W_{\mathrm{c}}/W_{\mathrm{g}} \tag{1.1}$$

where

W_{c} is the curb weight of the vehicle
W_{g} is the payload weight

The average value of K_{wp} of high-mobility machines is about 15% lower than that of those with improved mobility. At the same time the values of K_{wp} for both of these machines is higher than that of general-purpose vehicles.

Both the improved- and high-mobility machines typically employ multiwheel and all-wheel drivelines, with the number of driving axles being two or more. The use of such systems, driven by the need to further improve the mobility and capacity, increased greatly during the 1960s, which saw the appearance of 8×8 and 12×12 vehicles that provided capacities in excess of 50 ton (Figures 1.22 and 1.23).

Thus, the Semex-Tatra 4727OL pipe carrier had the 10×8 wheel formula; 10×6, 12×6, 14×8, and 16×8 chassis were utilized for Faun truck cranes. High-mobility chassis cranes with 8×8 and 12×12 wheel formulae were designed in the U.S.S.R. The Central Research Institute of Automobiles and Automobile Engines (NAMI, U.S.S.R.) designed an articulated dumpster consisting of a three-axle articulated truck tractor and a two-axle semitrailer. The Titan company (Germany) produced specialized automobiles with wheel formulas from 6×4 to 10×8 that were used as airport and in-plant truck tractors, communal vehicles and on oil fields.

The need to improve productivity by increasing the payload capacity under the existing limitations on the loading of road surfaces has resulted in an extensive use of multiwheel drive systems in ordinary and articulated trucks (combinations of a truck tractor and trailers and semitrailers) used for intercity transportation over upgraded roads. This not only reduces (or at least prevents increasing) the load on the wheels, but ensures high tractive and velocity operational properties in particular in the case of heavy- and extra-heavy duty highway trucks.

Statistical analysis of the use of various wheel formulae on highway heavy-duty trucks showed the following (Table 1.2). Two-axle drives (4×2 and 4×4) are used in lower-power vehicles as compared with automobiles designed to the 6×4 and 6×6 formulae. Here 4×4 vehicles having a somewhat lower mean engine power are used over a larger range of power classes than those 4×2 machines, which is seen from the values of the power of the coefficient of variations, equal to the ratio of the standard deviation to the mean value of the power. The same applies to 6×4 and 6×6 vehicles.

These statistical data are validated by results of the economic studies. Thus, all-wheel drive vehicles not only have longer service lives than 4×2 vehicles, but their resale value is also much higher. The depreciation costs for 10 years of operation of a 4×4 vehicle are, as a

TABLE 1.1
Truck Mass Parameters

Truck	Full Mass, ton	Wheel Formula	K_{wp}	Year of Truck's Data
Improved off-road mobility trucks, $K_{wp}^{average} = 1.398$				
GAZ-66	5.460	4×4	1.820	1982
ZIL-43273H	8.460	4×4	1.683	2006
Mercedes-Benz 1018A	10.500	4×4	0.736	2006
Mercedes-Benz 1318A	13.500	4×4	0.561	2006
ZIL-433440	10.715	6×6	1.777	2006
ZIL-131	11.685	6×6	1.337	1982
Ural 375D	13.025	6×6	1.605	1982
Ural 4320	13.245	6×6	1.645	1982
Ural 4320-41	15.400	6×6	1.567	2006
KrAZ-255Б1	19.525	6×6	1.604	1982
KrAZ-260	22.000	6×6	1.445	1982
KrAZ-6322	23.000	6×6	1.257	2006
MAZ-6317	24.050	6×6	1.370	1990
Steyr 26M 39	26.000	6×6	1.167	1994
High off-road mobility trucks, $K_{wp}^{average} = 1.215$				
Chrysler M-410	6.980	8×8	1.083	1967
Ford M-656	11.800	8×8	1.622	1967
Bussing NAG	17.000	8×8	1.427	1967
M.A.N.	22.000	8×8	1.193	1972
KamAZ-4310 + active semitrailer	23.175	10×10	1.295	1978
M977 A2	28.123	8×8	1.673	2006
M1074	39.200	10×10	1.376	1993
MAZ-543	40.500	8×8	1.025	1995
MAZ-79091	43.500	8×8	0.813	1995
MAZ-7916	82.000	12×12	0.640	1995
Road trucks, $K_{wp}^{average} = 0.802$				
GAZ-53A	7.400	4×2	0.849	1982
Mercedes-Benz 1018	10.500	4×2	0.544	2006
ZIL-130-76	10.525	4×2	0.754	1974
International 4300	10.660	4×2	1.255	2006
Mercedes-Benz 1018	13.500	4×2	0.495	2006
Mercedes-Benz 1328	13.820	4×2	0.531	2006
MAZ-53371	16.000	4×2	0.883	1982
KrAZ-5133B2	18.000	4×2	1.046	2006
Ural 377H	14.950	6×4	0.992	1982
KamAZ-5320	15.305	6×4	0.912	1982
ZIL-133ГЯ	17.835	6×4	0.783	1982
ZIL-6309HO	18.225	6×4	0.800	2006
KrAZ-257Б1	22.600	6×4	0.883	1982
KrAZ-65101	26.000	6×4	0.675	2006
KrAZ-65053	28.000	6×4	0.626	2006

FIGURE 1.22
MAZ-7310, 8 × 8.

FIGURE 1.23
MAZ-537, 12 × 12.

TABLE 1.2
Statistical Analysis of Heavy-Duty Highway Trucks

	Year of Production		
Wheel Formula	**1982**	**1990**	**2000**
4 × 2	$\frac{147.5^{a}}{34.0}$	$\frac{167.6}{36.1}$	$\frac{186.4}{34.8}$
4 × 4	$\frac{126.7}{42.8}$	$\frac{155.3}{44.0}$	$\frac{178.6}{43.1}$
6 × 4	$\frac{191.0}{17.0}$	$\frac{237.3}{18.3}$	$\frac{254.2}{19.2}$
6 × 6	$\frac{190.1}{26.6}$	$\frac{211.8}{28.3}$	$\frac{228.7}{27.6}$

[a] Mean power in numerator, kW; coefficient of variation in denominator, %.

rule, identical to those of a 4×2 vehicle operated for 8 years. It was also assumed reasonable that the value of a 4×4 machine be 25%–30% higher than that of a 4×2 vehicle. On the basis of transported freight above 10 ton the profit of a user can, in many cases, be greater from using a 6×4 vehicle than one with a 4×4 wheel formula. Truck tractors with 6×4 or 6×6 wheel formulae are extensively used in towing semitrailers with a full mass of 38–42 ton and more. These are MAZ-6422, Mercedes-Benz 2232, Scania-142N, and others.

Starting with 1940s trisection articulated machines consisting of a truck tractor, semitrailer and trailer came into extensive use in the United States. This arrangement is at present the most prevalent there, where at times four-section vehicles consisting of a truck tractor with a semitrailer and two trailers are also used. In Europe, the standard arrangement is a truck tractor towing two trailers.

The axles of the trailers and semitrailers may be powered. Thus, for example, Multidrive Ltd. (England) has designed transmissions for articulated carriers consisting of truck tractors and semitrailers with driving axles. The wheel formula of these carriers may be 8×6, 10×6 or bigger. Figure 1.24 shows an all-terrain truck tractor with a semitrailer with engine-powered wheels. The overall wheel formula of this carrier is 10×10.

Using a similar principle, the Krane Fruehauf Company has constructed a self-dumping carrier (truck tractor and a semitrailer) with a body volume of 25 m^3. The front wheels of the 6×2 truck tractor are steerable, whereas traction is supplied by the rear axle with double wheels. The front wheels of the semitrailer are also driving wheels. At the negligible penalty of the added weight of the transmission, the 38-ton carrier has a load-carrying capacity of 22.08 ton, which has corresponding to it a value of the weight/payload ratio (K_{wp}) of 1.721.

As a result of the tendency toward constant increasing the load-carrying capacity, many companies initiated, during the 1980–1990s, the manufacture of four- and five-axle vehicles. Companies such as Daimler-Benz, M.A.N., and Volvo also produce a wide range of such machines. Their 8×4 and 8×6 paved-road vehicles are usually operated

FIGURE 1.24
KrAZ-260D, 10×10.

at full mass of 32–35 ton and at times as high as 41 ton. The 8×8 all-wheel drive vehicles are usually designed for off-road work and have a full mass of 48–65 ton.

The increase in the number of axles improves the vehicle capacity. The total cost of transporting 1 ton of cargo by a four-axle vehicle is, as a rule, 30% lower than when using a three-axle truck.

The FTF Company (the Netherlands) was one of the first to produce five-axle vehicles with the 10×4 wheel formula (three steering axles). They could be provided with truck bodies or be used as truck tractors for pulling heavy loads. Chassis with the 10×4 wheel formula (three steering axles) are also used on Mercedes-Benz 4335 K dumpsters.

The Tatra Company has complemented the range of its multiaxle vehicles by the 10×8 five-axle vehicle. At full mass of 53 ton, the two leading axles carry 9 ton each, whereas the three rear axles—12, 12, and 11 ton.

Articulated carriers with truck tractors employing a large number of axles and with trailers that also employ a large number of axles have come into extensive use. In Sweden, articulated carriers with a full mass of 52 ton employ trailers with four axles. Articulated trucks in the Netherlands with a full mass of 50 ton employ 8×2 M.A.N. truck tractors.

