INVESTIGATION OF

TUBE TRANSIT SYSTEMS

A Thesis

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by

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Abstract

This thesis contains a detailed literature survey of tube transit systems. Of particular interest in this survey is the technical information on different types of vehicles and guideways, suspension systems and propulsion systems. An air bearing suspension system and combination pneumatic plus gravity propulsion system are proposed and investigated. In particular, control of the pneumatic system is investigated in detail and resulted in a controller which performed adequately in controlling the velocity and acceleration of the vehicle.

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Chapter 1. Introduction

1.1 The Transportation Trend

The increasing population concentration in the large cities and within the area between these cities has resulted in the growth of corridors of high population density. People move along these corridors in linear flow, as contrasted with the area flow or radial flow of urban transportation travel.(1)* The chief modes of corridor transportation have been by automobiles along expressways, airplanes, and railroads. The first two modes of transportation are near their capacities for safety and efficiency and present railways are generally outdated and undesirable for passenger travel. As a matter of fact, superhighways are loaded to capacity as soon as they are built and are beginning to saturate our landscapes. Air travel is inefficient for relatively short trips, and some air lanes are crowded to the danger point. Moreover, railroads move passengers no faster than they did 50 years ago, and they lose money in the process.(2) Furthermore, there is an increasing requirement on transportation to reduce door-to-door travel time. So the improved coordination of intercity transport systems will be needed to fill this requirement. In addition, an increase in transit speeds will also be required. Therefore, the extrapolation of existing railroad, highway, and airway solutions to the increased travel demands of the 1980's and beyond may not be adequate. Transportation also must fill the requirements for population increase, increase in business

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*(1) refers to first reference.

activity, leisure time that people may have due to industrialization, affluent living, and mobility of jobs. These demands overload new facilities as soon as they are built. Thus there is a need to develop a new transportation system to fill these requirements.

The achievements of some development programs in specific areas show that there are a large number of factors involved in evaluating the methods of providing new transportation systems which are possible in today's technology. Therefore, alternative new approaches need careful research and study. High speed ground (or underground) transport is a prime example of such a new mode which could provide safe and high-capacity transportation.

A high speed ground transportation system has been constructed between Tokyo and Osaka in Japan. This system is called Japanese New Tokaido Line. An interesting side effect which resulted when the Japanese New Tokaido Line was in full operation for one month was that the Japan Air Lines suffered a 38 per cent decline in patronage on the Tokyo-Osaka route. To the Japanese businessman, the three-hour city-center to city-center train trip apparently looks attractive in comparison with the air trip which, with poor ground connections, takes a similar amount of time.[3] So it is interesting to speculate on the impact of an even higher speed ground transportation system in this Japanese corridor. Typically, population corridors such as the one in Japan are several hundred miles long and 50 to 100 miles wide.[1] Such corridors existing in the U.S.A. are the Northeast Corridor between Washington, D.C.

and Boston, and the Pacific Corridor between San Diego and San Francisco. There are many other routes that would offer similar potential, including New York-Chicago, New York-Miami, and so on. So it also will be interesting to study the possibility of higher speed ground transportation in these American corridors.

1.2. High Speed Ground Transportation

Several development programs have been performed to investigate different modes for high speed transit systems. These studies have shown that ground transportation at speeds considerably higher than those available today are technologically feasible. These systems will, however, differ significantly from the presently available modes of transportation.

Many studies have been made on possible high speed mass transit schemes around the world. Included in these studies are the following types of systems:

(1) High Speed Rail System. Examples are the Japanese New Tokaido Line and the San Francisco Bay Area Rapid Transit (BART). The New Tokaido Line was opened to traffic in October, 1964. It traveled a 320 mile route between Tokyo and Osaka, and has 19 intermediate stations. It is powered by electric motors supplied by 25 Kv. single phase 60-cycle source and all passenger services are operated by multiple-unit 12-car trainsets capable of running at 130 mph.[4]

BART, which will serve San Francisco and its urban community, will be 75 miles in total length, double tracked, and will make use of automatic high-speed operation concepts. BART is designed to attain speeds up to 90 mph between stations.[4] BART is scheduled for completion in 1972. Its total cost is projected at 1.2 billion dollars, most of which has been provided by local funding.

(2) Tracked Air Cushion Vehicle (TACV) System. There are many examples such as French Aerotrain, British Tracked Hovercraft, General Motor's Hovair Vehicle, and Grumman Tracked Air Cushion Vehicle (Grumman TACRV). Among these schemes, only the French Aerotrain, carrying 80 passengers at a cruising speed of 207 mph., is in operation.[5] The French Aerotrain has a flexible skirt, open plenum air cushion system for vertical and lateral support and runs on an inverted tee section monorail. The British Tracked Hovercraft is designed to run on a rectangular cross-section monorail with a linear induction motor for propulsion. General Motor's Hovair Vehicle design employs a series of flexible-plenum, air cushion pads beneath the vehicle.[3] The Grumman TACRV is designed to go 300 mph. and utilizes a linear induction motor for propulsion.[6] The types of suspension and propulsion systems mentioned here will be discussed in detail in Chapters 4 and 5.

(3) Linear-Induction-Motor (LIM) Vehicle. Examples include

German Linear Induction Motor Car with Electromagnetic Suspension, Grumman Tracked Air Cushion Vehicle (Grumman TACRV) using linear induction motor as propulsion, and Linear Induction Motor Vehicle developed by the Garrett Corporation. The experimental German Linear Induction Motor Car has an electromagnetic suspension, weighs 11,500 pounds, is 25 feet long, 7 feet wide, and 6 feet high. It utilizes a linear induction motor which provides 2,200 pounds of starting thrust. This vehicle uses a magnetic suspension which is made up of eight levitation and four guidance magnets.[7] The Linear Induction Motor Vehicle designed by the Garrett Corporation is the only vehicle of this type in operation in the United States.

(4) Tube Vehicle System. All the systems of this type will be discussed in Chapter 2. These systems are Tubeflight, proposed by Dr. Joseph V. Foa of Rensselear Polytechnic Institute[8]; Gravity Vacuum Transit, first proposed by L. K. Edwards of Johns Hopkins University, then by Tube Transit, Inc., Palo Alto, California[1,2,9]; State-of-the-Art Tube Vehicle System proposed by M. King and J. W. Smylie[10,11]. Fast Transit Link proposed by E. G. Chilton, of Arizona State University[12]; High Speed Ice Train proposed by M. M. Eisenstadt, of University of Puerto Rico and J. D. Riley of Stanford University[13]; and Steam Tube Transit System, proposed by Dr. Raymond Chuan of the Susquehanna Corporation's Atlantic

Research Division in Costa Mesa, California, and Norman V. Peterson, a Los Angeles consultant(14).

(5) Auto-passenger, Train-Ferry System. Examples include the RRollway, proposed by General American Transportation Corporation(3), and Cabin Tax proposed by Dusseldorf traffic engineer, Ulrich Voss(7).

(6) Automated Highway System. Example is Highway Automation proposed by R. E. Fenton and K. W. Olson of Ohio State University(15, 16).

1.3. Outline of This Study

Of the aforementioned transit systems, this study will concentrate on the tube transit system. This system is probably the least developed of the systems. Since literature on the tube transit systems is somewhat scarce, a comprehensive literature search into various studies of this type of system will be a portion of this thesis.

This literature search is presented in Chapter 2 with additional literature on suspension and propulsion systems being presented in Chapters 4 and 5, respectively. Also presented in Chapter 2 is a brief review of some currently proposed tube transit systems. This study makes a suggestion of a new pneumatically powered and air-bearing suspended tube transit system.

Chapter 3 suggests some design criteria for the vehicle and its guideway. A discussion of tunneling is also presented. Also, the effects of the radius of curvature of the guideway and vehicle clearance on vehicle length are determined in this chapter.

Some suggestions on suspension design criteria are presented in Chapter 4. This study proposes some possible ways to furnish the suspension. Among these, the air-bearing suspension is investigated in detail. The governing equations for the air-bearing and friction due to the air bearing are studied.

Chapter 5 suggests some possible propulsion systems and the design criteria for propulsion systems. Among these, a detailed study is made of a pneumatic propulsion system. The control of the vehicle velocity during acceleration is studied in detail in this chapter.

Finally, conclusions and recommendations are stated in Chapter 6.

Chapter 2. Tube Transit Concepts

2.1. Definition of Tube Transit System

The tube transit system is a system that requires the vehicle to travel in an entirely enclosed guideway and will provide safe ultra-high speed intercity transportation. Reasons for the complete enclosure may include the ability to reduce pressure, thus reducing aerodynamic drag,and providing continuous support for the vehicle suspension. Also, the enclosed guideway provides all-weather capability, avoids unfavorable terrain, and allows for location of the guideway where it will not mar the landscape.

There may be a misunderstanding that the subway is a mode of tube transit system, in view of the vehicle moving in the entirely enclosed guideway. In reality, this is not the case. Subways are merely an underground railroad, and are designed to operate mainly within the city or inner and outer urban areas. The tube transit system provides intercity transportation along the corridors of high population density and is a form of high speed ground (or underground) transportation which would differ from railways as we know them today. Finally, the tube transit system could be capable of achieving an ultra high speed, 300 to 500 mph (miles per hour), but the subway moves at the same speed as the railway.

2.2. Advantages and Disadvantages of Tube Transit Systems

The best enclosed guideway that can provide a safe and all-weather capability is the tube. Also, the tube itself provides the lateral support and guidance for the vehicle while it is moving at very high speeds. This function eliminates the important drawback of many surface high speed vehicles, that is, their lack of lateral control. Furthermore, the tube can operate as a waveguide for communication systems and perhaps for the transmission of electric power for vehicle propulsion.

The tube transit system could transport people comfortably, efficiently, and rapidly over distances of about 40 to 500 miles[(1)]. In this range, the tube transit system could offer some advantages. It would have shorter trip times than those of surface transportations having the equivalent comfort level. It also could provide safe, automatically controlled transportation with higher capacity than air travel. Furthermore, it is entirely independent of weather because rain, snow, or ice can not block trains and fog cannot close down the terminal. It would be light in weight without danger of being blown over in a high speed wind. It would not give danger to wandering animals and mischievous children. It also would use little or no surface land or air space. Its terminal could be in the heart of the city with little disruption of what is there, or in a peripheral location best matching feeder transportation, with little community objection[(1)].

The tube can also be evacuated to reduce aerodynamic drag. Also, gravity can be used to assist in acceleration and braking. These two factors will reduce overall energy and equipment requirements compared to an equivalent atmospheric system. If the tube transit system makes use of gravity to assist in acceleration and braking, it could bring the tunnel depth to several thousand feet below the ground level. This could afford the following important benefits:(2)

(1) There is no weathering because the temperature at depths below 50 feet is always constant and above the freezing point of water.

(2) It would bring most of the tunnel down into deep bedrock which is very homogeneous in consistency. Also, there is a low likelihood of water inflow at these depths. This would reduce the cost of tunneling, which could be achieved either by blasting or boring. The chance of earth shifts would also be reduced at these low levels.

(3) The nuisance to property owners decreases with depth, so that cost of easements should be lower.

(4) A deep tunnel does not interfere with subways, building foundations, utilities or water wells.

(5) The vehicle coasting along a guideway of suitably calculated slope, gives the passengers a comfortable feeling.

(6) In the event of an emergency, air would be admitted to the tube from the tunnels, which open to atmospheric air, simply by opening valves between the tube and these tunnels.

Finally, the tube transit system does not directly pollute the atmosphere and any pollution occurring within the tube may be controlled.

Contrary to these advantages is the high cost of the guideways and tunnels. This cost depends greatly upon the required precision of construction and alignment of the tube in order to provide a safe and comfortable ride for the passengers. To emphasize the level of costs which might be incurred, Tube Transit Corporation has estimated the cost of one such system, based on our present day technology, at 25 million dollars per mile. This is many times the cost of an expressway, but as technology develops and society requirements change, the development of such a system is indeed a future possibility. Another disadvantage may be the relatively high degree of earthquake hazard exposure of the tube transit system relative to surface transportation systems.

2.3. The Elements and Selection of the Tube Transit System

The development of any tube transit system is very complex and requires the study of many subsystems which are necessary for its successful and safe operation. These subsystems include the vehicle, its guideway, vehicle power, passengers, control systems, and safety considerations. Considerations which must be made in developing these subsystems include social and ecological issues such as effects on people in nearby communities, required power generating facilities, transportation feeders for passengers to and from the tube vehicle terminals, the administrator of the system, the tunnel construction, and so on. These elements must be carefully investigated individually at first, but they only have full meaning when they are considered together as an integrated whole.

Many of the above considerations have been investigated by others in projects sponsored by the Office of High Speed Ground Transportation (OHSGT) within the Department of Transportation. These studies have looked at many combinations of elements which could be combined to form a successful tube transit system and resulted in the selection of three possibilities which seem feasible. These possibilities are shown as follows:(1)

(1) Propulsion by a linear induction motor, wheel suspension, and low pressure air within the tube.

(2) Pneumatic power, wheel suspension, and low pressure air within the tube.

(3) Power supplied by a gas turbine, ram wing suspension, and atmospheric pressure air within the tube.

In the first possibility listed, the linear induction motor has an electrically powered stator in the guideway and a passive rotor fin on the vehicle. This technique would work satisfactorily in a vacuum. The stator coil would be located at intervals along the roadbed due to its high costs. This method yields a boost-glide operation which is an intermittent power impulse and subsequent coast. This method may be unpleasant for the passengers, since they would feel an alternate acceleration and deceleration, but may be economical for short stages having high traffic density. Also, it is conceivable that rapid changes in acceleration and deceleration can be designed out of the system.

The pneumatic powered system is best in a boost-glide operation. The vehicle is pushed by air which is admitted behind the vehicle in an evacuated tube or it is pushed by air combined with the use of gravity to get the desired cruise speed. It then coasts to the next stop. The tube must be re-evacuated before the next vehicle is placed in operation. The necessary use of valves and air inlets causes some complexity and high cost. The capacity is limited because of the required power of air pumps to evacuate the guideway being excessive.