A similar gradual increase in the use of multiaxle machines was and is observed in the manufacture of agricultural tractors. This was aided by the continuous rise in engine power and in the energy density of tractors. For example, the mean power of farm tractors produced in the United States increased approximately 3.5-fold during the last 35 years of the twentieth century. The power rating of class 1.4 Belarus tractors increased from 27.2 kW of the MT3-2 (in the 1950s) to 78.3 kW of the MT3-1025 as of today.

The first step in more completely utilizing the engine power was to increase the work velocities. Extensive experience in operating farm tractors with such energy density has validated this approach. At the same time, analysis shows that, as the engine power is increased, the rates at which the tractor velocities increase still decreases. Statistical studies of parameters of wheeled tractors with an energy density of about 20 kW/ton show that the dependence of the average design velocities V_{mid}, m/s, on the tractor mass m_{t}, ton, has the form

$$V_{\text{mid}} = 3.032 m_{\text{t}}^{0.321} \tag{1.2}$$

where $V_{\text{mid}} = 0.5(V_{\text{min}} + V_{\text{max}})$.

The rate of rise in V_{mid} decreases with increasing weight and power of the tractors and, starting with approximately $P_{\text{e}}^{\text{max}} = 120$ kW and $m_{\text{t}} = 6$ ton, the average velocity remains virtually the same. This behavior of V_{mid} is attributable in the first place to the conditions under which agricultural operations are carried out, the increase in the resistance of soil-working implements with increasing vehicle velocity, etc.

The limitations on the velocity increase has shifted the efforts of utilizing the high engine power to increasing the width of worked soil and combining agricultural operations that allow to significantly increase the tractor's drawbar pull. At the same time, increasing the traction capacity increases the use of the adhesive weight (the weight taken up by the driving wheels) and, as a result, increases the power lost for slippage.

In addition, the use of heavy, wide span implements and combined agricultural operations significantly increases the normal loads on tractor wheels. The latter causes compaction and damage to the soil structure, reducing the yield. The yield of barley in track passages of MTZ-80 and K-700 tractors decreased by 12%–14% and that of potatoes—by 27%. In view of this, when outfitting tractors with their associated agricultural machinery it becomes necessary to reduce the worked span, which decreases the traction and under-utilizes the engine power.

Investigations and experience shows that the most cardinal way of increasing the tractor traction performance and utilizing the engine power is using all-wheel driveline systems.

The high traction performance of 4×4 tractors allows increasing the time available for agricultural work and enhancing the tractor's utilization on a year-round basis. In addition to improved traction, 4×4 tractors have velocities higher than their 4×2 counterparts do. Analysis shows that the average velocities of 4×4 tractors at nominal drawbar pulls are 10%–14.5% higher than the average velocities of 4×2 tractors.

A statistical analysis of more than 4000 agricultural tractors was undertaken. Comparison of power-distribution densities shows that a 4×2 tractor's power P_e^{max} the coefficient of variations reduced from 39.7% in 1970 to 24.8% in 1995, while 4×4 enjoyed growth of this parameter from 33.6% up to 37.8%. The coefficient of variation changes of that kind attest to a still bigger advantage of all-wheel drive tractors and a decrease of power P_e^{max} range in which 4×2 tractors are used. Studies of power distribution densities $f\left(P_e^{max}\right)$ of 4×2 and 4×4 tractors having identical and nonidentical wheels produced in the years to follow brought forth approximately similar results (Table 1.3).

This manner of changes in the coefficient of variations points to ever increasing use of all-wheel drive tractors and the reduction in the range of P_e^{max} within which 4×2 tractors are used. The higher capacity of 4×4 tractors as compared with 4×2 ones is responsible for the fact that all the currently manufactured and under-design wheeled tractors, including those with moderate capacities either have all-driving wheels or employ all-wheel drive modifications.

Positive locking units secure the best indicators of a tractor's tractive properties in the field. From the design point of view, this is done as follows. Interaxle drives employ positive locking axle engagement with the possibility of disengaging one of them or have interaxle differentials with a backup positive locking system installed in them. Locking and self-locking differentials are widely used in tractors' driving axles.

Two-axle drives have gradually penetrated also into passenger car designs. At the end of the 1950s, an original four-wheel driveline design was developed at the Ferguson Company. It included an interaxle differential with overrunning clutches (see Figure 1.25).

TABLE 1.3
Statistical Analysis of Agricultural Tractors

	Agricultural Tractors		
Year of Production	**Agricultural Tractors 4×2**	**4×4 with Different Front and Rear Wheels**	**4×4 with the Same Front and Rear Wheels**
1982	$\frac{48.1^{a}}{33.1}$	$\frac{60.1}{45.3}$	$\frac{98.2}{48.8}$
1984	$\frac{46.0}{28.4}$	$\frac{60.1}{46.3}$	$\frac{93.0}{53.9}$
1986	$\frac{47.7}{29.9}$	$\frac{63.9}{42.0}$	$\frac{116.3}{40.1}$
1989	$\frac{51.2}{28.1}$	$\frac{64.8}{41.8}$	$\frac{98.3}{56.3}$
1995	$\frac{53.9}{29.2}$	$\frac{67.4}{45.2}$	$\frac{112.8}{48.8}$

[a] Mean power in numerator, kW; coefficient of variation in denominator, %.

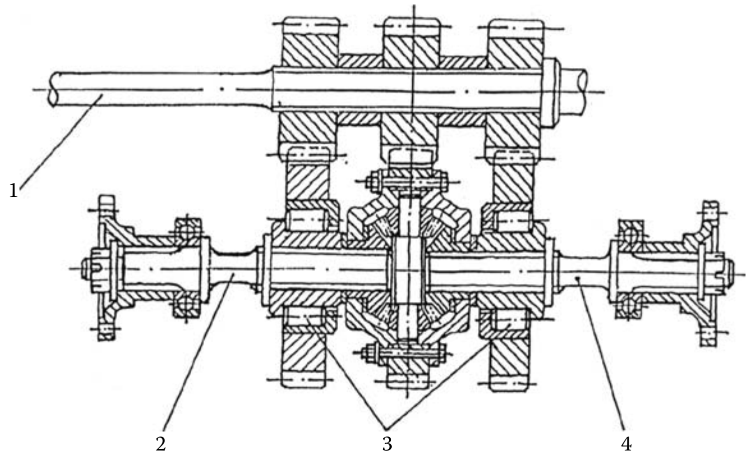


FIGURE 1.25
Concept of Ferguson's center differential and free-running clutches: 1, input shaft of transfer case; 2, output to front axle; 3, roller-type overrunning clutches; 4, output to rear axle.

Power to the interaxle differential in this drive was fed by the input shaft 1 of the gearbox through overrunning clutches 3 and transmitted it, dividing it into halves, through shaft 2 to the leading axle and through shaft 4 to the rear axle. The overrunning clutches allowed the leading axle wheels to turn freely when the vehicle took a turn. In addition, the clutches automatically locked the differential when the wheels of one of the axles slipped. Then one clutch locked the interaxle differential upon slipping of the front-axle wheels and the second—upon slipping of the rear-axle wheels.

Ferguson's design was developed further in the Jensen FF car, in which Jensen, an engineer with the Harry Ferguson Research Ltd., used a nonsymmetrical interaxle differential of the epicyclical type. He supplied 63% of the torque to the rear axle and 37%—to the front axle. This improved the road stability of the vehicle. Secondly, the overrunning blocking clutches were replaced by a multidisk clutch that operated on an organosilicon fluid. This was one of the first cases of utilizing a viscous clutch as a locking mechanism. This clutch perceptibly simplified the design of Jensen's driveline system. Finally, Jensen replaced the mechanical locking device employed in the Ferguson driveline by an electronic locking device.

The Ferguson and Jensen driveline designs started seriously attracting the attention of automobile manufacturers during the 1980s. As an example of a similar drive produced at that time, the ''Zahnradfabrik'' can be referred here (Figure 1.26).

The autumn of 1964 saw the start of the series production of the Porsche 911 all-wheel drive passenger car, designed by the son of the famous Porsche, whose first name was also Ferdinand and was known as Ferry Porsche. This machine was regarded as the best automobile of 1960–1969. The Porsche 911 had an air-cooled rear-mounted six-cylinder in-line engine rated at 130 HP. Its primary driving axle was in the rear. Power to the front steering axle was supplied with the same arrangement used previously on the Kubelwagen designed by Porsche Senior, with the only difference that the driveline of the Kubelwagen employed a dog clutch, whereas the Porsche 911 used a hydraulically controlled multidisk friction clutch. The engine, the gearbox, the rear-axle differential, the interaxle multidisk friction clutch together with the drive shaft for the front driving axle of this vehicle comprised a single compact subassembly. A permanent interaxle

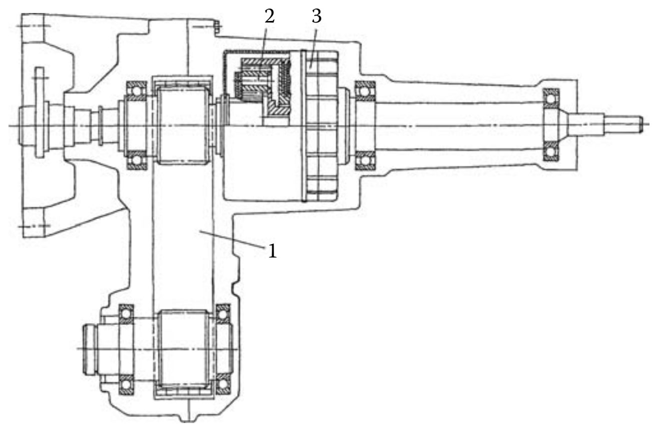


FIGURE 1.26
Kinematic diagram of a ZF A-95 transfer case: 1, front-axle chain drive; 2, planetary type central differential; 3, viscous clutch.

differential driveline was also employed by Porsche. Later, in 2005, there appeared the new, 977 version of the Porsche 911 with a multidisk viscous coupling which transfers from 5% to 40% of the tractive force to the front wheels.

Toward the end of the 1970s, the Matra Company that manufactured Formula 1 sports cars, attempted to design a mixed driveline for its all-wheel drive automobile. In it, the rear wheels were to be driven mechanically, whereas the front wheels, by means of a hydrostatic drive (see Figure 1.27).

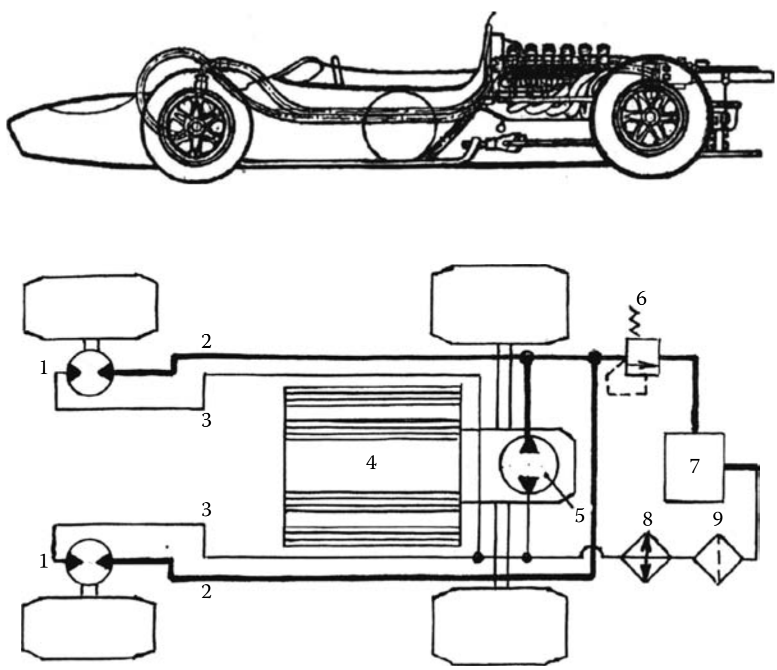


FIGURE 1.27
Tentative design of hydrostatic drive of the forward axle of a Matra sports car: 1, hydraulic motors; 2, hydraulic motor fluid supply lines; 3, hydraulic motor fluid drain lines; 4, internal combustion engine; 5, fluid pump; 6, pressure (safety) valve; 7, oil tank; 8, radiator; 9, filter.

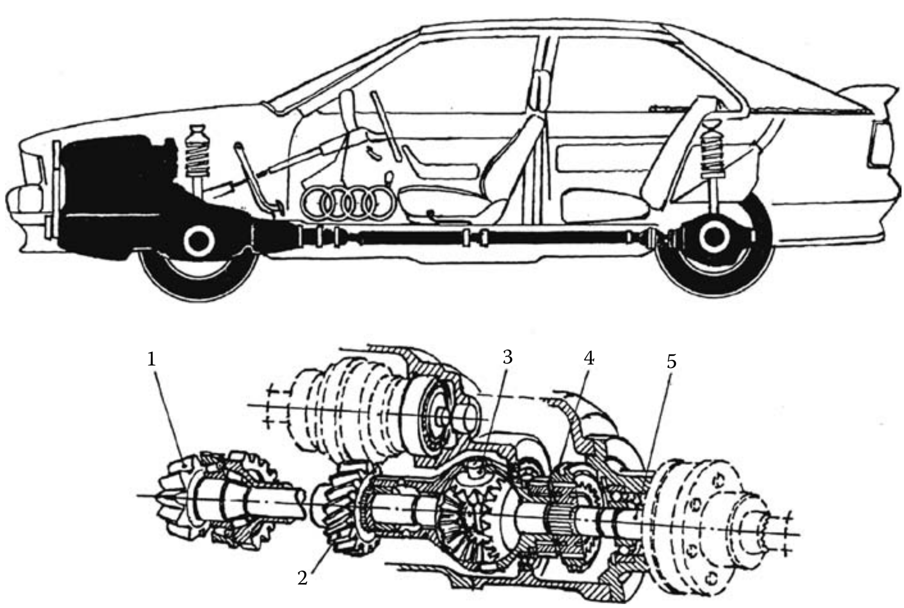


FIGURE 1.28
Schematic of permanent drive of Audi-200 Turbo Quattro: 1, pinion of final drive of the front axle; 2, gear pair connecting the case of the central differential with the output shaft of the gearbox; 3, central differential; 4, locking clutch; 5, differential output shaft—an element of the drive to the rear driving axle.

In 1980, the manufacture of the Audi-200 Turbo Quattro was started. The vehicle combined the response of a sports automobile and a high comfort and safety level. It owed this to turbocharging and other modern achievements of this time in vehicle manufacture that were embodied in it. The Audi-200 additionally boasted a ''pure,'' highly lightened drive of the wheel pairs (see Figure 1.28).

The interaxle (central) differential together with the front-axle differential were built into the Quattro's transmission. As a result, the driveline was ''pure'': there was no transfer case.

Irrespective of the merits of the first all-wheel drive passenger cars, there is no doubt that the Audi-200 was precisely the vehicle from which the boom in manufacturing similar cars by other makers started. Somewhat later, there appeared the VW Passat Variant Syncro with the same driveline arrangement. The all-wheel drive concept that was developed for the Lancia Delta Turbo 4×4 became an alternative to the Audi. In this vehicle, the engine and the transmission are located transversely, which made it possible to place the planetary interaxle differential in the housing of the final drive of the transversely placed power unit. The interaxle differential divides the power between the leading and trailing axles in a 58/42 proportion and the cardan shaft transmits rotation to the rear wheels. This arrangement also dispenses with the transfer case. The 2 VW Type was one of the first trucks with a ''pure'' drive in which the front steered and driving axle was coupled automatically to the engine by a viscous clutch. It should be noted that heavy-duty all-wheel drive trucks also entered the realm of the permanent drive. The operation of drive axles in these vehicles is often controlled by specially designed electronic systems. According to some estimates, the ratio between cars with 4×2 and 4×4 wheel formulae will soon be one to one.

1.1.9 Power Dividing Units

The use of all-wheel drives does not fully exhaust the potential for improving traction-velocity properties even though it is assumed that the coefficient of utilization of the traction weight of all-wheel drive vehicles is equal to unity. That is the entire weight participates in generating the traction power, as compared with ordinary-driveline (or nonall-wheel drive) vehicles, in which a part of the weight devolves upon the driven wheels

$$K_{\mathrm{w}} = W_{\mathrm{dr}}/W_{\mathrm{a}} \tag{1.3}$$

where
W_{a} is the gross weight of the vehicle
W_{dr} is the adhesion weight, i.e., the weight devolving upon the driving wheels

Note that for ordinary-drive vehicles, $K_{\mathrm{w}} < 1$.

Nevertheless, the possibility still exists that even the adhesion weight of all-wheel drive vehicles may be underutilized. This happens, for example, in the case of open differentials and when the wheels of the vehicles move under different road conditions, when the driving wheels with better conditions are not able to develop the required traction. For this reason, together with the active development of all-wheel and multiwheel designs, a continuous search is under way for designs of power dividing units (PDUs) that would be capable of dividing the power among the driving wheels in accordance with the tire surface grip.

A multidisk clutch that employed an organosilicon fluid—polymethylsiloxane (abbreviated to siloxane)—was patented in the United States in 1917 for use as a locking mechanism. As opposed to ordinary incompressible fluids, siloxane—the working fluid of the clutch—is a compressible (non-Newtonian) fluid. When this polymer is stirred, its spirally shaped macromolecules increase in volume with attendant increase in viscosity as a function of the shear strain (gradient of velocity between the layers). Figure 1.29a shows

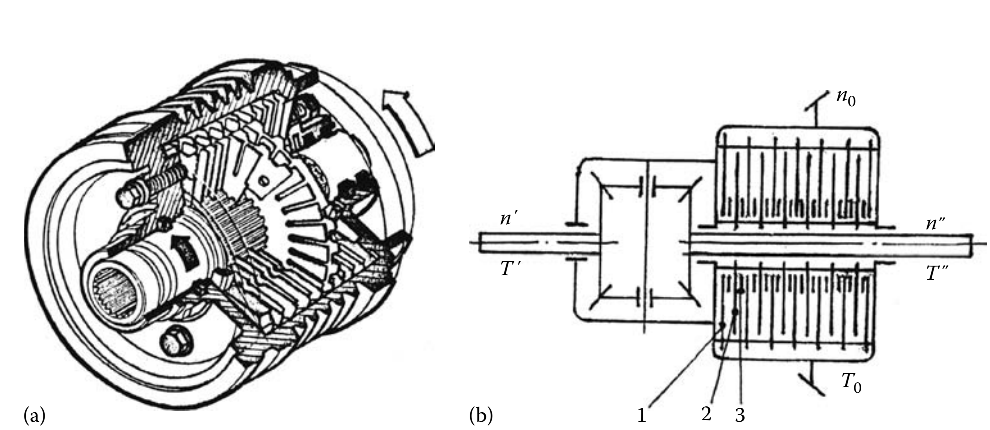


FIGURE 1.29
Viscous clutch and differential schematics: (a) viscous clutch; (b) differential with a viscous clutch, equipped with disks 1, coupled to the case; disks 2, coupled to the output shaft; 3, spacer rings.

an example of such a viscous clutch. It consists of a housing and shaft with which alternating driving and driven disks, made of soft 0.25–1.0 mm thick sheet steel are coupled by means of splines. A 0.1–0.2 mm clearance is left between the disks. In certain viscous clutches, spacing rings are used for maintaining the spacing between the disks constant. When no such rings are installed, the disks are polished and a 5–50 μm thick antiscuff coating is applied to their surfaces. The disks of the viscous clutch are perforated by slots and holes of different configurations to enhance their mechanical effect on the working fluid. The configuration (shape) of the slots and holes is selected experimentally. The leak-proof housing of the viscous clutch is filled to 90%–93% of its volume. The fluid-free parts of the volume serve as a ''safety chamber'' that does not allow the expanding siloxane to exert a destructive force on the viscous clutch.