The gas-turbine propulsion system is limited in its capacity because the vehicle cannot be effectively coupled into trains because propellers on the successive vehicles will not develop enough thrust in the turbulent air region created by the preceding vehicle(1). It also produces noise and exhaust products which must be blown out of the tube through vent shafts, consequently creating air pollution.

As for the suspension system for these three possibilities, in two of the systems, wheels on rails are proposed as the primary element for supporting and guiding the vehicle along the tube. Since most of our present ground transportation modes utilize wheels, a great deal is known about this form of suspension. The research work required would be somewhat less than for some of the more novel suspensions. One of the major drawbacks with wheels is that the vehicle cannot be driven at very high speeds due to the retarding force of bearings and suspension systems. At very high speeds, prohibitively costly tolerances would be required in order to avoid destructive dynamic loads in the bearings and at the contact point between the wheel and rail. Another difficulty which may be experienced in a very low pressure tube environment is that the friction coefficient between the wheel and rail will be significantly greater than that at atmospheric pressure. This will result in an additional retarding force which will tend to slow the vehicle down.

The ram wing suspension on the third system is a form of air cushion support. The low coefficient of friction at higher speeds is a particular advantage of the ram wing.

In conclusion, the selection of a system depends on the elements that are needed for a specific location or environment. Further studies of the elements are needed in order to develop optimum tube transit systems.

2.4. A Review of Currently Proposed Tube Transit Systems

2.4.1. Tubeflight[8]

The third method of the previous section has been the topic of several studies by researchers within the aeronautical engineering department at the Rensselaer Polytechnic Institute. The system proposed by them, called Tubeflight, has an aerodynamically supported vehicle traveling in a tube at atmospheric pressure. The upper speed limit for the vehicle will be between 200 to 400 miles per hour (mph).

The tubeflight vehicle propels itself by means of an on-board flow induction device which transfers air from the front to the rear of the vehicle. This induction device will utilize either a fan or a propeller which will be powered by an electric motor or an internal combustion engine.

This mode of propulsion is called internal propulsion. The power required for propulsion depends not only on the gas dynamics of the transferred air flow from in front of the vehicle to its rear, but also upon the amplitude of the disturbances that are generated by the vehicle in the far field (a significant distance in front of the vehicle).

Active control devices on-board the aerodynamically suspended vehicle limit roll instabilities which may occur.

The vehicle could contain up to 250 passengers and would be equipped with an aircraft type air conditioning system.

Large-clearance peripheral jet air cushion pads are located at 120 degree intervals around the vehicle for suspension. At high velocities these pads operate as a wing close to the ground (a ram wing) with jet flaps. Many of their studies have concentrated on the aerodynamics of this concept.

The guideway is a circular tube having a diameter of at least 15 feet. Since the vehicle is able to tilt itself to the correct bank angle on turns, it is possible to obtain a relatively sharp turning radius without causing severe suspension difficulties.

The use of microwave power is suggested as a means of powering both their electrical driven system and the environmental control system in the vehicle. However, microwave power is still conceptual and has not been developed on the scale necessary to power a large vehicle.

The tubeflight system is the only atmospheric pressure system which has been studied very extensively. Atmospheric pressure has some advantages in that air for suspension, internal propulsion, and environmental support inside the vehicle is readily available. The major disadvantage is the aerodynamic drag and limitations in the passenger capacity.

2.4.2. Gravity Vacuum Transit [1,2,9]

The Gravity Vacuum Transit (GVT) system was first proposed by L. K. Edwards of Johns Hopkins University, then by Tube Transit, Inc., Palo Alto, California. GVT system makes use of gravity and pneumatic pressure for its propulsion and is supported by wheels riding on tracks. The pneumatic propulsion force is obtained by applying air at atmospheric pressure behind the vehicle while the pressure in front is reduced. The acceleration due to the gravity force is simply obtained by means of the vehicle moving along a guideway with a slightly downward slope. This propulsion system does not require any transfer of electrical power from the ground to the vehicle. This is a prime advantage of this system. However, an elaborate flow control and vehicle sensing system would be required.

If gravity is used for propulsion, the guideway tunnel could be located as much as 2,000 feet below the surface. The GVT tunnel contains a pair of side by side steel tubes supported on springs, for the vehicle guideway. The guideway is evacuated by large electrically powered pumps. The spring suspension for the guideway along with accurately aligned and smooth rails would help achieve a comfortable ride.

The vehicle is a cylindrical steel vessel which is designed for operating in a vacuum tube. Its interior is similar to that of present day jet aircraft. Fresh air is circulated through the vehicle at each station. Several vehicles may be joined together to form a train via a flexible joint above the wheel wells at the ends of each vehicle. Front and rear vehicles of the train are equipped with necessary auxiliary equipment for the vehicle operator and his crew.

2.4.3. State-of-Art Tube Vehicle System[10,11]

The State-of-Art Tube Vehicle System proposed by M. King and J. W. Smylie is a concept similar to that of Gravity Vacuum Transit. Here, the vehicles are operated in a reduced pressure environment in deep tunnels in order to reduce the aerodynamic drag and make use of gravity to assist in acceleration and deceleration.

The vehicle is designed with a width of 10.5 feet and weight of 68,000 pounds. The vehicle is designed to accommodate 72 passengers and is powered by four 150-horsepower DC traction motors. Wheel support is suggested as the suspension of this system.

2.4.4. Fast Transit Link System[12]

The Fast Transit Link System was proposed by Z. G. Chilton, Professor of Engineering Science at Arizona State University. The design of Fast Transit Link (FTL) System combines the advantages of Gravity Vacuum Transit with magnetic suspension and a linear induction motor for propulsion.

At start up, the linear induction motor and gravity are used to accelerate the vehicle at 6.8 mph/sec until the speed reaches 250 mph. In steady state operation, the vehicle is propelled by the linear induction motor and proceeds horizontally at constant speed until it reaches the up ramp leading to the next station where gravity slows it down. The vehicle is constructed with a pressure-tight fuselage without windows and has two sets of sliding doors for entrance of passengers. A conservative maximum estimate weight is 22.5 tons, which includes 9.6 tons for shell and seats, 3.5 tons for the suspension, 5.5 tons for propulsion and electric system, and 3.9 tons for 52 passengers.

A magnetic suspension, which could provide a smooth and comfortable ride, is proposed for FTL systems. The magnetic suspension, using superconducting coils on the vehicle above an aluminum track, affords the ability to brake, independent of electric power supply, and the availability of damping.[17]

Each vehicle is propelled by a two-sided linear motor on its roof, straddling an aluminum secondary winding attached to the ceiling of the tunnel. The linear induction motor develops a maximum power of 1500 hp.

The pressure of the tunnel is maintained at 0.1 atm by means of vacuum pumps located one mile apart on or near the ground above the tunnel. Each pump can handle approximately 8,000 scfm of air. This vacuum condition in the tunnel requires that the vehicle must be tight at all times to keep air from leaking out because human exposure to the tunnel vacuum would be fatal.

2.4.5. High Speed Ice Train[13]

M. M. Eisenstadt, Associate Professor of the Department of Mechanical Engineering, University of Puerto Rico and J. D. Riley of the Department of Aeronautics and Astronautics, Stanford University, have proposed this High Speed Ice Train.

The ice train, which is propelled by a linear induction motor and moves on ice, would operate at speeds between 350 and 400 mph in a tunnel. The basic concept is that ice is attached to the train and the friction at the ice-track interface will cause melting and thus create a lubricating layer of water. Since the propulsion is provided by a linear induction motor, the ice train can be operated in an evacuated tunnel. The drag due to the water lubricating layer is higher than the drag for either air-cushion or magnetic suspensions. The melting of a 24-inch thick ice block placed under a train would provide suspension for a 300-mile trip at 350 mph.

2.4.6. Steam Tube Transit System[14]

The Steam Tube Transit System was proposed by Raymond Chuan of the Susquehanna Corporation's Atlantic Research Division in Costa

Mesa, California, and Norman V. Peterson, a Los Angeles consultant. This system is designed to move a vehicle through a tunnel at more than 400 miles an hour. The designers have experimentally verified the system's concept in the laboratory.

Being propelled by a pressure difference ahead and behind, the vehicle requires no propulsion power on board. The vehicle could easily achieve a speed of 400 miles an hour by applying steam at four psi at the rear of the vehicle and a pressure of 0.5 psi in front of it.

The reduced pressure in front of the vehicle is achieved by the condensation of steam vapor along the walls of the tunnel. The condensed water behind the vehicle will tend to evaporate because of the partial vacuum created by the rapidly moving vehicle. Apparently, this scheme is designed to keep a slightly higher pressure at the rear of the vehicle during its cruising operation. A few steam booster stations are required along the route to provide the higher pressure behind the vehicle.

2.5. Description of the New System Proposed by this Study

As can be seen by these many different types of tube transit schemes which have been investigated, there are many routes one could take in designing such a system. There are also a great many constraints which must be satisfied in order to get a feasible working system.

This study is intended to investigate the various possibilities for such a system and generate a propulsion and suspension system which, though rather technologically advanced in concept, would fit the constraints and be a feasible system for the future. The system will not be totally developed in this work but will be studied to the point of feasibility from an engineering standpoint.

The system proposed by this study will be pneumatically powered and suspended with an air-bearing. In looking at propulsion and suspension systems, it is also a necessity to investigate the general configuration and dimensions for the guideway and the vehicle. The guideway and vehicle dimensions will not be investigated in detail, but will only be developed to the extent that they may be utilized in the analysis of the propulsion and suspension systems.

Chapter 3. Vehicle and Guideway

3.1. Introduction

The vehicle and its guideway are the most apparent subsystems of a tube transit system. Basic vehicle and guideway criteria must be satisfied in order that the passengers receive a safe, comfortable ride. Among these criteria are maximum acceleration rate, construction tolerances, costs, and guideway shapes. The following section will discuss these and other guideway and vehicle criteria and will set limits for use in the design of suspension and propulsion systems.

3.1.1. Guideway

The guideway subsystem consists of many elements such as the tunnel, rail and rail supports where used, and tube. The term track mentioned in this study refers not only to wheeled tracks but also to the ground-vehicle interface for any air cushion or magnetic type of suspension.

For the underground tube transit system, tunneling is required. Tunneling is heavily influenced by underground geological conditions whose characteristics may be known with varying degrees of uncertainty. The Office of High Speed Transportation suggests some typical tunnel cross sections such as circular, horseshoe, vertical sidewall, and basket handle. Those shapes are shown in Figure 1. The choice of cross section depends on the geology, type of suspension required, and excavation techniques. The tunneling costs are a function of

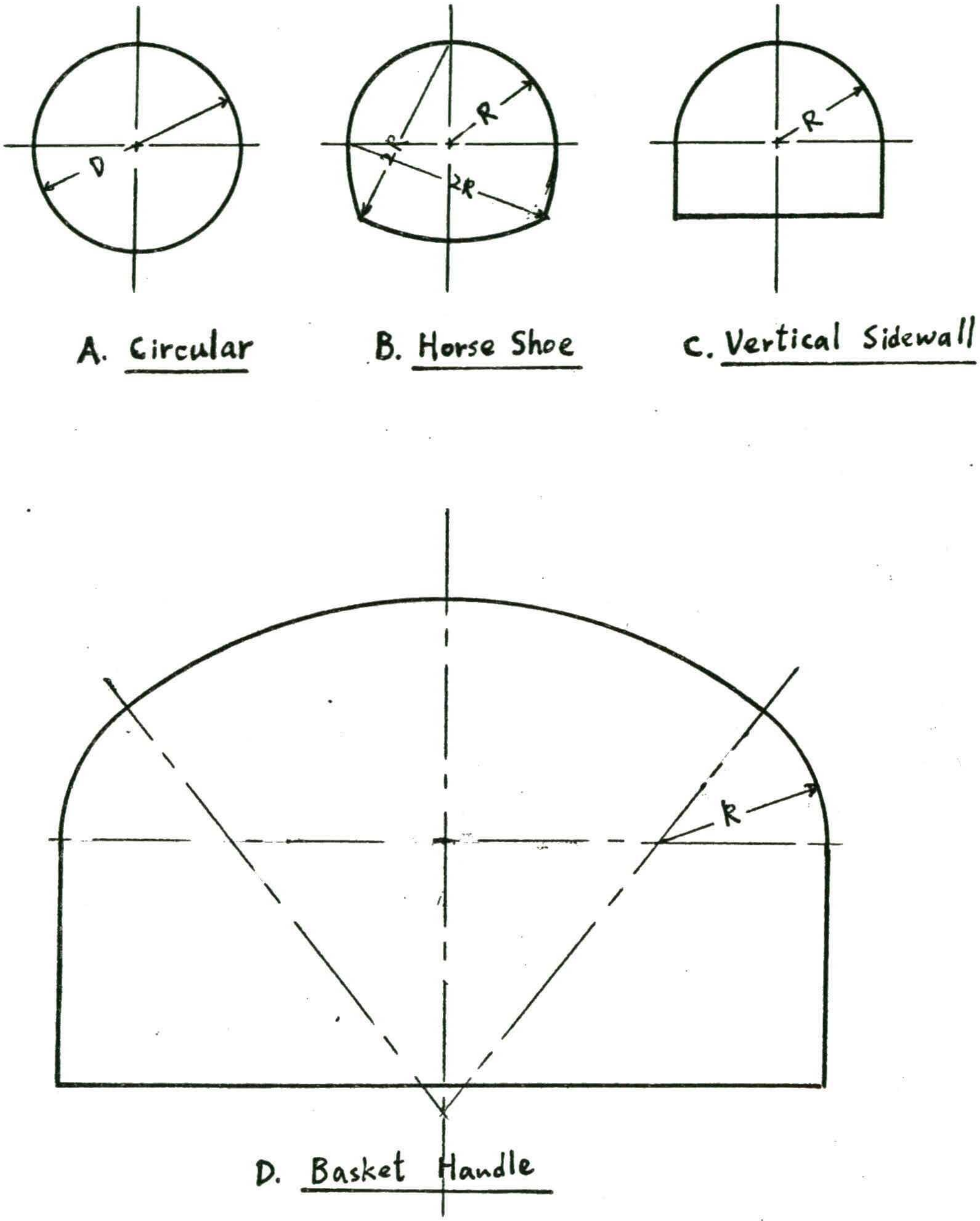

Figure 1. Typical Tunnel Sections (After Mayo)[1].

geological conditions, tunnel size, and shaft spacing. Geological conditions influence tunneling costs the most, because geological characteristics affect excavation techniques, support requirements and control of underground water. Tunneling costs are not of prime concern in this investigation, but are reported by the California State Department of Water Resources. This report goes into detail on the costs of building underground water ducts by conventional drilling and blasting techniques.[18] The cost of tunneling techniques is a function of transport of excess soil and rocks to the surface, pumping during excavation, pumping during liming, and ventilation during excavation. The effects of the various tunneling parameters, such as rock quality, depth of profile, and diameter of tunnel, upon costs have been investigated by Harza[19] and by Mayo[20].

As for rail and rail support for high speed operation, the rails for the Japanese New Tokaido Line furnish a typical example. The rail is made of high carbon steel with about 1 per cent manganese and 0.04 per cent silicon and is welded. It weighs 107 pounds per yard (lb/yd) and is installed in 4900-foot lengths[1]. New rails are being developed in America and include the Holloman Air Force Base high speed rocket sled test track and RZ-140 developed for the Penn Central Metrolinear Train. The accurate placement of the rails and the maintenance of their positions becomes very important when the vehicle operates at speeds higher than 130 mph, the maximum speed of the Japanese New Tokaido Line. As the train speed increases, more critical alignment of the guideway is required. As for rail support, there is a growing trend to use a concrete rail bed such as the one used by the Japanese. Also, the Germans have recently experimented with a pre-cast concrete track bed, with three different designs being tested in late 1967 in a high-speed experimental line between Forcheim and Bamburg. The reinforced concrete solid bed track tried in service on the Milan Metropolitan Railway and the Adda high speed interurban train in Italy in 1964 is another example of the use of a concrete rail bed.

The Office of High Speed Ground Transportation suggests a steel tube through which the vehicle could pass. Some steel tube characteristics, such as design requirements, tube tolerance, and tube support, are investigated in Reference 1.

3.1.2. Vehicle

These criteria depend upon one another a great deal and can get quite detailed if one investigates all aspects of them. In this study, only those aspects affecting the vehicle's outer dimensions will be investigated.

The configuration of the vehicle is determined by the passenger accommodations and type of suspension system used. The desired passenger accommodations affects the cross sectional area of the vehicle. If pneumatic propulsion is used, the pressure and velocity needed for the system also can affect the cross sectional area of the vehicle. The suspension system determines the shape of the bottom of the vehicle.

The energy consumed by the vehicle is equal to the summation of the energy to drive the vehicle against its mechanical and aerodynamic resistance and the energy to accelerate and brake the vehicle. Among these, the energy to drive the vehicle against mechanical resistance (friction) and the energy to accelerate and brake the vehicle are functions of the vehicle's mass. If the vehicle moves uphill, the energy needed against gravity is also a function of the vehicle's mass. Therefore, a lighter weight is desired since it will reduce the total energy consumed. Also the tube provides an additional advantage of avoiding the danger of being blown over in a high wind no matter what the vehicle weight is. Therefore, a lighter weight vehicle is preferred.

The tube transit system will be competing with the airplane and should have an interior design which is comfortable for passengers. This will put a minimum limit on the interior size of the vehicle. Also, consideration should be given to allow for joining several vehicles in train form.

The vehicle design must fulfill the requirements not only for structural strength, but also should provide a comfortable environment. Also to be considered are the vibration and noise of the internal mechanical equipment and vehicle suspension. Environmental control, such as temperature, humidity, oxygen concentration, carbon dioxide concentration, trace contaminant concentration, and pressure must also be considered in designing the vehicle interior. The Office of High Speed Ground Transportation has proposed a comfortable environment for people which specifies a temperature of 63°-70° in winter, and 66°-75° in summer, a pressure being equal to the terminal ambient pressure ± 0.06 psi, carbon dioxide being less than one per cent by volume, and oxygen partial pressure between 2.57 psi and 3.09 psi. While the vehicle moves within the evacuated tube, the air circulation in the vehicle must be considered. The vehicle must be sealed in order to prevent air from leaking out since depressurization of the vehicle while operating in an evacuated tube would be extremely dangerous.

3.2. Guideway Criteria

The guideway must provide a safe, smooth, and comfortable ride. In this function, the guideway interacts with the suspension system and results in lateral motion of the vehicle. In order to keep these motions to a minimum, strict guideway tolerances must be adhered to. In addition, the guideway, which includes all supporting structure and internal tube structure, must be safe and externally pleasing. Therefore, in selecting guideways, five criteria have been selected:

(1) Comfort criteria

(2) Safety criteria

(3) Social criteria

(4) Aesthetic criteria

(5) Cost criteria

In other modes of transportation, measures of comfort have been specified which related to the passenger's feeling and to the smoothness of ride. These same measures, which include vibration, noise, and temperature, will be applied to tube transit systems. To be competitive with aircraft, it is necessary that tube transit systems be much more sophisticated in these areas than the trains which are presently in service. The smoothness of the ride is dependent upon the dynamic response of the vehicle to perturbations along the guideway surface. To achieve a very low level of vehicle response, it is necessary to keep these perturbations very small. This can be achieved by accurately aligning the tubes, keeping track surfaces smooth, and keeping welded or expansion joints smooth. The vibration motion of the guideway due to vehicles passing along it is also a factor.

As for safety criteria, enclosing the guideway of a tube transit system eliminates many safety problems of conventional surface systems. Examples include keeping people and animals from being struck by the train or from being electrocuted by its power lines and providing protection from wind, rain and other weather conditions. But some safety problems are created with tube transit systems. Newly created safety problems are largely related to the evacuated environment of the tube and the great depths which may be encountered. In case any emergencies occur, the guideway and vehicle must have the capability to bring the vehicle or passengers safely to the surface.

Social criteria include minimizing community dislocation, avoiding pollution to the service area of the system, reducing noise, eliminating the congestion of commuters, and gaining community acceptance. These criteria must take into account land use and zoning policies.

Aesthetic criteria are needed only for the vehicle interiors and terminals. But there may also be a need to use aesthetic design approaches in areas where the tube will be exposed to the public.

Cost criteria are simply those of keeping the costs as economical as possible.

3.3. Curved Guideway

It is desirable for all tube transit vehicles to travel in straight lines. However, due to underground rock structure, locations of cities and tunnels, and the need to travel up and down, it will be necessary to provide curves along the guideway. When a vehicle moves in a curved path, the constraint from moving in a straight line results in an inertia force directed toward the outside of the curve. This force is directly proportional to the square of the vehicle velocity and inversely proportional to the radius of curvature. This inertia force is the so-called centrifugal force and must be equal to the centripetal force. The acceleration resulting from this force follows the relation

$$A_n = \frac{V^2}{R} \qquad (3.1)$$

where A_n is the centrifugal acceleration

V is the velocity

R is the radius of curvature

The radius of curvature must be such that passenger limits for comfortable lateral accelerations are not exceeded. The Office of High Speed Ground Transportation has suggested that the comfortable lateral accelerations for passengers are 0.08 g in the normal condition and 0.20 g in emergencies. But an M.I.T. report[21] has suggested that the allowable lateral acceleration may vary from 0.14 g to 0.3 g. When the vehicle moves along a vertical curve such as those which are needed to send the vehicle downward at take-off and upward at its destination, the centrifugal acceleration due to this motion similarly must fulfill equation (3.1) and the limits of comfortable vertical acceleration for passengers. Also, the Office of High Speed Ground Transportation has suggested a comfortable vertical acceleration for passengers of 1.0 $\pm$ 0.09 g in normal conditions and 1.0 $\pm$ 0.15 g in emergencies. Another reason for limiting the radius of curvature on curves is to reduce the horizontal reaction force which must be supplied to keep the vehicle centered on its path. This is particularly important for air cushion, air bearing, and magnetic suspensions. Some calculated radii of curvature which compare to these accelerations are presented in Table 1.

An (ft/sec^2)	V (M.p.h.)	R (Feet)
0.08 g	300	75,100
0.14 g	300	42,800
0.2 g	300	30,010
0.3 g	300	20,100
0.5 g	300	12,020
1 g	300	6,010

Table 1. Radii of Curvature.

3.4. Vehicle Criteria

The vehicle subsystem is largely determined by the other subsystems such as the suspension, propulsion systems, and guideway. The criteria for the vehicle that are independent of the other subsystems are the structural integrity and the passenger's safety, comfort, and convenience. The structural integrity must be such that the structure is in accordance with requirements for other subsystems and can deliver high performance. The geometry of the supporting structure must be incorporated with other subsystems and fill the requirements for design simplicity, economy of materials, reliability, and maintainability.

The safety, comfort, and convenience criteria are very important for the system because the vehicle is the prime contact between the system and passengers.

3.5. Vehicle Length

Vehicle length is determined by the interaction of the structural stiffness of materials, radii of curves, number of passengers and cargo, cost, and ease of operation. The vehicle length, constrained only by radii of curvature, will be investigated here. The maximum allowable length of the vehicle is achieved when one side of the vehicle is tangent to the inner sidewall of the guideway as shown in Figure 2. Thus a gap is created between the middle point of the vehicle and the outer sidewall of the guideway which is twice the normal clearance, C, between the vehicle and guideway when operating in a straight line.

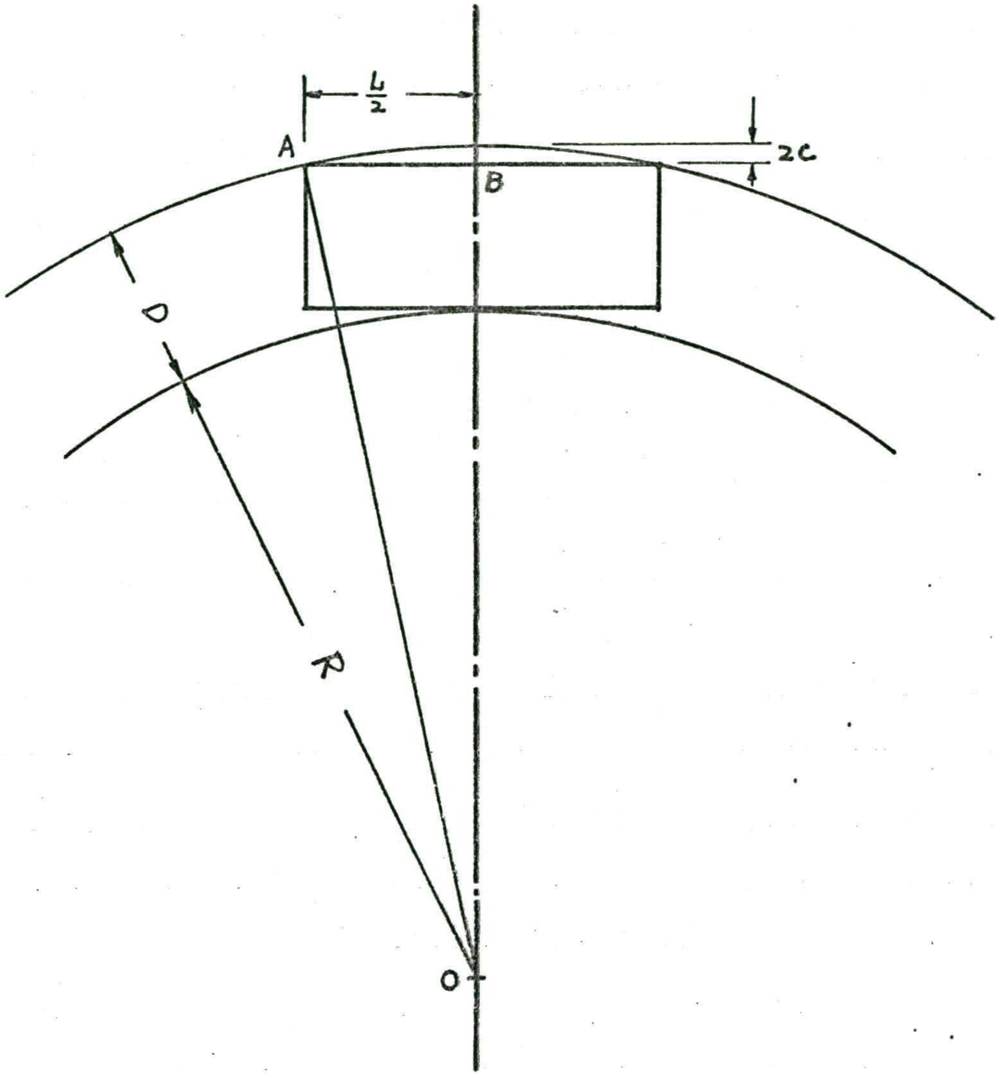

Figure 2. Maximum Vehicle Length Under the Constraint of Radius of Curvature Only

Pythagorean relation for right triangle OAB shows

$$\overline{AB}^2 = \overline{OA}^2 - \overline{OB}^2$$

or $$\left(\frac{L}{2}\right)^2 = (R+D)^2 - (R+D-2C)^2$$

$$= (R+D)^2 - [(R+D)^2 - 4C(R+D) + 4C^2]$$

$$= 4C(R+D) - 4C^2$$

or $$L^2 = 16\,[C(R+D) - C^2]$$

Then $$L = 4\sqrt{C(R+D)-C} \qquad (3.2)$$

where L is the maximum vehicle length under the constraint of radius of curvature only

C is the normal radial clearance between the vehicle and the guideway while the vehicle is operating in a straight line

R is the radius of curvature

D is the diameter of the guideway

3.6. Vehicle and Guideway Characteristics of the New Proposed System

The system being proposed in this thesis uses an air bearing type suspension. The development of a governing equation for the air bearing is presented in the next chapter. However, the major requirement of the air bearing is to have very smooth surfaces on the mating bearing and guideway interfaces. In order to achieve a smooth surface by using a relatively simple vehicle geometry, the vehicle and guideway cross sections shown in Figure 3 are proposed. This shape is essentially a circle having a filled in segment. The flat portion will be used to support the air bearing. The clearance between the vehicle and guideway will be one inch around the circular section.