The first viscous clutch was designed during the 1960s by T. Rolton and D. Gordner, who were on the staff of the Formula Ferguson Research Company. The design of the viscous clutch was similar to that described above. However, no such clutch was put into production during the year when it was designed since the Formula Ferguson Research Company closed down and the patent was transferred to the GKN Company. This company, together with the Zahndradfabrik corporation, formed the Viscodrive company in Germany that started producing viscous clutches for BMW cars (Germany) and for Ford automobiles. Viscodrive opened a branch in Japan that started producing viscous clutches for Toyota and Nissan automobiles. Honda obtained a license from GKN and started producing viscous clutches for its own cars. In 1979, mass production of viscous clutches, also under GKN license, was undertaken for Eagle car models of the American Motors by New Process Company, a Division of Chrysler. Viscous clutches were also produced by the Steyr-Daimler-Puch Company (Austria) for the Caravelle-Syncro Volkswagen minivan. The viscous clutch is used for automatically coupling one of the axles of the vehicle and also as a locking mechanism of differentials (Figure 1.29b).

The MacPower Divider self-locking differential was invented in the United States in 1929. It was used on 4×4 trucks as an interaxle PDU and from 1948, it also came into use as an interwheel differential for driving axles. In both cases, it improved mobility and traction performance of the vehicles.

Mule and Scarlock invented in 1932 a self-locking worm-gear differential. Initially nothing was known about its locking performance and no use was found for it. Then, after its properties were learned, it gradually came into use as an interwheel differential on heavy-duty trucks and on truck tractors. It came into wide use in the 1950s. It exhibited good performance in drives of tri-axle very heavy-duty trucks and special tractors used in quarries and in snow removing and similar machines. A modification of the worm-gear differential that lacked additional satellite gears with radial axes was extensively used in drives of vehicles produced by the Walter Company.

A double-acting overrunning clutch under the name of Tronton-Tandem was designed in the United States in 1937. This was a differential of a kind that provided for automatic uncoupling of the outer wheel from the drive at the time when the machine took a turn. Because of this property, it was used by different companies on 22 models of trucks rated at from 0.25 to 4 ton. The Tronton-Tandem was the second double-acting free-running differential. The first such was the Multi-Pull differential (Figure 1.30a).

Unlike the Tronton-Tandem, the Multi-Pull had a number of shortcomings and did not come into such extensive use as the former. A further development of the double-acting free-running clutch is exemplified by the NoSPIN differential designed by the engineering staff of the Detroit Automotive Products Corporation. This reliable mechanism is in

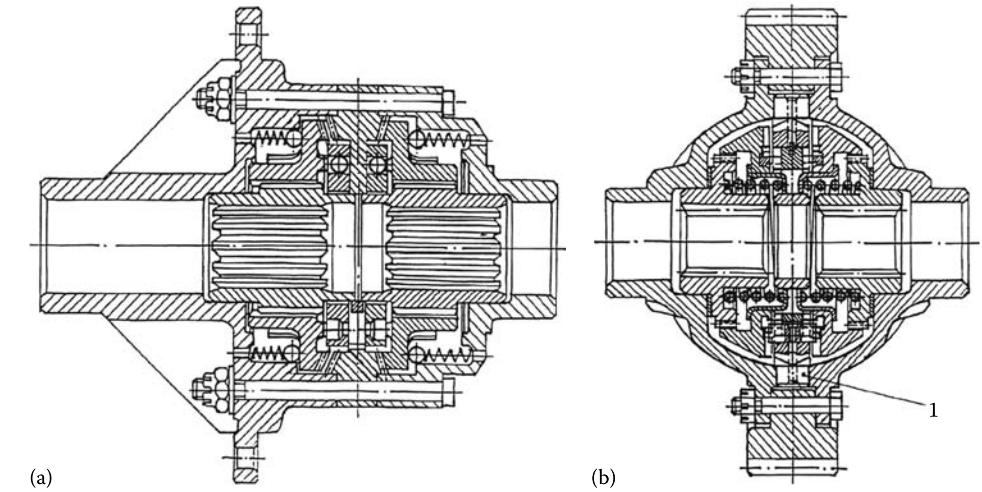


FIGURE 1.30
Overrunning self-locking differentials: (a) "Multi-Pull"; (b) MAZ, Belarus.

production even currently. When it first appeared, the Ford Motor Company engineers used it as a basis for designing nine NoSPIN models for different vehicles from trucks to passenger cars. Differentials that operated on the NoSPIN principle were developed under the leadership of Dr. A. Kh. Lefarov (Figure 1.30b) and have been used in multiaxle vehicles of MAZ company (Figures 1.22 and 1.23).

The Thornton Power Lock self-locking differential that utilized friction disks (Figure 1.31a) was designed in the United States in 1956. Its locking properties were first tested out

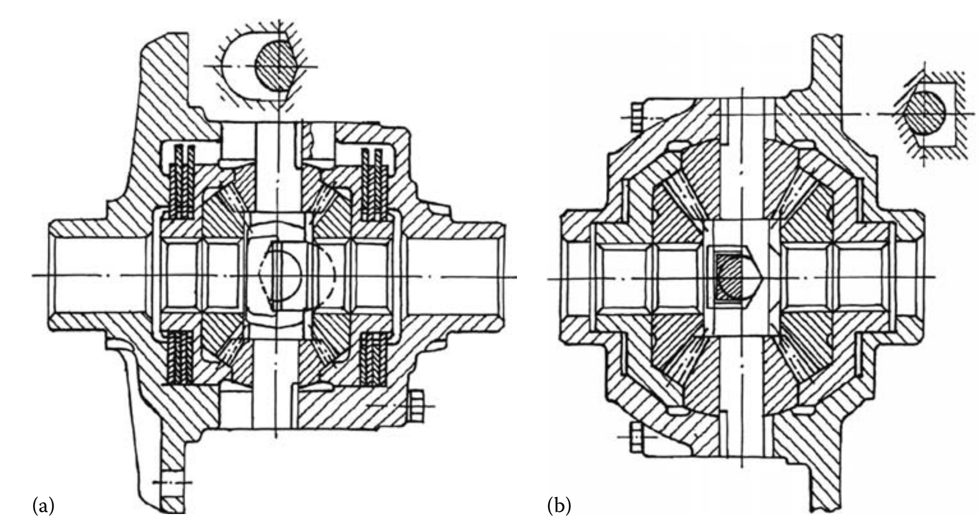


FIGURE 1.31
Thornton Power Lock self-locking differential: (a) with disk-type friction clutches (1956); (b) with cone-shaped friction clutches.

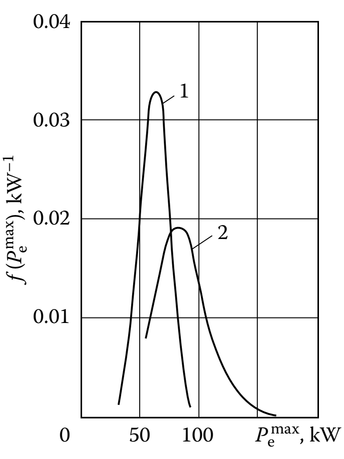
on certain models of Packard passenger cars and then, on popular demand, it was also installed in Studebacker cars. As of the middle of 1958, the Thornton Power Lock Company produced and installed about 200,000 differentials with friction disk clutches.

The Thornton Power Lock differential had, instead of the spider two pins located one across the other at a right angle with a pair of pinions on each of the pins. One pin presses by its ends, beveled at a certain angle, on the beveled slots in the apertures of the plain half of the differential case, whereas the other pin, with its ends, beveled at a certain angle, presses on the beveled slots in the apertures of the flange half the differential case. This allows the pins with their pinions to move each in its direction, relative to the principal axis (axis of rotation) of the differential and press either the friction cones (Figure 1.31b) or, by means of the pressure rings, the disk stacks (see Figure 1.31a) to the right and left halves of the differential housing. This will be accompanied by an increase in the friction torque inside the differential and this will cause the half axles to lock and the torque to be redistributed between the driving wheels of the axle. In addition, this provided for transmitting a higher torque to the wheel having the better adhesion with the supporting surface.

Self-locking differentials with a large variety of features came into extensive use. These differentials came into wide use in passenger cars, by customer order, starting with the middle of the 1980s. Analysis shows that these mechanisms were installed on vehicles with a wide range of power ratings: from 50 to 150 kW (Figure 1.32). Such a tendency is also retained in modern designs.