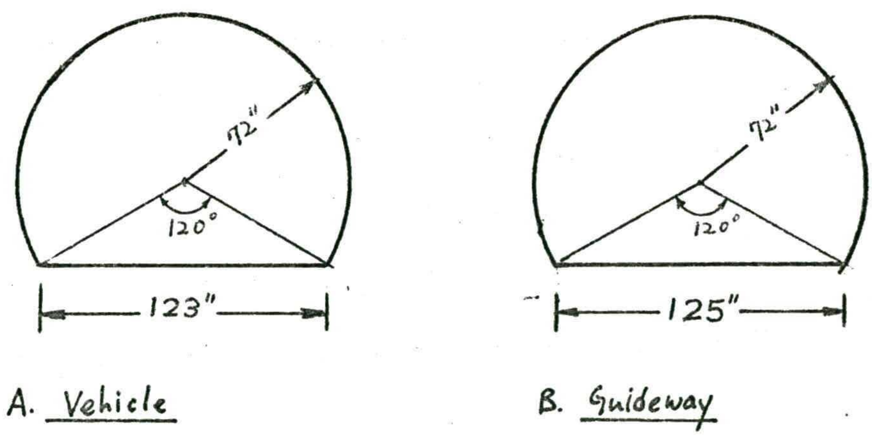

Figure 3. Configurations and Dimensions of the Vehicle and the Guideway

A tube diameter of 12 feet is proposed. This diameter was chosen such that adequate passenger facilities could be included in the vehicle. The 12 foot diameter is feasible for excavation, but could be quite costly. The total cross sectional area of the vehicle will then be 88.5 square feet, or 12,750 square inches.

The minimum radius of curvature of the corners is determined by setting a maximum allowable centrifugal acceleration. For an allowable acceleration of .1 g the minimum radius will be 60,000 ft., or approximately 11 miles. For a one inch clearance between the vehicle and walls, this would allow the vehicle's length to reach 650 feet before it wedges itself in the curve. This length is certainly greater than any length anticipated for a tube transit vehicle. In choosing a length for the vehicle it will be assumed that its capacity will be 100 passengers. For the diameter tube used, this would require a length of approximately 60 feet. A typical vehicle of this length weighs approximately 100,000 lbs.

Chapter 4. Suspension

4.1. Introduction

Three design alternatives have been proposed for suspensions of tube transit systems. These three suspensions utilize: (1) fluid, (2) electromagnetic, and (3) mechanical coupling between the vehicle and the ground based track or guideway. Among these, only the mechanical suspension provides material contact between the vehicle and roadbed; present railroads are a prime example of this type. The fluid suspension must have equipment such as a compressor in order to supply pressurized air to form an air cushion for supporting the vehicle. The electromagnetic suspension uses the repulsive forces of electromagnets to support the weight of the vehicle above the guideway.

The design criteria for the suspension and some detailed individual discussions of these three suspension systems are presented in the following sections. Also, this study makes a proposal utilizing an air bearing for the suspension. The pressure distribution and friction force acting on the vehicle are investigated for the air bearing suspension.

4.2. Design Criteria

The major purposes of the suspension are to provide a comfortable ride for the passengers and to reduce the friction force. As previously mentioned, the comfort criteria for the suspension system must be considered together with its interactions with the guideway, guideway supports, secondary vehicle suspension, and cushions of the seat, since these all constitute parts of the complete suspension system supporting the passenger. The comfort criteria include vertical and lateral vibration within comfort limits for passengers, the control of vertical and lateral displacements, track or guideway alignment, track surface smoothness, the smoothness of welded or expansion joints, surface smoothness of wheels, and any combinations. Other criteria which must be considered for the suspension system are vehicle speed, lateral force imparted to the rails, and wheel adhesion.

All of these considerations must be included in a study of the dynamics of the total vehicle system. To do this, actual data on the individual components would be required.

4.3. Mechanical Suspension

Three similar types of mechanical suspensions have been proposed for tube vehicles. They are steel wheels on rails, rubber tires on concrete, and plastic rimmed steel wheels on concrete. Their individual characteristics are discussed in this section.

For a suspension with steel wheels on rails, the rails must be smooth in both the vertical and lateral directions in order to provide a comfortable ride. Also the wheels must be equally smooth and very concentric about the neutral axis. Therefore, the careful analysis of steel wheels and rails must be made before designing such a suspension system. Much work on this type of suspension has been performed by the railway industry for lower speeds (less than 100 mph). Some higher speed wheel-rail systems have been built in the past decade. These include the Japanese New Tokaido Line, General 70-E powered truck designed by the General Steel Industry, the Budd Company Pioneer III track designed for the Chicago Transit Authority, and the San Francisco BART. However, the fastest speed attained by any of the operating systems is 130 mph, by the Japanese New Tokaido Line. However, the Office of High Speed Ground Transportation[1] has mentioned analyses which claim that it is possible to construct rail surfaces sufficiently smooth to permit wheel-rail suspensions to travel up to 450 mph. No further information on this study could be obtained, but if it is indeed the case, this system would offer advantages of not being affected by the air pressure in the tube and flanged wheels being used for lateral support.

Compared with steel wheels, rubber tires have shorter life and lower load-handling capabilities and require additional wheel sets for each vehicle. Also the rolling resistance of rubber tires on smooth concrete is 9 to 10 times that of steel wheels on steel rails, with speeds up to 150 mph.[22] Furthermore, rubber material failure caused by high temperature limits the tire life at higher speeds. Rubber tires offer advantages of being softer and quieter and providing better traction than steel wheels. However, the large temperature rise developed at high speeds overrides the advantages of the rubber wheels and makes it unfeasible for use on high speed tube transit vehicles.

As for plastic rimmed steel wheels on concrete, they are designed to have the advantages of steel wheels on steel rails, that is, less rolling friction and longer life, and the advantages of tires on concrete, that is, quieter and better traction. Designs of this sort have not yet proven successful, largely because of fatigue and wear problems in the plastic materials.

4.4. Electromagnetic Suspension

The basic concept of the electromagnetic suspension is not new, but many of the techniques necessary to economically support a large vehicle are rather unique. The basic concept is that superconducting solenoids in the vehicle interact with non-superconducting coils imbedded in the ground; when the vehicle moves over the track coils, currents are induced in the coils to generate a repulsive force between the vehicle and the guideway. This suspension would eliminate the conventional vehicle-guideway friction. Such systems have been considered for use in France and for a hypervelocity rocket sled.(23,24,25) Powell and Danby have proposed a Null Flux Magnetic Suspension.(1) Recently the Germans have developed and built an experimental linear-induction-motor car which rides on a magnetic suspension constructed with eight levitation and four guidance magnets.(7)

One problem with this scheme is that it utilizes superconducting magnets which must operate at a very low temperature (about 4°K)(26). The cryogenic refrigeration equipment required to maintain this operation temperature adds much weight to the vehicle. Also, present materials which are available for thermally insulating the cryogenic system are structurally weak and could present problems. Furthermore, electromagnetic suspensions are inherently unstable at higher speeds. For the time being, the components of this suspension are too large and complex to be packed into the available volume of a tube transit system, which is limited by the present technology for excavating tunnels. Once these problems are solved, the magnetic suspension could come to serve the tube transit system.

4.5. Fluid Suspension

Several kinds of air cushion suspensions have been developed for high speed ground transportation. These are the peripheral jet, labyrinth seal, diffuser, plenum chamber, solid wall, air bearing, and ram wing[27]. The tubeflight project has shown that a large-clearance fluid suspension device, such as the peripheral jet or plenum chamber, could be capable of providing a satisfactory solution to the problem of tube vehicle suspension at higher speeds. Therefore, the peripheral jet and plenum chamber will be discussed.

In a peripheral jet air cushion, a jet of air is blown through one or more nozzles, or slots, located on the bottom of the vehicle to form a curtain of pressurized air in the gap between the vehicle and the guideway. This pressurized air is much higher than the ambient pressure in the tube and lifts the vehicle. A vehicle utilizing the peripheral jet for suspension must be equipped with a power source to pump the air flow downward to the gap. The Tubeflight vehicle is suspended by a peripheral jet at low speeds and suspended by a ram wing at high speeds.

In a plenum chamber, air is pumped into the cavity under the vehicle and then leaks out through a rather narrow gap between the periphery of the vehicle and the ground. The volume of this air cavity is larger than that of the air-film of the peripheral jet, but the pressure in the chamber is lower than that under the peripheral jet when supporting a vehicle having a similar weight.

Recently air cushion suspensions have been extensively adapted in designing the tracked air cushion vehicle (TACV). The French Aerotrain, using a flexible skirt, open plenum air cushion system for vertical and lateral support; the British Tracked Hovercraft, using peripheral jet pads providing vertical and lateral support; and the General Motor's Hovair Vehicle, employing a series of flexible-plenum air cushion pads, are the typical tracked air cushion vehicles.

4.6. Air-Bearing Suspension

4.6.1. Air Bearing

Generally, the air bearing when used as a suspension system for a high speed vehicle consists of a flat plate attached to the bottom of the vehicle, with a hole in its center through which pressurized air is forced into a very narrow gap between this plate and the supporting surface on the guideway. If the distance between the bearing and the supporting surface is small enough (typical values range from 0.05 to 0.5mm) and by providing adequate air pressure (typical supply pressures vary from 1 to 6 atm), the air viscosity produces a very high resistance to the air flowing out through the gap and thus creates a very thin high-pressure air film in this narrow gap[27]. This provides an almost ideal lubrication between the vehicle and the guideway. The only requirement for the air bearing is that both surfaces must be very smooth.

Because the gap between the vehicle and the guideway is very small for an air-bearing, the air consumption for suspension is substantially reduced when compared with other air cushion suspensions. This low air usage is essential if the tube is evacuated or partially evacuated, since it will reduce the air storage necessary for suspension and thus will reduce the weight of the vehicle. This also makes the power required for pumping air for suspension smaller and thus reduces operating costs. The surface of the guideway must be as smooth as possible in order to provide a comfortable ride.

4.6.2. Governing Equations for the Air-Bearing Suspension

For simplicity, this analysis assumes that the vehicle is moving forward at a constant velocity. The coordinates to be used are shown in Figure 4.

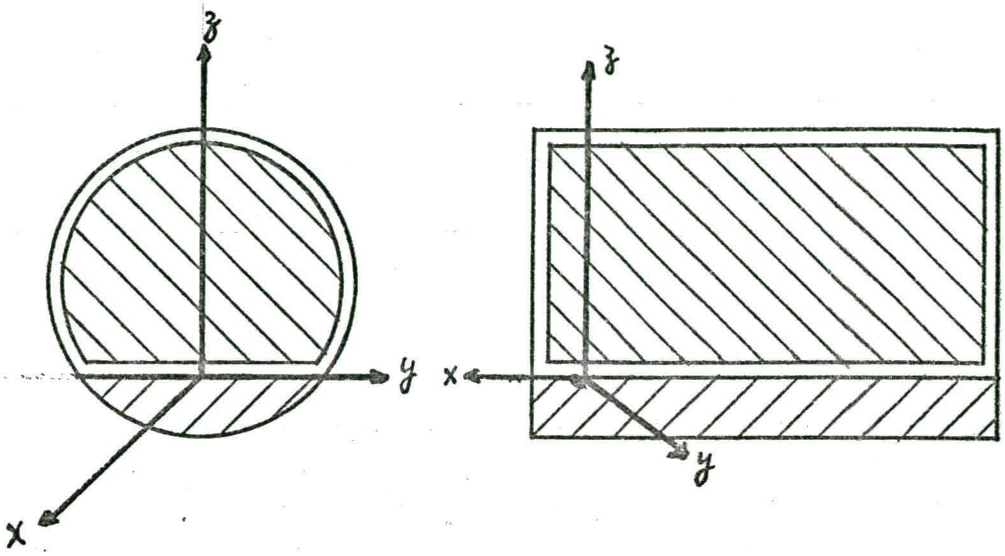

Figure 4. Coordinates for Air Bearing Suspension

Assumptions: (1) Both the surface of the air-bearing plate and the supporting surface of the guideway must be very smooth.

(2) There is a constant gap between the air bearing and the supporting surface of the guideway. Actually, the air bearing causes vibration of the vehicle in the z direction, because the air bearing acts essentially as a spring. Compared with the forward motion of the vehicle, the z direction motion is assumed to be insignificant for the steady state operation. However, in evaluating the dynamics of this suspension, the vibration in z direction is very important and must be considered.

(3) The air bearing film is nearly isothermal. Tipei[28] has stated, "Experimental data on air bearings show that there is little difference in the temperature of the lubricating film and the environment (the differences do not exceed 5°C)."

(4) The absolute viscosity of air in the air bearing film is nearly constant. Tipei[28] has stated, "When gases are used as lubricants, there is negligible variation of viscosity with either temperature or pressure." The viscosity of gas depends on temperature and, to a lesser extent, on pressure. Also gas films are usually assumed to be isothermal. The absolute viscosity of air in the turbulent air bearing film is also nearly constant. Tipei[28] has also stated, "As in laminar gas films, the viscosity is effectively constant in turbulent gas films."

(5) The air flow within the air bearing film is three-dimensional. Because this air bearing has a finite width in the y direction, the air flow cannot be considered as two-dimensional.

(6) The air flow within the air bearing film has a Reynolds number of 4440 (see Appendix A) which lies in turbulent flow region while the vehicle moves at 300 mph velocity. So it is feasible to use tubulent flow equations and properties. However, at start up

and slow down, this assumption may not be totally valid since the Reynolds number will be significantly less placing air flow in the laminar region.

(7) The flow velocity component along the z direction can be neglected compared with the flow velocity components along x and y directions.

(8) The vehicle moves at constant speed.

(9) Steady state operation of the vehicle.

Equations of motion for a turbulent lubricating film are:[28]

$$\frac{\partial \bar{P}}{\partial x} = \mu \frac{\partial^2 \bar{v}_x}{\partial z^2} + \frac{\partial}{\partial z}\left(-\rho\, \overline{v_x' v_z'}\right) \qquad (4.1)$$

$$\frac{\partial \bar{P}}{\partial y} = \mu \frac{\partial^2 \bar{v}_y}{\partial y^2} + \frac{\partial}{\partial z}\left(-\rho\, \overline{v_y' v_z'}\right) \qquad (4.2)$$

$$\frac{\partial \bar{P}}{\partial z} = \frac{\partial}{\partial z}\left(-\rho\, \overline{v_z'^2}\right) \qquad (4.3)$$

where x is the coordinate along the direction of vehicle motion

y is the coordinate normal to Oxz

z is the coordinate along the normal to the supporting surface

$\bar{v}_x$ is the mean flow velocity along the x direction

$\bar{v}_y$ is the mean flow velocity along the y direction

$\bar{v}_z$ is the mean flow velocity along the z direction

v_x' is the flow velocity pulsation along the x direction

v_y' is the flow velocity pulsation along the y direction

v_z' is the flow velocity pulsation along the z direction

P is the pressure of air flow

ρ is the density of air flow

μ is the absolute viscosity of air flow

By utilizing the Prandtle mixing length concept, from which

$$-\rho \overline{v_i' v_z'} = \rho \ell^{*2} \frac{\partial \bar{v}_i}{\partial z} \left| \frac{\partial \bar{v}_i}{\partial z} \right| \quad (4.4)$$

where ℓ^* is the Prandtle mixing length

$\bar{v}_i$ is the mean flow velocity along the i direction

v_i' is the flow velocity pulsation along the i direction

Thus equation of motion is deduced as follows:(28)

$$\rho \ell^{*2} \frac{\partial \bar{v}_i}{\partial z} \left| \frac{\partial \bar{v}_i}{\partial z} \right| + \mu \frac{\partial \bar{v}_i}{\partial z} - \frac{\partial \bar{P}}{\partial x_i} z + C_i(x_i) = 0 \quad (4.5)$$

where x_i represents the x, y, and z coordinates individually

C_i is the integration constant

Equation can establish the velocity distribution as follows:(28)

$$\frac{\partial \bar{v}_i}{\partial z} = \frac{-\mu \pm \sqrt{\mu^2 \mp 4\rho \ell^{*2} [C_i - (\frac{\partial \bar{P}}{\partial x_i}) z]}}{\pm 2 \rho \ell^{*2}} \quad (4.6)$$

And the Reynolds equation for a turbulent film is:(28)

$$\frac{\partial}{\partial x}\left(\frac{h^3 \rho}{\mu k_x} \frac{\partial \bar{P}}{\partial x}\right) + \frac{\partial}{\partial y}\left(\frac{h^3 \rho}{\mu k_y} \frac{\partial \bar{P}}{\partial y}\right) = \rho (U_{vz} - U_{gz}) +$$

$$\frac{1}{2} \frac{\partial}{\partial x}[\rho h (U_{gx} + U_{vx})] - \rho \left(U_{vx} \frac{\partial h}{\partial x} + U_{vy} \frac{\partial h}{\partial y}\right) + h \frac{\partial \rho}{\partial t} \quad (4.7)$$

where h is the air bearing film thickness or the gap between the air bearing the the supporting surface of the guideway

U_{vx} is the velocity component of the vehicle along the x direction

U_{vy} is the velocity component of the vehicle along the y direction

U_{vz} is the velocity component of the vehicle along the z direction

U_{gx} is the velocity component of the guideway along the x direction

U_{gz} is the velocity component of the guideway along the z direction

and $$k_x = 12 + 0.14\left(\frac{\sigma^{*2}}{0.16} Re\right)^{0.725} \tag{4.8}$$

$$k_y = 12 + 0.0897\left(\frac{\sigma^{*2}}{0.16} Re\right)^{0.65} \tag{4.9}$$

where Re is the Reynolds number of air bearing

and $$\left(\frac{\partial \ell^*}{\partial z}\right)_{\substack{z=0 \\ z=h}} = \pm \sigma^* \tag{4.10}$$

For a stationary guideway, $U_{gx} = 0$ and $U_{gz} = 0$

From assumption 2, h=constant

Thus $\frac{\partial h}{\partial x} = 0$ and $\frac{\partial h}{\partial y} = 0$

From assumption 2, $U_{vz} = 0$

From assumption 4, μ = constant

From assumption 9, $\frac{\partial \rho}{\partial t} = 0$

Therefore equation (4.7) is reduced to:

$$\frac{h^3}{\mu}\frac{\partial}{\partial x}\left(\frac{\rho}{k_x}\frac{\partial \bar{p}}{\partial x}\right) + \frac{h^3}{\mu}\frac{\partial}{\partial y}\left(\frac{\rho}{k_y}\frac{\partial \bar{p}}{\partial y}\right) = \frac{h}{2}\frac{\partial}{\partial x}(\rho\, U_{vx}) \qquad (4.11)$$

From assumption 8, U_{vx}= constant

Thus equation (4.11) is reduced to:

$$\frac{2h^2}{\mu U_{vx}}\left[\frac{\partial}{\partial x}\left(\frac{\rho}{k_x}\frac{\partial \bar{p}}{\partial x}\right) + \frac{\partial}{\partial y}\left(\frac{\rho}{k_y}\frac{\partial \bar{p}}{\partial y}\right)\right] = \frac{\partial \rho}{\partial x} \qquad (4.12)$$

From assumption 3, the air bearing film is isothermal

$$\frac{P}{\rho} = \text{constant} \qquad \text{or} \qquad \rho = K P$$

where K is a proportionality constant

Thus equation (4.11) becomes

$$\frac{2h^2}{\mu U_{vx}}\left[\frac{\partial}{\partial x}\left(\frac{KP}{k_x}\frac{\partial \bar{P}}{\partial x}\right) + \frac{\partial}{\partial y}\left(\frac{kP}{k_y}\frac{\partial \bar{P}}{\partial y}\right)\right] = \frac{\partial (kP)}{\partial x}$$

$$\frac{2h^2}{\mu U_{vx}}\left[\frac{\partial}{\partial x}\left(\frac{P}{k_x}\frac{\partial \bar{P}}{\partial x}\right) + \frac{\partial}{\partial y}\left(\frac{P}{k_y}\frac{\partial \bar{P}}{\partial y}\right)\right] = \frac{\partial P}{\partial x} \qquad (4.13)$$

This is a nonlinear partial differential equation because of nonlinear terms $\frac{P}{k_x}\frac{\partial \bar{P}}{\partial x}$ and $\frac{P}{k_y}\frac{\partial \bar{P}}{\partial y}$. No straightforward methods were found to get the closed form solution of this equation and no solutions were found in the literature. But it may be possible to obtain a solution by using the digital computer. Due to the complexity of the equation and the lack of experimental data to check out the results, this equation will not be solved here. Perhaps the best means to obtain a solution would be to perform experiments to get the pressure distribution curve in the air bearing. Then a curve fitting method could be applied to get the approximate solution. These were not done here, but are suggested for future studies.

The future analysis will be based on the assumptions that the gap distance is 0.02 inches and the air flow in the air bearing is turbulent. Since the air pressure in the air bearing film ranges from supply pressure at the entrance of air to ambient pressure at the edge of the air bearing plate, the air bearing film is assumed to be a whole volume having a constant average pressure throughout. Since the air bearing film is isothermal and its temperature is near the ambient temperature of the guideway, the temperature of the air bearing is assumed to be the same as that of the guideway, 70°F. The ambient pressure of the guideway is assumed to be 1 psia, the supply pressure is suggested as 21 psia, and the average pressure is assumed to be one-fourth of the difference between this supply pressure and ambient pressure above the ambient pressure. This fraction would be one-half if the flow were laminar, since a linear pressure drop would prevail. However, for turbulent flow, the rate of pressure drop will be greater near the center of the bearing, thus resulting in a lower average pressure. If the pressure distribution were a cubic function of the distance from the center of the bearing, the average pressure would be one-fourth of the pressure difference between supply and ambient above the ambient pressure. Therefore, the average pressure for air bearing film is 6 psia, which is 5 psi above ambient pressure. In order to support the weight of the vehicle, which is assumed to be 100,000 pounds, a total area of 20,000 square inches of air bearing plate is required. The width of the vehicle is 123 inches, so the length of plate must be 162.3 inches. Now assume the total length of air bearing plate is 168 inches. Since the vehicle is much longer than 168 inches, it will be necessary to use at least two separate bearing pads for support. Assuming the use of two pads, one at the front and one near the rear of the vehicle, the total length may be equally divided, giving each pad a length of 84 inches. The total air bearing scheme is shown in Figure 5, on the next page.

4.6.3. Friction Force Results from the Air Bearing

In the analysis of the propulsion system in Chapter 5, knowledge of the friction force due to the suspension system will be required. This section investigates two approaches to calculate the friction force resulting from the air bearing while operating in steady state. They are the shear stress approach and the pipe flow approach.

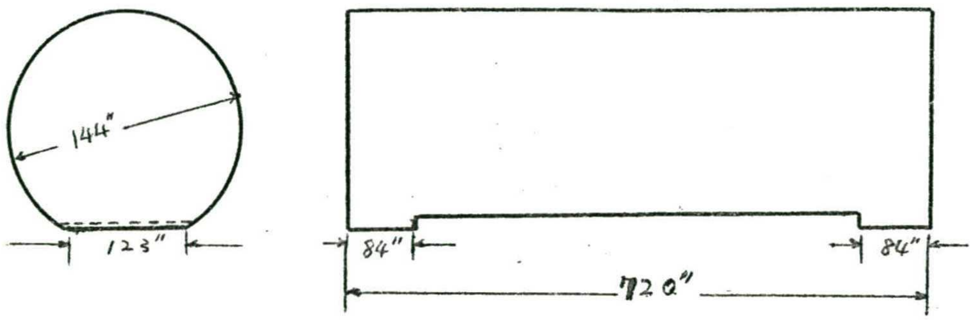

Figure 5. Air Bearing Scheme

4.6.3.1. Shear Stress Approach

Air bearing flow will be assumed to be turbulent. The total shear stress due to the turbulent motion of compressible viscous gas is

$$\tau_{ij} = \tau_{ji} = \mu \left(\frac{\partial \bar{v}_i}{\partial x_j} + \frac{\partial \bar{v}_j}{\partial x_i} \right) - \rho \, \overline{v_i' v_j'} \qquad (4.14)$$

where τ_{ij} is the shear stress acting on the plane which is perpendicular to i axis and in the direction of j axis

$\bar{v}_i$ is the mean flow velocity along x_i direction

and $\mu(\frac{\partial \bar{v}_i}{\partial x_j} + \frac{\partial \bar{v}_j}{\partial x_i})$ is the viscous shear stress term in laminar flow

$-\rho\overline{v_i' v_j'}$ is the apparent or virtual stress of turbulent flow or Reynolds stress

Therefore, the shear stress acting on the vehicle in the opposite direction of the vehicle motion due to this air bearing is

$$\tau_{zx}\Big|_{z=h} = \left[\mu\left(\frac{\partial \bar{v}_x}{\partial z} + \frac{\partial \bar{v}_z}{\partial x}\right) - \rho\overline{v_x' v_y'}\right]_{z=h} \quad (4.15)$$

Because Schlichting[30] has stated, "Prandtle's equation has been successfully applied to the study of turbulent motion along walls (pipe, channel, plate) and to the problem of so-called free turbulent flow." Therefore to solve for this Reynolds stress, it is necessary to introduce Prandtle's mixing-length hypothesis. This hypothesis results in Prandtle's equation which is very useful in the calculation of turbulent flow and is shown below:

$$\tau_t = -\rho\overline{v_x' v_y'} = \rho \ell^{*2} \frac{\partial \bar{v}_x}{\partial z}\left|\frac{\partial \bar{v}_x}{\partial z}\right| \quad (4.16)$$

where τ_t is the Reynolds stress

Substituting equation (4.16) into equation (4.15), equation (4.15) becomes:

$$\tau_{zx}\Big|_{z=h} = \left[\mu\left(\frac{\partial \bar{v}_x}{\partial z} - \frac{\partial \bar{v}_z}{\partial x}\right) - \rho \ell^{*2} \frac{\partial \bar{v}_x}{\partial z}\left|\frac{\partial \bar{v}_x}{\partial z}\right|\right]_{z=h} \quad (4.17)$$

Furthermore, Schlichting[30] has stated, "In numerous cases it is possible to establish a simple relation between the mixing length, 1^*, and a characteristic length of the respective flow. For example, in flows along smooth walls, 1^*, must vanish at the wall itself, because transverse motions are inhibited by its presence. In flows along rough walls the mixing length near the wall must tend to a value of the same order of magnitude as the solid protrusions." Also, Tipei[28] has stated, "The mixing length must vanish on the solid walls." In addition, the air bearing plate is very smooth, consequently the Prandtle's mixing length at the air bearing plate must be equal to zero. Therefore equation (4.17) becomes

$$\tau_{zx}\Big|_{z=h} = \left[\mu\left(\frac{\partial \bar{v}_x}{\partial z} + \frac{\partial \bar{v}_z}{\partial x}\right)\right]_{z=h} \tag{4.18}$$

The total frictional force, F_a, due to the air bearing is found by taking the integral of the shear stress over the air bearing area and is

$$F_a = \iint \tau_{zx}\Big|_{z=h} dx\, dy \tag{4.19}$$

From assumption (7), $\bar{v}_z = 0$

Equation (4.18) becomes

$$\tau_{zx}\Big|_{z=h} = \left[\mu \frac{\partial \bar{v}_x}{\partial z}\right]_{z=h} \tag{4.20}$$

Substituting equation (4.20) into equation (4.19), the desired frictional force is

$$F_a = \iiint \left[\mu \frac{\partial \bar{v}_x}{\partial z} \right]_{z=h} dx\, dy \qquad (4.21)$$

For the proposed system, the desired friction force is

$$F_a = 8 \int_0^{61.5} \int_0^{42} \left[\mu \frac{\partial \bar{v}_x}{\partial z} \right]_{z=h} dx\, dy \qquad (4.22)$$

From equation (4.6), it is known that after solving equation (4.13) and getting the pressure distribution equation, a desired

can then be calculated. Because the pressure distribution equation is not available, it is necessary to assume a $\bar{v}_x$ in order to show how to use this concept.