With increasing engine power and traction loads on axles, differential mechanisms with different locking methods came into increasing use in driving-axle reducers of highway truck tractors (particularly of heavy-duty highway trucks). Table 1.4 lists the percentage breakdown of various driving axle differentials installed on 1990 model year trucks.

Approximately the same tendency prevails until now. Approximately 60% of trucks use locking driving-axle differentials, 14% self-locking differentials, and the remaining 26% open differentials. These differentials are distributed among trucks of different power classes and different wheel formulae in the following manner. One half of trucks with a full mass of 2.5 ton and above with 4×2 and 4×4 wheel formulae have locking differentials on their rear axles, and approximately 17% of these trucks have self-locking differentials. The remaining 33% (usually these are light- and medium-duty trucks) are



N	Parameters
1	$\frac{68.0^*}{22.7}$
2	$\frac{89.9}{22.3}$

FIGURE 1.32
Power distribution density (P_e^{max}) of 4×2 passenger car engines: (1) with open differentials; (2) with self-locking differentials on customer order; *the numerator gives the mean power, kW; and the denominator—the coefficient of variation, %.

TABLE 1.4

Power-Dividing Units on 1990 Trucks—Statistical Data

	Percentage of PDUs on Trucks with Full Mass of				
PDU in Driving Axles	**Less than 2.5 ton**	**2.5–4 ton**	**4.5–9 ton**	**9–12 ton**	**12 and More ton**
Open differentials	65%	65%	33%	—	—
Locking differentials	—	—	34%	100%	100%
Limited-slip differentials	35%	35%	33%	—	—

equipped with open differentials. About 90% of extra-heavy duty trucks with wheel formulae of 6×2, 6×4, 6×6, 8×6, and 8×8 use locking interwheel differentials. The remaining 10% use self-locking units.

Statistical analysis shows that the selection of a given PDU for use in agricultural tractors depends on the power, weight and geometries of the tractors as well as on general engineering considerations. In the overwhelming majority of cases, tractors with identical front and rear wheels use interaxle locking differentials. Tractors with smaller front-wheels have front axles with either free- or self-locking differentials. As a rule, locking or in some cases self-locking differentials are used in rear-driving axles of such tractors. Table 1.5 presents statistics on the power distribution densities of 4×4 tractors employing different types of front-axle differentials.

It is seen that the range of applicability of limited-slip differentials on tractors of different power classes is much broader than the range of applicability of free differentials. Locking differentials are used on tractors of the same power classes as limited-slip differentials. The locking action of the differentials is usually attained by hydraulic and much less frequently—by pneumatic control.

An overall assessment of the development of designs of components and systems for distributing power among axles and their wheels, shows that this development proceeded from the simple to the complex. It started in 1930–1940s with the development of mechanical self-locking differentials of the worm-gear type, differentials with friction clutches, speed-sensitive and torque-sensitive differentials, and continued during the 1970s and

TABLE 1.5

Statistical Analysis of Agricultural Tractors

	Differentials		
Year of Production	**Free**	**Self-Locking**	**Locking**
1982	$\frac{49.8}{36.8}$	$\frac{76.1}{48.2}$	—
1986	$\frac{44.5}{22.3}$	$\frac{68.4}{42.5}$	$\frac{71.8}{42.5}$
1989	$\frac{48.7}{31.9}$	$\frac{70.6}{44.9}$	$\frac{68.3}{43.1}$
1995	$\frac{49.4}{32.7}$	$\frac{72.4}{45.3}$	$\frac{73.6}{44.2}$

Note: Mean power in numerator, kW; coefficient of variation in denominator, %.

1980s with design of electromechanical locking and electronically controlled units. Special mention is deserved by the Auto-Lock system for locking the interaxle differential of the 6×4 truck (Rockwell International, New York), the 4MATIC electronic system for the Mercedes (Daimler-Benz, Stuttgart, Germany) passenger car, the locking device for a differential that is actuated by the difference in wheel speeds of agricultural tractors (SIGE) and many others.

Today, some automotive companies are introducing ''torque-bias coupling,'' ''torque-vectoring,'' and ''torque management'' devices to control power delivery to the front and rear axles and the left and right wheels (see Chapter 7). These driveline systems are actually mechatronic systems, and the introduction of such mechatronic designs is an epoch-making step in driveline system design. The mechatronic driveline systems are more ''flexible'' and proactive in distributing the engine power to the driving wheels than automated mechanical systems, e.g., the different limited-slip differentials, mechanically/electronically lockable differentials, viscous clutches, NoSPINs, on-demand systems and many others.

At the very beginning of the development of driveline systems and PDUs, the sole purpose was improving the mobility of the vehicles. Today an increasing number of OEMs and suppliers recognize the fact that vehicle operational properties such as tractive and velocity properties, stability of motion, turnability, handling, braking properties highly depend on characteristics of driveline systems that distribute power to the driving wheels. It is important to emphasize that the fuel economy and safety of vehicles that are vehicle consumer properties also depend to a large extent on driveline system designs. For this reason, driveline systems for distributing the power to the driving wheels should be designed with consideration of their combined effect on many of the vehicle operational and consumer properties.

This book is concerned with problems of driveline design precisely from the point of view of their combined effect on the performance and dynamics of wheeled vehicles.

1.2 Classification of Driveline Systems and Power Dividing Units

Depending on the manner of power transmission, all drives can be classified as electrical, hydraulic, mechanical, and hybrids. This section examines in detail mechanical driveline systems, including those of vehicles with different steering systems. Electronically controlled mechanical driveline systems are defined as mechatronic driveline systems. When speaking about mechanical and mechatronic driveline systems, many publications as a rule refer solely to the wheel formula, and at times adding information on types of the differentials. These data are clearly insufficient for complete description of driveline systems and for understanding their integration within the overall design of the vehicle. After all, frequently even vehicles of the same type use different mechanisms: symmetrical and asymmetrical differentials, positive locking engagement, self-locking differentials, limited slip differentials, free running clutches, viscous clutches, etc. This is particularly intrinsic to off-road vehicles. Thus, the driveline system the International Harvester XM-409 was fully differential with redundant locking. The 8×8 M.A.N. and Tatra-813 vehicles have the same driveline systems. In the latter, the distance between the axles of the front bogie (or tandem) is somewhat larger than that between the axles of the rear tandem. Many vehicles use mixed systems with different coupling-unit mechanisms. For example, the central differential of the MAZ-79091 8×8 truck is a symmetrical, bevel-gear differential with redundant locking, the interaxle drive of the rear tandem is constantly locked and the interaxle drive of the front tandem and also the interwheel drives of the front tandem axles employ symmetrical open differentials. NoSPIN type differentials are used in the interwheel drives of the axles of the rear tandem. The wheel drives of the XM-453E1 truck with an 8×8 wheel formula and with tandem arrangement of the axles employ limited slip Power-Lock differentials.

The above example demonstrates the need for a tool that would provide a detailed description of the various existing driveline systems as a part of the overall design of the vehicles, from which it will be easier to understand the reasons for using the particular driveline system design. Design engineers need such a tool for describing and representing the driveline system that they design. The SAE J1952 Standard All-Wheel Driveline Systems Classification contains definitions to be used to outline the basic nomenclature and to classify all-wheel drive concepts. Some information is also contained in the *Terrain-Vehicle Systems Standards*, worked out by the International Society for Terrain-Vehicle Systems (ISTVS).

Historically, the terms and definitions pertaining to the classification of drive schemes were developed in the course of new driveline system design. The definitions of different types of drives were worked out primarily with consideration of marketing needs rather than from the point of view of the technical substance of the systems. For example, the term permanently engaged rear-wheel drive with the on-demand engagement of the front wheels by means of the transfer case is applied to 4WD. The same term, 4WD is applied to vehicles in which the two axles are coupled by an interaxle differential in the transfer case. This means that the same term is applied to vehicles, the driveline systems of which have an entirely different effect on vehicle performance. At the same time, the term AWD (all-wheel drive) is also applied to vehicles in which the interaxle differential is combined with the interwheel differential of the front axle. Following this logic, the same vehicle could be labeled as 4WD or AWD depending on the location of the interaxle differential—in the transfer case or together with the front-axle interwheel differential. However, the location of the interaxle differential does not affect the entire vehicle's performance.

In this book, driveline systems with two, four and more driving wheels are classified from the point of view of their effect on vehicle performance. The specifics of a given arrangement are reflected by means of additional indicators that integrate the driveline system with the overall vehicle design. The effect of the driveline system on vehicle performance depends on the characteristics of the locking properties of mechanisms and subsystems of the driveline system, i.e., their ability to distribute the power among the driving wheels. The overall vehicle design layouts first introduced by P. V. Aksenov include the type and location of the driving axles within the overall arrangement of all of the vehicle's axles, the steering system, type of suspension, and dimensions of the tires of the different axles of the vehicle. Consider details of the developed classifications.

The wheel formula of all types of drivelines is designated by a single expression $2m \times 2n$, in which n is the number of the driving axles to which the engine power is transmitted, whereas m is the total number of driving and driven (not coupled to the driveline system) axles. If $m = n$, then the vehicle is called an all-wheel drive vehicle. For example, 4×4, 6×6, 8×8, 10×10, 12×12, and 16×16 are all-wheel drive vehicles. Vehicles with different numbers of driving and driven axles, for which $m \neq n$, are known as multiwheel drive vehicles. Thus, for example, vehicles with wheel formulas 6×4, 8×4, 8×6, and 14×6 are multiwheel drive vehicles. The term nonall-wheel drive is applied to vehicles with two driving wheels. These include FWD and RWD vehicles with 4×2 wheel formula; vehicles with wheel formulas of 8×2 and 6×2 are also nonall-wheel drive vehicles.