Assume $z = C\, v_x^2$

where C is an arbitrary constant

Since the boundary condition at $z = h = 0.02''$, $\bar{v}_x = U_{vx} = 300\,\text{mph}$

Then $h = C\, U_{vx}^2$

or $C = \dfrac{h}{U_{vx}^2}$

Then $z = \dfrac{h}{U_{vx}^2}\, v_x^2$

Differentiating with respect to z, $1 = \dfrac{h}{U_{vx}^2} \cdot 2 \cdot v_x \dfrac{\partial v_x}{\partial z}$

or $\dfrac{\partial v_x}{\partial z} = \dfrac{1}{2}\, \dfrac{U_{vx}^2}{h}\, \dfrac{1}{v_x}$

Then $$\left.\frac{\partial v_x}{\partial z}\right|_{z=h} = \frac{1}{2}\,\frac{U_{vx}}{h} \qquad (14.23)$$

Substituting equation (4.23) into equation (4.22), the friction force is

$$F_a = 8\int_0^{61.5}\int_0^{42} \mu\left(\frac{1}{2}\cdot\frac{U_{vx}}{h}\right)dx\,dy$$
$$= 4\mu\,\frac{U_{vx}}{h}\,(42-0)(61.5-0)$$

For air at 70°F, $\mu = 0.026 \times 10^{-7}$ lbf sec/in^2

Then $$F_a = 4 \times 0.026 \times 10^{-7} \times \frac{440.01 \times 12}{0.02} \times 42 \times 61.5$$
$$= 7.08 \text{ lbf}$$

This friction force seems quite small, but since this author could not find any reliable data on friction forcesin air bearings,it is inappropriate to pass judgment on this numerical result at this time. The greatest fault of this analysis is in the use of the assumed velocity profile. Using other profiles could well yield larger friction forces.

4.6.3.2. Pipe Flow Approach

The air flow in an air bearing is considered in a case analogous to flows in pipes. Then the friction force, F_a, is

$$F_a = f\cdot\frac{\rho U_{vx}^2}{2}A \qquad (4.24)$$

where f is the coefficient of friction

A is the area of air bearing plate

Since the air bearing film is assumed to be as a whole volume, the equation of state is

$$P = \rho R T \tag{4.25}$$

Substituting equation (4.25) into equation (4.24), the friction force becomes

$$F_a = f \cdot \frac{U_{vx}^2}{2} \frac{P}{RT} A$$

Since the weight of the vehicle W is equal to PA, then the friction force becomes

$$F_a = f \cdot \frac{U_{vx}^2}{2} \frac{W}{RT} \tag{4.26}$$

From Appendix A, Re = 4440

From friction factor curves on page 184 of Reference 31, the coefficient of friction is taken as 0.044 which corresponds to a smooth pipe.

Then the friction force for the proposed system is

$$F_a = 0.044 \times \frac{(440.01)^2}{2} \times \frac{100,000}{53.35 \times 32.2 \times 530}$$
$$= 102 \text{ lbf}$$

Although this result appears satisfactory, this approach only considers flow in one direction. The fluid motion is actually three-dimensional, so this approach also has some shortcomings. The shear stress approach provides for three-dimensional flow, but it's main drawback is the need to know an accurate velocity profile in order to get an accurate friction force. For this purpose, it is necessary to solve the pressure distribution equation.

4.6.4. Summary

An air bearing may be feasible in view of its low friction force, but the requirement of a very smooth surface and the small gap between two surfaces may cause some difficulties. First of all, it is very difficult to build a guideway having a very smooth supporting surface. Although it is possible, it's cost would be considerable. The small gap requirement would necessitate an exceptional control system to keep the air bearing from contacting the supporting surface. However, further studies must be performed in order to ascertain the feasibility of such a control system.

If further studies show that the air bearing is not feasible, it would be necessary to utilize either a wheeled system or an electromagnetic suspension. At present, wheeled suspensions are not available for operation at the speeds desired for tube transit systems, but this does not mean future systems will not be developed. Electromagnetic suspensions are in the infancy of development and at present have their technological difficulties and are quite costly. However, it is again quite likely that future developments will render it feasible for tube vehicle systems.

Chapter 5. Propulsion

5.1. Introduction

The propulsion system is a primary source for providing the required cruising speed for the tube transit vehicle. Therefore, it is the heart of the tube transit system and must be thoroughly investigated.

Jet propulsion, electric propulsion, gravity propulsion, and pneumatic propulsion have been proposed for tube transit systems. Jet propulsion includes gas-turbine propulsion and hydraulic-jet propulsion. Electric propulsion includes rotary-electric-motor propulsion and linear-induction-motor propulsion. Gravity propulsion can be used alone or used together with other propulsion systems. All of these will be discussed in the following sections. Also the design criteria for propulsion will be discussed.

This study also cites some approaches for analysis of pneumatic propulsion. The characteristics of flow control for pneumatic propulsion are also investigated.

5.2. Design Criteria

Acceleration, deceleration, cruising speed, and braking largely affect the size of the propulsion unit and are therefore the major design criteria for the propulsion system.

The magnitudes of acceleration and deceleration must not exceed the ride comfort criteria for passengers. The Office of High

Speed Ground Transportation has suggested the comfortable acceleration criteria for passengers shown in Table 2.

	Normal	Emergency
Backward longitudinal acceleration, g	0.07	0.30
Forward longitudinal acceleration, g	0.09	0.35
Lateral acceleration, g	0.08	0.20
Vertical acceleration, g	1.0I 0.07	1.0I 0.15

Table 2. Comfortable Acceleration for Passengers

But Beckwith(17) has suggested a maximum steady acceleration of 0.5g and a maximum emergency deceleration of 2g.

The cruising power largely depends upon the aerodynamic drag and the mechanical resistance of the vehicle. Aerodynamic drag will be substantially reduced for an evacuated guideway. This evacuation pressure largely depends upon the capabilities of the equipment used in creating the vacuum.

5.3. Gas-Turbine Propulsion

Gas-turbine propulsion produces thrust by either aerodynamic means such as ducted propellers and turbofans, or via shaft power which provides torque to wheels for traction. Gas-turbine propulsion has been demonstrated on the United Aircraft Corporation Turbo Train operating on the Canadian National Railways and on the Budd railcar, and has been planned for the British Railways Advanced Passenger Train and the French National Railways. The internal propulsion in the tubeflight project is a typical example of gas-turbine propulsion for a tube transit system. The proposed tubeflight vehicle makes use of an on-board flow induction device to transfer air from the front to the rear of the vehicle.

Gas-turbine propulsion can not operate in an evacuated tube because of insufficient oxygen for combustion and because air is necessary to get thrust from propeller type devices. Also, the dissipation of combustion by-products is very expensive and may cause pollution problems. It is difficult to connect several gas turbine propelled vehicles in a train because the preceding vehicle produces air flow disturbances for the next vehicle. But it is still a reasonable choice for atmospheric pressure tube transit systems.

5.4. Hydraulic-Jet Propulsion

Hydraulic-jet propulsion as proposed by Sterling Beckwith[32] utilizes a single row of hydraulic buckets fixed in a straight line along the bottom of the vehicle, as shown in Figure 6. The basic con-

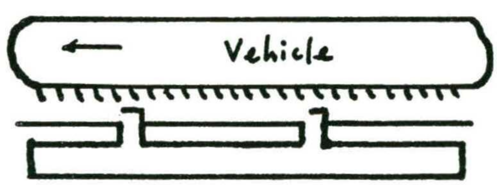

Figure 6 Hydraulic-jet Propulsion

cept is the same as the principle of impulse or Pelton-type water turbines where high pressure water in hydraulic penstocks is converted into rotary motion. Hydraulic jets over which the vehicle passes are aimed in the direction of vehicle travel so that they impinge on these buckets and provide a propulsive force.

For higher efficiency, the speed of water leaving the nozzle must be at least twice the speed of the moving buckets so that when the water flow is reversed by the curvature of the moving bucket the water will have lost all of its motion and kinetic energy in the direction of jet. Thus ,there is a relationship between the vehicle's velocity and the water pressure, which is essentially fixed, so this relationship will be a constant.

There are many advantages for the use of impulse hydraulic turbines for high speed vehicle propulsion. The major elements of a hydraulic-jet propulsion system such as pumps, high-pressure pipes, and hydraulic jet and bucket technology are all available and in wide commercial use for other purposes. Propulsion equipment is not needed on the vehicle. Also, trolleys and third rails for electric propulsion are not needed. Hydraulic-jet propulsion can be used for speeds which are above the traction limit of wheels and it eliminates the small clearance air gaps required for linear induction motor. It is also capable of higher accelerations and decelerations and is much quieter than systems using propellers or jet engines. But any hydraulic jet at high pressure can be as deadly as a third rail in electric propulsion, so that suitable enclosure of the buckets and jet would be essential. The control of bucket direction, noise, and corrosion are the disadvantages of this propulsion.

5.5. Rotary-Electric-Motor Propulsion

This propulsion is similar to the present electric streetcar. The only difference between them is that rotary-electric-motor propulsion for the tube transit system is equipped with a motor that can produce higher speeds. Rotary-electric-motor propulsion using relatively heavy, series-wound dc motors has been demonstrated in the new Penn Central Metroliners, the Japanese New Tokaido Line, and the San Francisco BART. But at higher vehicle cruising speeds, the required power for this type of motor increases as the cube of speed which makes the weight of these motors a critical factor in design. Approaches for reducing the weight of motor include improving the conventional dc motor design or developing other types of motors. Some conventional electric motors, such as the dc commutator motor, the single-phase ac commutator motor, the pulsating dc commutator motor, and the three-phase induction motor, have been suggested for use in design. Several nonconventional electric motors, such as the brushless dc, the thyristor motor, the dc Nadyne, and the R-motor, have also been considered for use in high speed transportation. Since power is applied at the wheels in this type of propulsion, the reduction in wheel-rail adhesion at higher speed may present problems.

5.6. Linear-Induction-Motor Propulsion

The linear induction motor (LIM) has recently been used in high speed vehicles. Since it does not use adhesion and it is non-contacting and non-polluting, it is suitable for high speed vehicles. A linear induction motor is an unrolled rotary induction motor and consists primarily of two stators and a rotor plate. Two air gaps exist between the rotor plate and each stator. Polyphase windings are distributed near the surface of each stator and are capable of providing a magnetic field across the air gap. Because of electromagnetic induction, currents are established inside the rotor plate, and the interaction between the induced currents and the magnetic field results in a force normal to both the field and currents. A typical linear induction motor is shown in Figure 7.

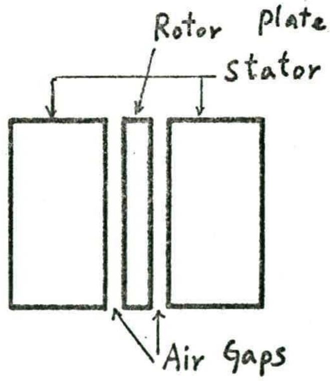

Figure 7 Linear Induction Motor

Two LIM design choices have been proposed for high speed vehicles. These are the vehicle-stator linear induction motor and the guideway-stator linear induction motor. For the first choice, the stator windings are mounted on the vehicle and the rotor plate, also called the reaction rail, is mounted on its edge on the center line of the guideway. This choice requires a power pickup device for the stator winding source and a third rail. For the second choice, the rotor plate is mounted on the vehicle and the stator windings are built into the guideway. Therefore there are no power pickups or third rails for the second choice.

Some advantages offered by the use of linear induction motors for propulsion are listed below:

(1) The propulsion system is quiet and locally nonpolluting.

(2) The traction system is completely independent of adhesion between wheels and rails.

(3) The propulsion is noncontacting because the driving force is developed in the air gap.

(4) The design is not affected by centrifugal force or motor diameter.

(5) While running, the rotor heating is not a problem.

But the high cost of stator coil is a major disadvantage.

5.7. Gravity Propulsion for the Proposed System

The proposed vehicle will be accelerated by a pneumatic force to a speed of 100 mph, it will then be continuously accelerated by gravity along a constant slope to a speed of 300 mph. A sketch of the total path of the vehicle is shown in Figure 8. This gravity propulsion is simply achieved by letting the vehicle move along the downward guideway and is shown in Figure 9. The pneumatic propulsion for the proposed

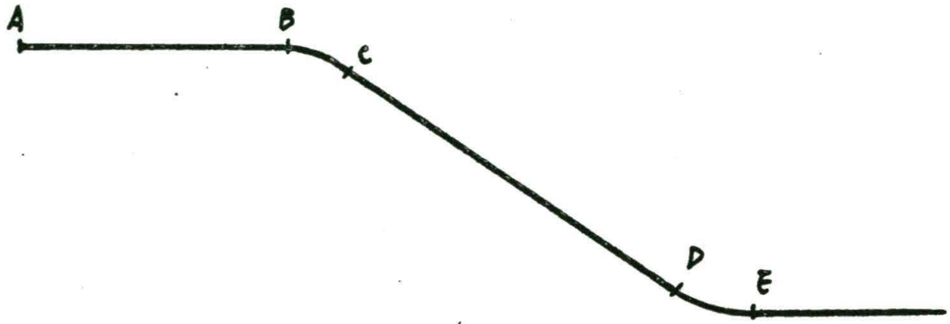

Figure 8. Total Path for the Proposed System

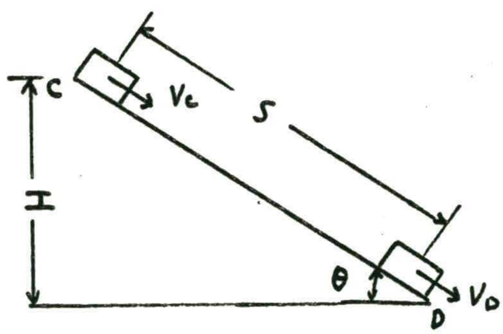

Figure 9. Gravity Propulsion for the Proposed System

system will be developed in the next section.