The relationship between the driveline system and the overall vehicle design is expressed by means of the following indicators.

The drive formula gives the location of the driving axles within the overall design of the vehicle. For example, the designation 0034 corresponds to an 8×4 vehicle with the third and forth axle powered. A 12×12 vehicle has a drive formula written as 123456, whereas a 6×2 vehicle with a rear-wheel drive is designated as 003.

The axle formula symbolizes the number of axles situated in a row. For example, a vehicle with four axles (let us say, 8×8, 8×6, 8×4) that form two tandems have an axle formula of 2-2, whereas vehicles with four axles in which the second and the third axles are located close to one another have the axle formula 1-2-1. If a vehicle has wheels of different dimensions on the different axles, then the height of numbers in the axle formula should be different. For example, an agricultural tractor with two axles and smaller tires on the front wheels has the axle formula 1-$\mathbf{1}$.

The steering formula gives the ordinal number of the steered axle. For example, the steering formula for a vehicle with three axles in which the second and third axles form a tandem, whereas the front axle is steered, is 1-00. A vehicle with two axles, of which the front axle is steered, has a steering formula of 1-0. For articulated vehicles, additional designations are provided in the steering formula:

v, for an articulated trailer without control
$\frac{v}{c}$, for an articulated trailer with control
s, for a fifth wheel (a tractor-semitrailer combination)

For example, the steering formula $0\frac{v}{c}0$ corresponds to an articulated vehicle with two nonsteered axles and that is able to take turns by means of a controlled relative rotation of two segments of the vehicle about a vertical hinge. Steering systems are depicted graphically in Table 1.6.

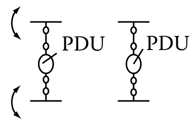
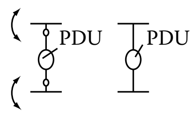
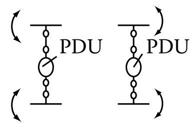
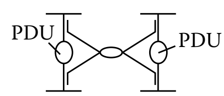
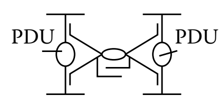
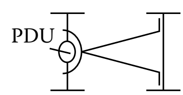
In this book, the systems that distribute the power to the wheels are defined by the concept of PDUs. Within the context of their effect on the vehicle performance, PDUs are classified by the attributes of their locking properties, i.e., on the basis of their effect on the distribution of power between the output shafts. The examples shown in Figure 1.33 illustrate types of PDUs, one of which requires additional explanation. These are planetary gear sets with two or more than two degrees of freedom and locking coupling. This is a new recently emergent type of PDUs that is used in sophisticated driveline systems. The designs of such PDUs include open differentials with additional planetary rows and locking couplings. The couplings consist of controllable friction, magneto-rheological or other clutches. This is concerned to one or another measure with virtually all types of PDUs shown in Figure 1.33.

To graphically represent the various PDUs, Table 1.7 presents designations for the most typical designs. Here there is no need to use different symbols for PDUs of the same type. For example, the same symbol $\otimes$ is used for all the limited slip differentials some of which are shown in Figure 1.33 for illustration purposes. Instead of using a large number of symbols, it is proper to write out which limited-slip differentials are used on the given vehicle. This approach is taken because a large number of designs with the most different features have already been produced and are under development.

In addition to PDUs, graphic representations of driveline units are described by Power Transfer Units (PTUs, see SAE J1952 Standard). However, in the present book the

TABLE 1.6

Steering Systems

Type of System	Designation	Steering Formula
Drive-steer and nonsteer articulated axles with independent suspension		1-0
Drive-steer and nonsteer conventional axles with dependent suspension		1-0
Drive-steer articulated axles with independent suspension		1-1
Lag hinge without steering: v		$0\overset{v}{-}0$
Lag hinge with steering: $\frac{v}{c}$		$0\overset{v}{\underset{c}{-}}0$
Fifth wheel hinge without steering: s *Notice*: Second axle is driven (nondriving)		$0\overset{s}{-}0$

definition of this term has been extended: PTUs are used for controlling power flow between their input and output elements. Typical graphical symbols of PTUs are presented in Table 1.8.

Using the PDUs and PTUs from Tables 1.7 and 1.8 it is possible to compose different driveline systems that shall be termed simple, combined, and integrated. Definitions of these systems are given in Figure 1.34. A simple driveline system may be exemplified by the drive of a 4×4 automobile with open differentials in the driving axles and an open differential in the transfer case. This means that in a simple driveline system identical mechanisms are used in all the PDUs. Should at least one of the differentials of the above vehicle be a limited-slip differential, then such a driveline system will be known as a combined system. If the vehicles under study are equipped, for example, by a traction control system, then their systems will be known as integrated, since the driveline system is functionally coupled to the braking system.

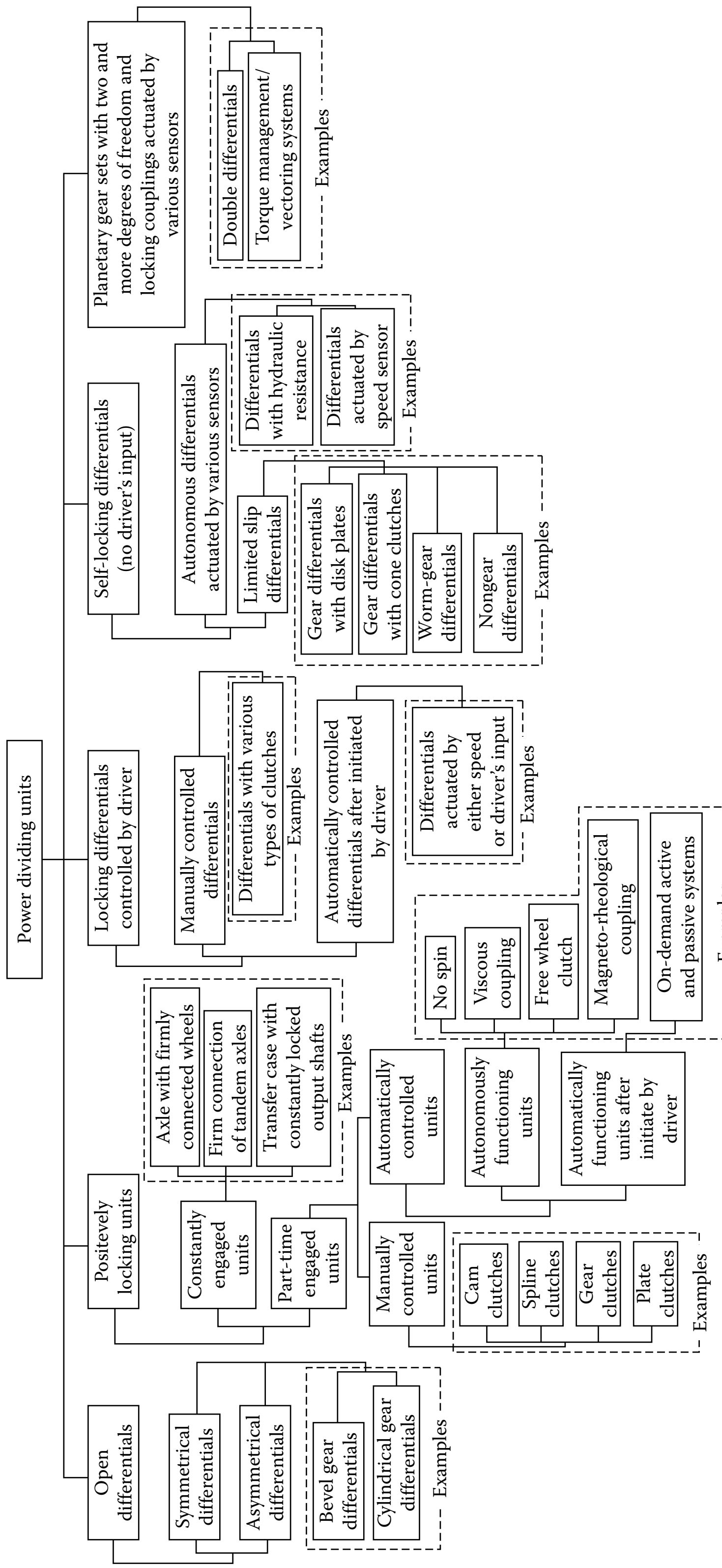
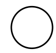
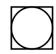
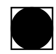
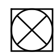
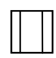
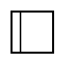
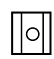
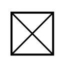
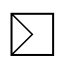


FIGURE 1.33
Major types of PDUs.