The relationship between V_C and V_D (developed in Appendix B) is

$$V_D^2 = V_C^2 + 2gH \tag{5.1}$$

or

$$H = \frac{V_D^2 - V_C^2}{2g} \tag{5.2}$$

and

$$H = S \sin\theta \tag{5.3}$$

where V_C is the velocity of the vehicle at the top of the downward guideway

V_D is the velocity of the vehicle at the bottom of the downward guideway.

For the proposed system V_C = 100 mph = 146.67 ft/sec

V_D = 300 mph = 440.01 ft/sec

By applying equation (5.2), H for the proposed system is 2680 feet.

This assumes no vehicle friction losses. The maximum acceleration of 0.5 g proposed by Reference 17 will be used in determining the slope of the incline the vehicle travels on. Here,

$$a = 0.5 \times 32.2 \text{ ft/sec}^2 = g \sin\theta$$

In solving this equation, the length of travel, S, comes out to be 5360 feet. Quite obviously, if smaller acceleration were utilized, the slope of the vehicle would be less and the distance S would be proportionately increased.

5.8. Pneumatic Propulsion

Pneumatic propulsion is to be used to drive the vehicle and will utilize a pressure difference between the front and rear of the vehicle to provide an acceleration. A great advantage of pneumatic propulsion is that there is no propulsion equipment on board and no power pickup devices required. The propulsion unit consists of the guideway, evacuation pumps, and flow control restrictions. Elaborate analyses must be made before designing such a system. A complex analysis of the air flow control system is also necessary for this system. The pneumatic propulsion phase for the proposed system is shown in Figure 10.

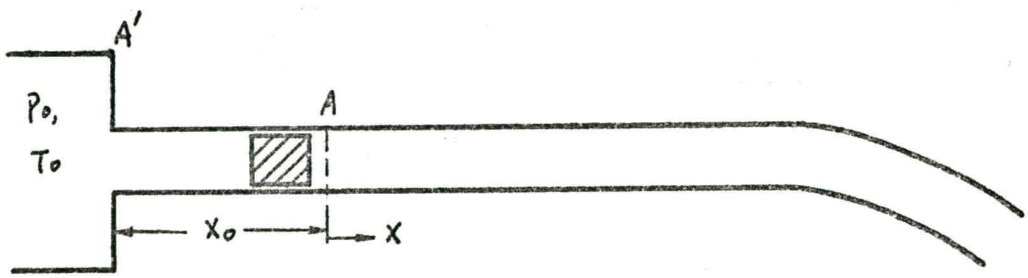

Figure 10. Pneumatic Propulsion Phase for the Proposed System

Major hardware which are treated in this system are a compressed air reservoir, a variable area air flow restriction, a special vehicle loading pad, and vacuum pumps for evacuating the tube. These will now be discussed in more detail.

5.8.1. Air Reservoir

This reservoir will hold 40 psi air which will be used for propelling the vehicle. It will be assumed for purposes of analysis that the air in this reservoir is always at the same pressure. However, in reality, as air is released, this pressure will decrease. To get an idea of the size of this reservoir needed to accelerate a 100,000 pound vehicle to 100 mph at 0.5 g, the tank size would have to be 75,300 ft^3 if a 4 psi drop in the reservoir pressure, Po, is assumed. If this size would prove unfeasible, higher pressures could be used for air storage.

5.8.2. Flow Restriction

There are three types of restrictions which are available for reducing the pressure between the reservoir and the guideway. These are the nozzle, orifice, and valve. These all behave in a similar manner and only vary in the discharge coefficient which is applied to the flow equation. The most efficient of these is the nozzle. However, in this study, the area of the restriction must be variable.

A simple means for getting this variable area is to utilize a conical poppet valve, which is shown in Figure 11.

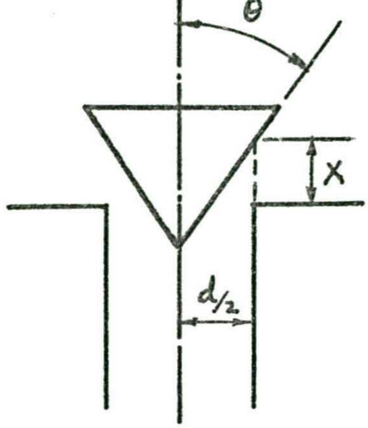

Figure 11. The Conical Poppet Valve

And the flow area here is

$$A_r = \pi X \sin\theta \,(d - X \sin\theta \cos\theta) \qquad (5.4)$$

5.8.3. Evacuation Pumps

Since pneumatic propulsion is to drive the vehicle by utilizing the pressure difference between the front and rear of the vehicle, a reduced pressure is needed at the front of the vehicle in order to reduce the power required for the propulsion system. For the best operating efficiency, the front of the vehicle needs to be evacuated. Therefore, vacuum pumps are needed to accomplish this function. The attainable vacuum pressure depends on the pump capabilities.

Some commercial pumps have been investigated for their feasibilities in this application. A typical commercial two-stage dry air-cooled steam ejector vacuum pump, performance data indicate a vacuum pressure of 29-inch Hg may be obtained when atmospheric pressure is 30-inch Hg. Therefore, the absolute pressure is 1-inch Hg. or

0.491 psi. Furthermore, data were obtained for an air-cooled two-stage vacuum pump which could evacuate to a pressure of 0.3-inch Hg or 0.1473 psi. With further development of the vacuum pumps, it may be possible to get a near zero absolute pressure such as 0.01 psi. However, the amount of horsepower necessary to reach such pressures would be very high.

5.8.4. Pneumatic Propulsion Analysis

5.8.4.1. Constant Pressure Approach Without Feedback Control of the Flow Restriction

In order to provide a comfortable ride for the passengers, it is feasible to design a pneumatic propulsion system which will provide a constant acceleration motion of the vehicle within comfortable acceleration limits. For constant acceleration motion, a constant pneumatic force must be applied behind the vehicle; therefore, it requires constant pressure air behind the vehicle at all times. This analysis will develop a closed form,time solution for the area of the variable restriction which will give the vehicle a constant acceleration at all times. This implies that the pressure behind the vehicle will always be at a constant pressure.

Assumptions which will be used in the analysis are:

(1) The air flow in the guideway behind the vehicle is adiabatic and frictionless, that is, it is isentropic flow.

(2) The air in the guideway behaves as a perfect gas.

(3) The physical properties are always homogeneous throughout the entire guideway behind the vehicle at any moment. This means that the volume behind the vehicle is considered as a whole volume having homogeneous properties, such as pressure or temperature, anywhere at any moment.
Since the flow velocity will only be 100 mph (≈ Mach number = 0.13). This assumption, which essentially says the flow is incompressible, is quite valid.[31]

(4) The air pressure in front of the vehicle is considered to be zero.

(5) The reservoir pressure, Po, is a constant.

The entire system is shown in Figure 12.

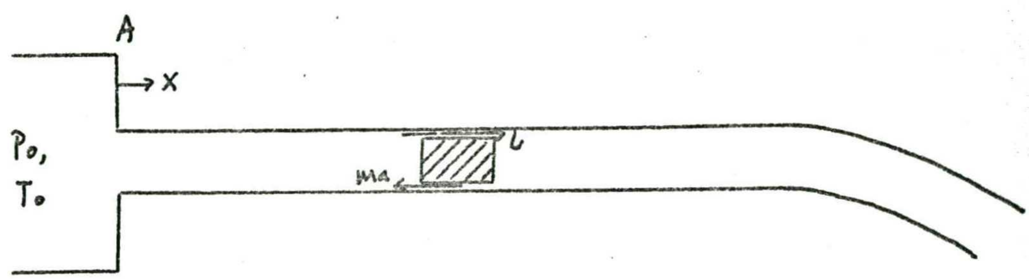

Figure 12. Pneumatic Propulsion for Constant Pressure Approach without Feedback Control of the Flow Restriction

The flow through the restriction at A is

$$\dot{m} = C_w \rho_r A_r U_r \tag{5.5}$$

where Ur is the flow velocity through this restriction

and

$$U_r = \left\{ \frac{2kgRT_0}{k-1} \left[1 - \left(\frac{P}{P_0} \right)^{(k-1)/k} \right] \right\}^{1/2} \tag{5.6}$$

C_w is the discharging coefficient

A_r is the flow restriction area

ρ_r is the density of flow at restriction

The flow leaks around the radial clearance between the vehicle and the guideway are considered as the linear relation

$$L = \rho K (P - 0) dt = \rho K P dt \tag{5.7}$$

where k is the leakage coefficient and L is the leakage mass flow rate.

The mass flowing into the volume behind the vehicle from the air bearing is

$$m_a = \dot{m}_a dt \tag{5.8}$$

where $\dot{m}_a$ is assumed to be a constant.

Then the conservation of mass yields;

$$C_w \rho_r A_r U_r dt + \dot{m}_a dt - \rho K P dt = dm \tag{5.9}$$

The density, ρ , will be constant because the flow pressure is constant and because the flow is assumed to be isentropic, and is therefore also isothermal. Then

$$dm = d(\rho V) = \rho dV + V d\rho = \rho dV = \rho d(A_g x) = \rho A_g dx \quad (5.10)$$

where V is the flow volume behind the vehicle

x is the displacement of the vehicle

A_g is the cross sectional area of the guideway

And also $\rho = \rho_r$

Therefore, equation (5.5) becomes

$$\begin{aligned} \dot{m} &= C_w \rho_r A_r U_r \\ &= C_w \rho A_r U_r \\ &= C_w \rho A_r \left\{ \frac{2kgRT_o}{k-1} \left[1 - \left(\frac{P}{P_o}\right)^{(k-1)/k} \right] \right\}^{1/2} \\ &= C_1 \rho A_r \end{aligned}$$

where $C_1 = C_w \left\{ \frac{2kgRT_o}{k-1} \left[1 - \left(\frac{P}{P_o}\right)^{(k-1)/k} \right] \right\}^{1/2}$

For considering the air in air bearing film as a volume having an average density, ρ , and pressure, P , then $\dot{m}a$ can be changed to $\rho \frac{\dot{m}_a'}{\rho}$

Therefore equation (5.9) becomes

$$C_1 \rho A_r dt + \rho \dot{m}_a' dt - \rho k P dt = \rho A_g dx$$

or $$C_1 A_r + \dot{m}_a' - kP = A_g \frac{dx}{dt} \quad (5.11)$$

The air losses and the friction forces existing between the vehicle and the guideway due to the air bearing film will be considered as an effective damping force proportional to the velocity of the vehicle. Also the friction force due to the mechanical seal between the vehicle and the guide is assumed to be a constant denoted by

By Newton's Law

$$PA_v - C_d \frac{dx}{dt} - f = M_v \frac{d^2x}{dt^2} \tag{5.12}$$

where A_v is the cross sectional area of the vehicle

C_d is the damping coefficient

M_v is the mass of the vehicle

From equation (5.11)

$$X = \frac{C_1 A_r + m q' - kP}{A_g D} \tag{5.13}$$

From equation (5.12)

$$X = \frac{PA_v - f}{M_v D^2 + C_d D} \tag{5.14}$$

where $D = \frac{d}{dt}$

Equality of equations (5.11) and (5.12) yield

$$\frac{C_1 A_r + m q' - kP}{A_g D} = \frac{PA_v - f}{M_v D^2 + C_d D} \tag{5.15}$$

Simplifying, the following equation results:

$$(M_v D + C_d)(C_1 A_r) = P A_v A_g - f A_g \tag{5.16}$$

When vehicle damping is neglected, the equation for the change in area will be simply $A_r = kt$. If a small amount of damping is included, the restriction area will reach some finite steady state value. Since this damping is likely to be very small relative to the forcing function, $P A_v A_g - f A_g$, over the range of speeds from 0 to 100 mph, it can for all practical purposes be neglected.

Equation (5.16) is a linear differential equation of first order. It's solution is

$$A_r = C e^{-\frac{C_d}{M_v} dt} + \frac{M_v (P A_v A_g - f A_g)}{C_1 C_d} \tag{5.17}$$

where C is an arbitrary constant

A major shortcoming of this approach is that it is in essence an open loop control system and changes in vehicle mass, changes in supply pressure and temperature, and valve performance are not automatically taken into account. Therefore, large errors in vehicle speed could occur.

5.8.4.2. Constant Pressure Approach with Feedback Control of the Flow Restriction

In this approach, feedback theory is applied in order to control the acceleration of the vehicle. In performing this analysis it will be assumed that signals proportional to the vehicle's velocity and acceleration are available for use as a feedback signal. The use of feedback should eliminate some of the problems which could occur in the system previously discussed.

The basic approach to be taken in this analysis is to simulate the equations for the process using a Continuous System Modeling Program (CSMP) on the IBM 360-75 computer. The equations which are developed turn out to be very non-linear but may be handled with very little linearization using the computer simulation.

Basic assumptions which will be made in the analysis include:

(1) The same assumption as Section 5.8.4.1. except assumption (a) in that section.

(2) The air flow in the guideway behind the vehicle is isothermal.