TABLE 1.7

Symbols of Mechanisms of Power Dividing Units

Mechanism in PDU	Designation
Symmetrical open (free) differential	
Symmetrical locking differential	
Asymmetrical open (free) differential	
Asymmetrical locking differential	
Symmetrical limited slip differential	
Asymmetrical limited slip differential	
Symmetrical differential with viscous or rheological clutch	
Asymmetrical differential with viscous or rheological clutch	
Overrunning self-locking differential (similar to NoSPIN)	
Automatic engagement/disengagement of one of the output shafts (e.g., on-demand systems)	
Freewheel (overrunning) clutch	
Constantly locking engagement of the output shafts	
Nonconstant engagement with manual disengagement of one of the output shafts	
Torque vectoring (torque management) device based on planetary gear sets with two and more degrees of freedom and locking couplings/mechanisms	$\boxed{\text{T}}$

Notes:

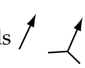
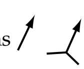
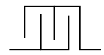
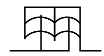
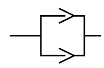
1. Intelligence symbols  (SAE J1952 Standard) may be added to the basic symbol to indicate that the PDU responds automatically to signals from one or more external control systems.
2. Additional abbreviations explains  symbols. Examples: AL/AS, antilock and antispin brake; IA/IW, interaction between interaxle and interwheel PDUs. The following abbreviations describe an interaction between driveline system (DL) and other vehicle systems and sensors: DL/DT, drivetrain (engine and transmission) system; DL/ST, steering system; DL/SS, suspension system; DL/BR, brake system; DL/LoA, longitudinal acceleration sensors; DL/LaA, lateral acceleration; DL/YAW, yaw (rate) sensors; DL/RO, rollover sensors.

TABLE 1.8
Symbols of Power Transfer Units

Power Transfer Unit	**Designation**
Multiple plate clutch	
Multiple plate clutch with viscous or rheological fluid	
Automatic or manual disconnect (e.g. disconnect hub)	
Hydrostatic drive	H
Electric drive	E

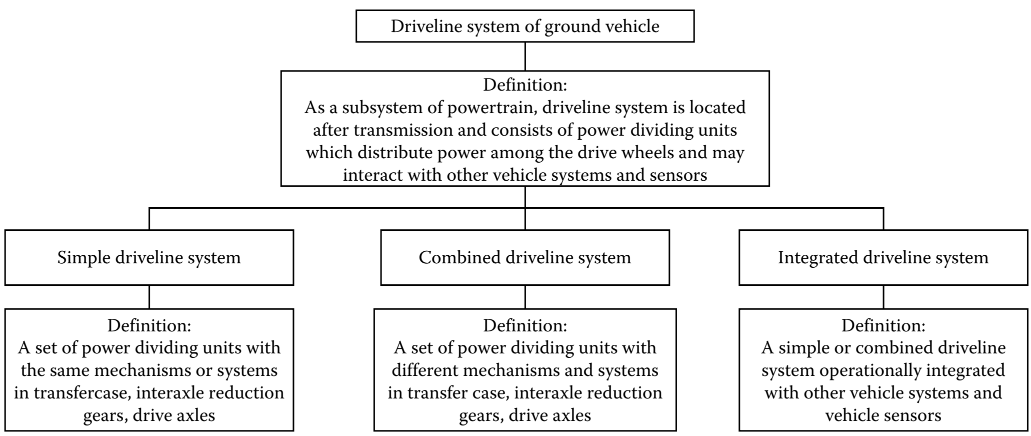


FIGURE 1.34
Simple, combined, and integrated driveline systems.

Note that the number of PDUs for a vehicle with a single engine and an axle system is always by one less than the number of the driving wheels. For example, a 12×12 vehicle has eleven PDUs. It is useful to note in connection with this that vehicles with a single driving axle may have either simple or integrated driveline systems, since they have only one PDU between the left and right driving wheels.

For characterizing wheel driveline systems as a part of the powertrain it is convenient to use powertrain layouts that are composed of gears, shafts, connected by couplings, locking clutches, splined and keyed joints, hinges and bearings. Table 1.9 lists the most typical elements. Certain other elements, not contained in Table 1.9, can be easily recognized in the layouts presented in the book.

TABLE 1.9

PTU Elements for Kinematic Diagrams

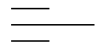
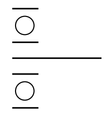
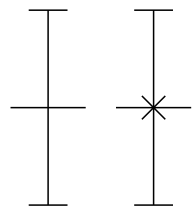
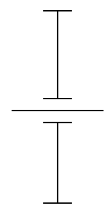
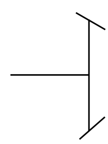
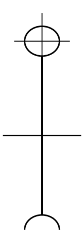
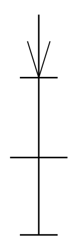
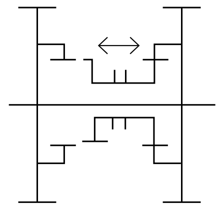
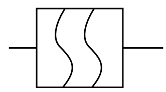
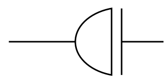
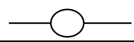
Power Transfer Unit Element	**Designation**
Shaft, axle	
Sliding bearing	
Rolling bearing	
Cylindrical gears firmly connected to shafts	
Cylindrical gears freely rotating on shafts	
Bevel gears firmly connected to shafts	
Worm—gear set	
Brake	

TABLE 1.9 (continued)

PTU Elements for Kinematic Diagrams

Power Transfer Unit Element	Designation
Gear-type coupling	
Cam-type coupling	
Universal joint	
Constant velocity joint	

Now examine examples of the use of the suggested classification of driveline systems in conjunctions with the general vehicle design.

Table 1.10 shows typical examples of general engineering layout and driveline systems of 4×4 agricultural tractors. A lockable interaxle driveline that provides for disengagement of one of the axles is widely employed both on articulated-frame tractors (e.g., K-710 or T-150K, positions 1 and 2 in Table 1.10) and tractors with steering wheels (MTZ—Minsk Tractor Works, LTZ—Lipetsk Tractor Works, positions 3, 6, and 7). Identical-wheel tractors (Profi Trac, position 4) are designed with interaxle symmetrical locking differentials. John Deere 3640 tractors with nonidentical front- and rear-axle wheels (position 5) make use of interaxle asymmetrical locking differentials. As to interwheel PDUs, the examples listed in Table 1.10 correspond to the above statistical data showing wide use of locking and self-locking (limited-slip) differentials.

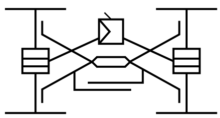
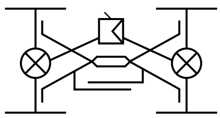
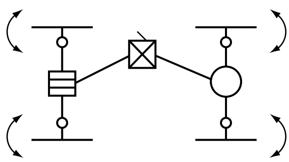
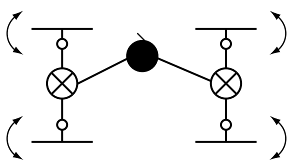
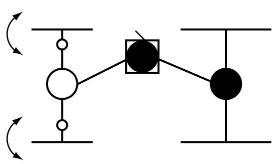
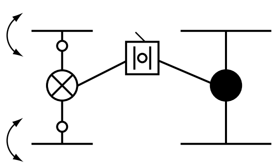
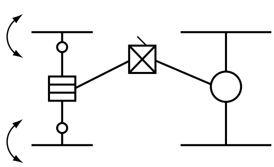
Figure 1.35 shows the kinematic layout of an articulated tractor listed under No. 1 in Table 1.10. This articulated tractor has overrunning self-locking differentials (similar to NoSPIN) in its driving wheels, which is shown by the  designation. The primary traction axle here is the front one. The rear axle is engaged and disengaged by a mechanical clutch M (see Figure 1.35).

Figure 1.36 shows the kinematic layout of the powertrain of the T-150K articulated tractor (Table 1.10, position 2). The primary driving axle of this tractor is in the rear, whereas the front axle is coupled mechanically by means of gear 19 that is capable of sliding along the shaft. The tractor is equipped with limited-slip differentials with disk-type clutches in the driving axles.

The powertrain layout of the MTZ-82 tractor, position 6 in Table 1.10, is shown in Figure 1.37. The front-axle differential P with floating pinion fingers (see Chapter 4) is a limited-slip device, whereas the rear axle differential D is locked by hydraulically controlled

TABLE 1.10

4 × 4 Agricultural Tractors

No.	Chassis and Driveline System Layout	Wheel Formula	Drive Formula	Axle Formula	Steering Formula
1		4 × 4	12	1-1	$0\overset{v}{\underset{c}{-}}0$
2		4 × 4	12	1-1	$0\overset{v}{\underset{c}{-}}0$
3		4 × 4	12	1-1	1-2
4		4 × 4	12	1-1	1-2 1-0
5		4 × 4	12	1-1	1-0
6		4 × 4	12	1-1	1-0
7		4 × 4	12	1-1	1-0

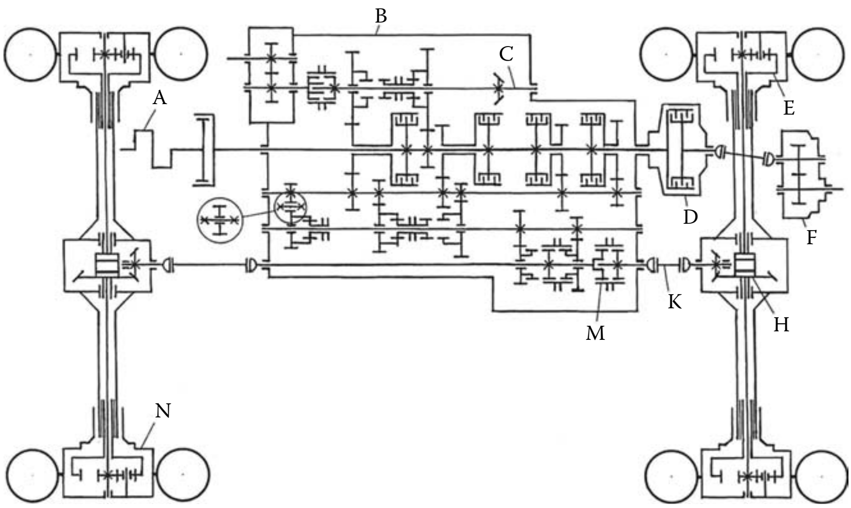


FIGURE 1.35
Tractor K-710: kinematic layout of powertrain. A, engine; B, transmission; C, pump shaft; D, power take off clutch; E, wheel-hub planetary gear set; F, power take off reduction gear; H, final drive with overrunning self-locking differential; K, drive shaft; M, positive locking engagement of the rear axle; N, front driving axle.

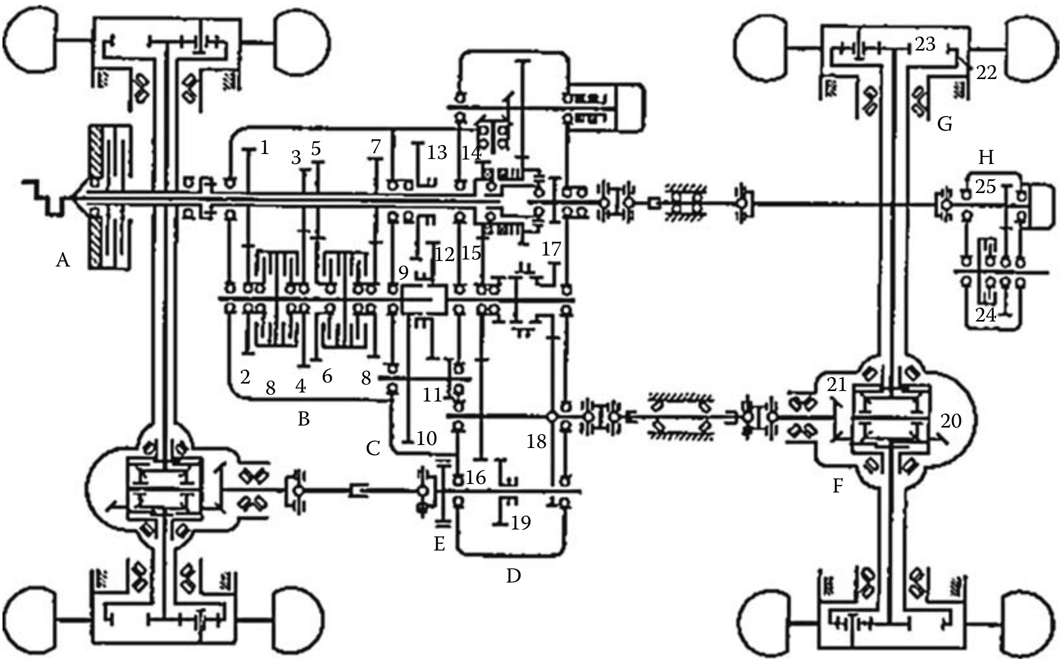


FIGURE 1.36
Tractor T-150K: kinematic layout of powertrain. A, engine with main clutch; B, transmission; C, reduction gear; D, transfer case; E, auxiliary brake; F, final drive with limited-slip differential in the rear axle; G, wheel-hub planetary gear set; H, power take off reduction gear.

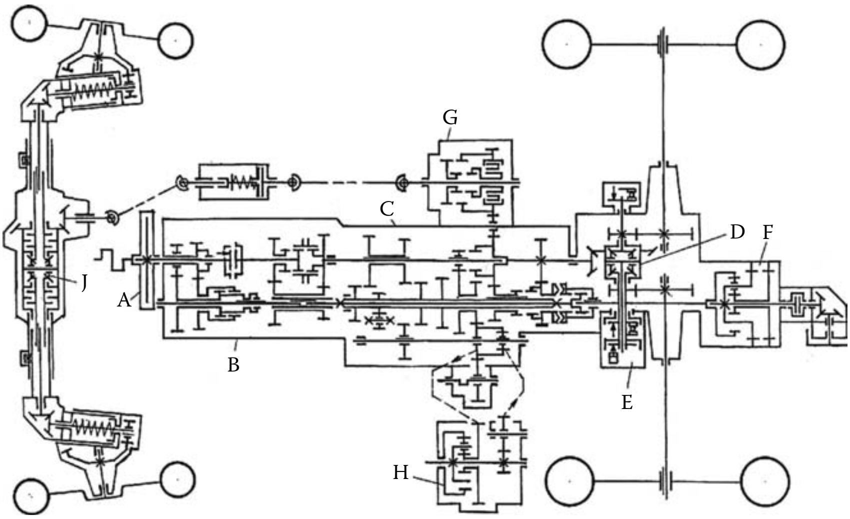


FIGURE 1.37
Tractor MTZ-82: powertrain kinematic layout. A, engine with main clutch; B, reduction gear; C, transmission; D, final drive with locking differential; E, disk brake and differential locking clutch; F, power take off reduction gear; G, transfer case with automatic positive engagement of front axle; H, reduction gear; J, limited slip differential of front axle.

clutch E. The front axle is engaged/disengaged automatically by means of a roller-type overrunning clutch, located in the transfer case G.

In addition to the fact that the kinematic powertrain layout shows clearly the location of the driveline system in the overall powertrain layout, these layouts have the added convenience of making it possible to compute the transmission ratios from the engine to the wheels when the different gears are engaged in transmission. For example, the engine power at the first gear of the T-150K tractor is transmitted to the rear wheels by gears 3, 4, 17, 18, 20, and 21 and the wheel-hub planetary reduction gear K. The gear ratio at the first gear is defined as the ratio of the number of teeth of the corresponding gears:

$$u_{\mathrm{I}} = (N_4/N_3)(N_{18}/N_{17})(N_{20}/N_{21})u_{\mathrm{k}},$$

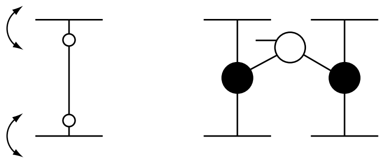
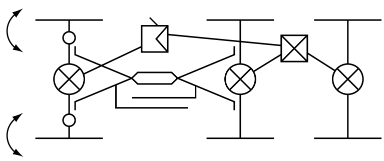
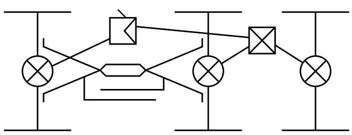
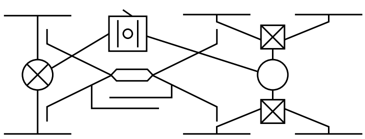
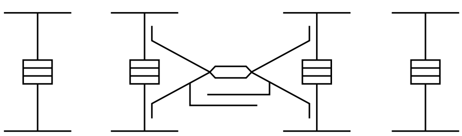
where $u_{\mathrm{k}} = 1 + N_{22}/N_{23}$ is the gear ratio of the planetary reduction gear. When the front axle is engaged, a part of the engine power starts being transmitted from the shaft of gear 18 through gear 16 to gear 19 and then to the front wheels. The front and rear axles are then rigidly coupled via the gear trains. Analytical methods of determining the power to the front and rear wheels are of definite interest for assessing the performance of the vehicles. These problems are examined in Chapter 3.

Analysis of the design of six-wheel tractors provides insight into the most typical designs and driveline systems (Table 1.11). For example, the 6×4 system is found in the Valmet 1502 tractor (Table 1.11 position 1).

6×6 tractors with a nonpermanently engaged front axle (known as part-time systems) may perform turns either by means of an articulated frame or by its combination with steered front wheels (Table 1.11, positions 2 and 3).

The 6×6 Locomo forestry tractors (Table 1.11, position 4) employ a differential drive between the side reduction gears of the forward rear and rearward rear axles.

TABLE 1.11
6 × 4, 6 × 6, and 8 × 8 Farm and Forestry Tractors

No.	Chassis and Driveline System Layout	Wheel Formula	Drive Formula	Axle Formula	Steering Formula
1		6 × 4	023	1-2	1-00
2		6 × 6	123	1-2	$1\frac{v}{c}00$
3		6 × 6	123	1-2	$0\frac{v}{c}00$
4		6 × 6	123	1-2	$0\frac{v}{c}00$
5		8 × 8	1234	2-2	$00\frac{v}{c}00$

The 8 × 8 Locomo forestry tractors (Table 1.11, position 5) make extensive use of NoSPIN differentials in their driving axles.

It should, however, be pointed out that there is a limit to increasing a tractor's traction and velocity performance by merely adding driving axles, something that would deliver more of the engine's power to the drawbar. This problem is solved by using a high-power tractor that, while towing a part of the trailer and/or semitrailer vehicular train also serves as a power module that supplies power to the wheels of another part of the train. These wheel drives are normally powered by a power takeoff shaft from the power module and less frequently by separate motors powered by the power module.

Table 1.12 presents examples of such vehicular trains comprised of a 4 × 4 power module and several trailers powered by it. Ordinarily the powered trailers are arranged in the rear of the power modules (Table 1.12, positions 1–3, 5). The front location (Table 1.12, position 4) is used when the power module provides power to agricultural implements that are propelled by power supplied by it.

In addition to driving the motorized towed units by means of a synchronous power takeoff shaft from the power module (Table 1.12, positions 1 and 3) these may be driven by