(3) The air flowing into the control volume from the air bearing will be neglected as it is very small relative to the flow impart through the valve.

(4) The air loss through the radial clearance between the vehicle and guideway will also be neglected.

This scheme is shown in Figure 13.

Considering the flow through the restriction at A' to be isentropic, the following equation for mass flow rate through the restriction may be obtained.(33)

$$\dot{m} = \frac{A_r P_0}{\sqrt{T_0}} \left\{ \frac{2 k g}{(k-1) R} \left[\left(\frac{P}{P_0}\right)^{2/k} - \left(\frac{P}{P_0}\right)^{(k+1)/k} \right] \right\}^{1/2} \qquad (5.18)$$

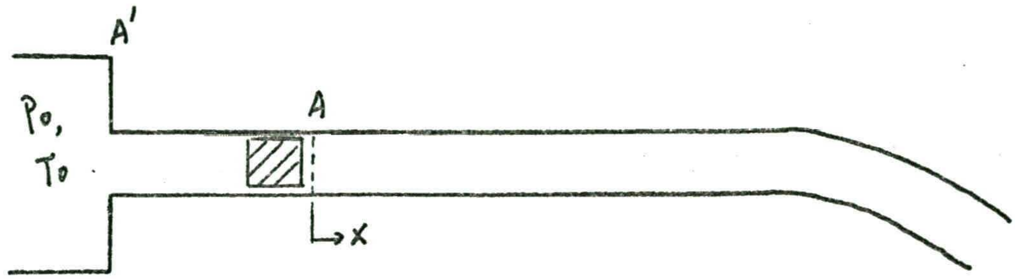

Figure 13. Pneumatic Propulsion for Constant Pressure Approach with Feedback Control of the Flow Restriction

This relation is quite burdensome to apply in its present form since it will be required that P be determined. The feasible way of accurately doing this is an iteration technique. Rather than performing iterations which are very time consuming on the computer, this equation was fitted with the approximate relation (see Appendix C).

$$\dot{m} = 0.2044 A_r + 0.0649 A_r P \qquad (5.19)$$

which is applicable for pressures, P, between 0 and 7 psi with little error.

Also

$$\dot{m} = \frac{dm}{dx} = \frac{d}{dx}(\rho v) = \rho \frac{du}{dx} + v \frac{d\rho}{dx} \qquad (5.20)$$

Substituting the perfect gas relation $\rho = \frac{P}{RT}$ into equation (5.20) yields

$$\dot{m} = \frac{P}{RT}\frac{dV}{dt} + V\frac{d}{dt}\left(\frac{P}{RT}\right) \tag{5.21}$$

Since the flow is isothermal, equation (5.21) becomes

$$\dot{m} = \frac{P}{RT}\frac{dV}{dt} + \frac{V}{RT}\frac{dP}{dt} \tag{5.22}$$

Equating equations (5.19) and (5.22) gives

$$0.2044 A_r + 0.0649 A_r P = \frac{P}{RT}\frac{dV}{dt} + \frac{V}{RT}\frac{dP}{dt} \tag{5.23}$$

Since $\quad V = A_g x$

Therefore equation (5.23) becomes

$$0.2044 A_r + 0.0649 A_r P = \frac{1}{RT}\left[P A_g \frac{dx}{dt} + A_g x \frac{dP}{dt}\right] \tag{5.24}$$

Applying constants so that equation (5.24) has units for P in psi, Ag in square feet, and x in feet gives

$$\frac{144}{RT}\left[P A_g \frac{dx}{dt} + A_g x \frac{dP}{dt}\right] = 0.2044 A_r + 0.0649 A_r P \tag{5.25}$$

$$\text{or} \quad \frac{dP}{dt} = \frac{RT}{144}\left[0.2044 A_r + 0.0649 A_r P\right] - \frac{P}{x}\frac{dx}{dt} \tag{5.26}$$

Since the air losses and air bearing behave as damping, Cd, in this instance, Newton's law for the motion of the vehicle is

The above equations can be combined to give the dynamic characteristics of the open loop system. However, feedback is desired and in this case acceleration feedback would be most desirable. Problems in computer algorithms occurred when using acceleration feedback so in this presentation only velocity feedback will be considered.

Here, $$e = \dot{x}_d - \dot{x}$$

where e is the velocity error and x_d is the desired velocity.

5.8.4.2.1. Integral Controller

When e is a positive number the vehicle must be speeded up and therefore the valve must be opened. Also, when e=0 the valve must have a constant area.

Therefore the relation $\frac{dA_r}{d\dot{x}} = k_i$ applies to the valve. This appears as integral control in the block diagram shown in Figure 14. The X symbols indicate the nonlinear multiplication of variables..

This is a non-linear system, it is impossible to predict it's response by analytical methods. The above system was simulated using the CSMP 11 computer program provided by the Mechanical Engineering Department of the Ohio State University. The results showed that the error between $\dot{X}d$ and $\dot{X}$ was oscillatory, very large, and diverged for a ramp input in velocity (constant acceleration input). This means

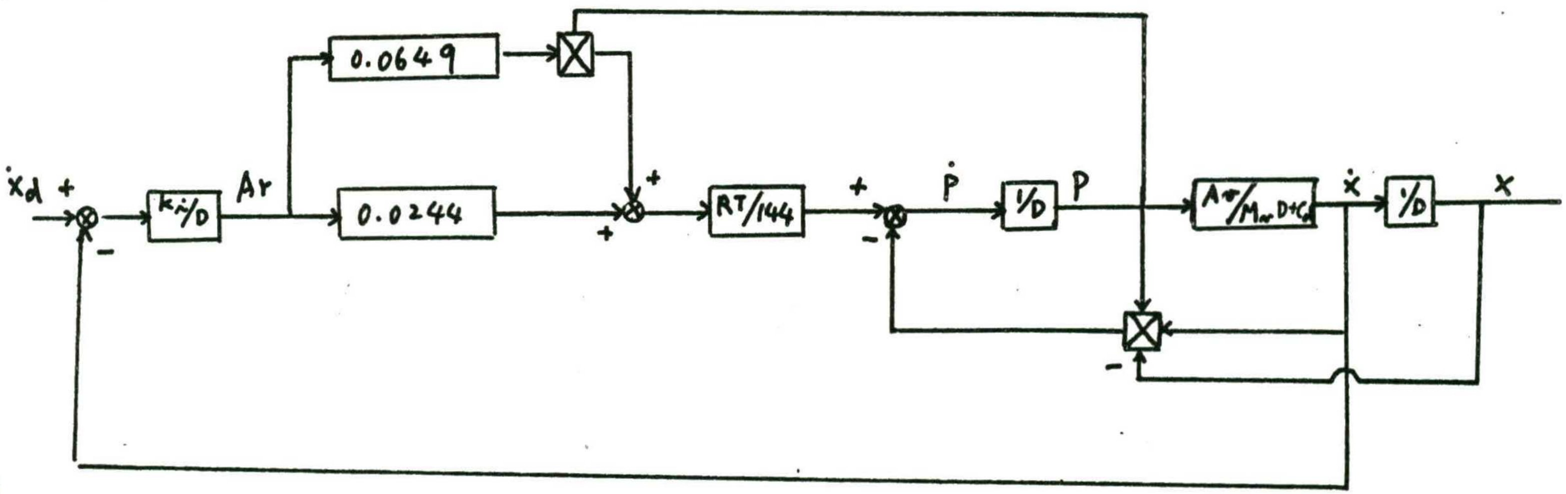

Figure 14. Block Diagram for Integral Controller

that the physical model of the valve as an integral controller is not feasible since the system goes unstable. The ramp input was based on a 0.5 g acceleration.

5.8.4.2.2. Linear Analysis

Since the integral controller resulted in an unstable system, it was necessary to look at other modes of control. In this case, both proportional and derivative controllers were added to the integral term to form a proportional plus integral plus derivative (PID) controller. Also, to predict the general effects each of these modes of control would have on the velocity of the vehicle, the system equations were totally linearized using the Taylor's series approximation.

The non-linear mass flow rate equation (5.18) may be linearized into the form $\dot{m} = C_1 Ar + C_2 P$ (5.27) where the constants are functions of operating points at $\dot{m}_o$, Ar_o, and P_o - all constants.

In linearizing equation (5.22), it will be assumed that the pressure is essentially a constant.

Therefore equation (5.22) becomes

$$\dot{m} = \rho \frac{dV}{dt} = \rho A g \frac{dx}{dt} \qquad (5.28)$$

Equality of equations (5.28) and (5.29) gives

$$C_1 Ar + C_2 P = \rho A g \frac{dx}{dt}$$

$$\text{or } C_2 P = \rho A g \frac{dx}{dt} - C_1 Ar$$

$$P = C_4 Ar - C_3 \frac{dx}{dt}$$

Therefore the entire block diagram is set up as shown in Figure 15. Figure 16 shows a simplified version of this block diagram.

The forward transfer function $G = \dfrac{(K_p + \frac{K_i}{D} + K_d D)\, C_4 A_v}{M_v D + C_d + C_3 A_v}$

$$= \frac{C_4 A_v (K_d D^2 + K_p D + K_i)}{D(M_v D + C_d + C_3 A_v)}$$

$$H = 1$$

The characteristic equation for the system is

$$C_4 A_v (K_d D^2 + K_p D + K_i) + D(M_v D + C_d + C_3 A_v) = 0$$

or

$$(C_4 A_v K_d + M_v) D^2 + (C_4 A_v K_p + C_d + C_3 A_v) D + K_i C_4 A_v = 0$$

This equation is of the form

$$\frac{D^2}{\omega_n^2} + \frac{2\zeta}{\omega_n} D + 1 = 0$$

where ω_n is the natural frequency

ζ is the damping ratio

This equation is second order and will always be stable provided all of the coefficients are positive, which they are in this case.

The effects of the controller modes can be determined from the characteristic equation. Increasing derivative control (K_d) is the same as increasing mass, thus it decreases the system natural frequency. Increasing proportional control (K_p) increases damping, thus reducing oscillation. Increasing integral control (K_i) increases the spring constant, thus increasing W_n and decreasing f. The

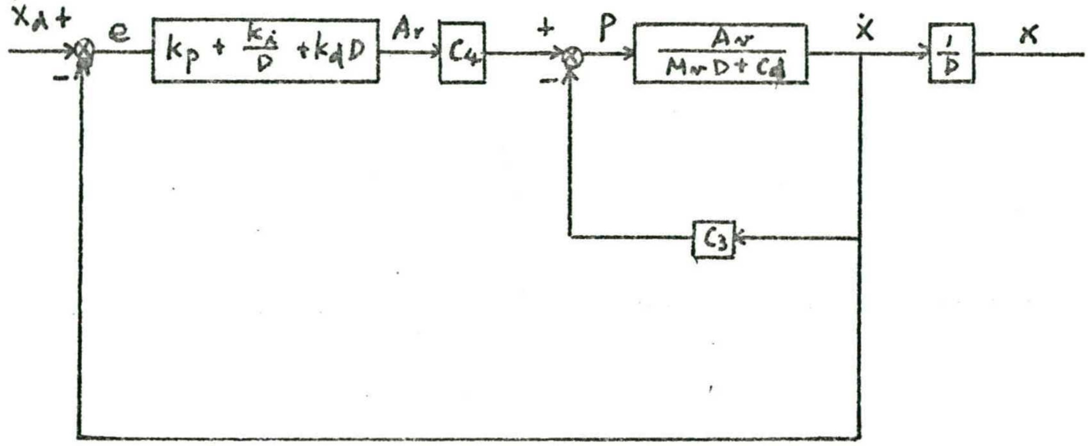

Figure 15. Linear System with a PID Controller

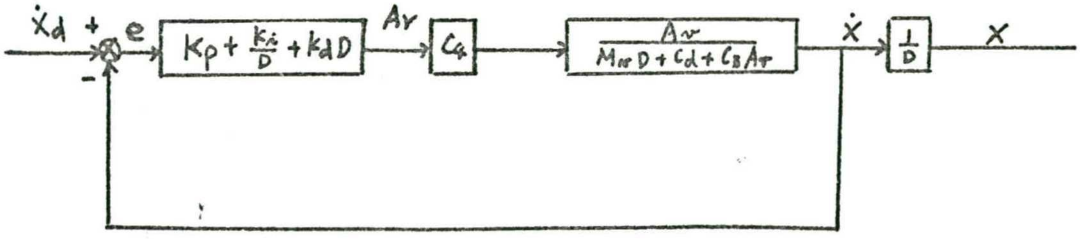

Figure 16. Simplified Version of Figure 15

system is Type 1 and therefore will have a constant steady state error for a ramp input.

5.8.4.2.3. PID Controller

Next, the integral controller in Figure 14 was replaced with a proportional-integral-derivative controller. The computer simulation was again run, this time for various combinations of K_p, K_i, K_d and C_d. Each computation was limited to a ten second duration because this is as long as is needed for the vehicle to reach a speed of 100 mph. Should higher speeds or different accelerations be desired, the program could easily accommodate them.

Figure 17 shows a plot of the error function, $v_d - v$, for the different controller coefficients. It is observed that the error function changes very little with large changes in the damping coefficient, C_d, when K_p, K_i, and K_d are held constant. This is evidenced by comparing Runs 1, 2 and 3.

Runs 2 and 4 show the effects of increasing K_p. It is seen that the frequency of oscillation tends to decrease and the height of the first peak decreases. This would seem to indicate an increase in damping as predicted in the linear analysis. However, the natural frequency change was not predicted. It would be necessary to run the data for a longer time duration to determine in more detail the effect of K_p on the natural frequency.

The effect of increasing K_i is observed in comparing runs 2 and 5. These runs show that as K_i is increased, the frequency of

Figure 17. Velocity Error for Various PID Controllers

Run No.	Kp	Ki	Kd	Cd
1	10	10	10	5
2	10	10	10	20
3	10	10	10	40
4	40	10	10	20
5	10	40	10	20
6	10	10	40	20

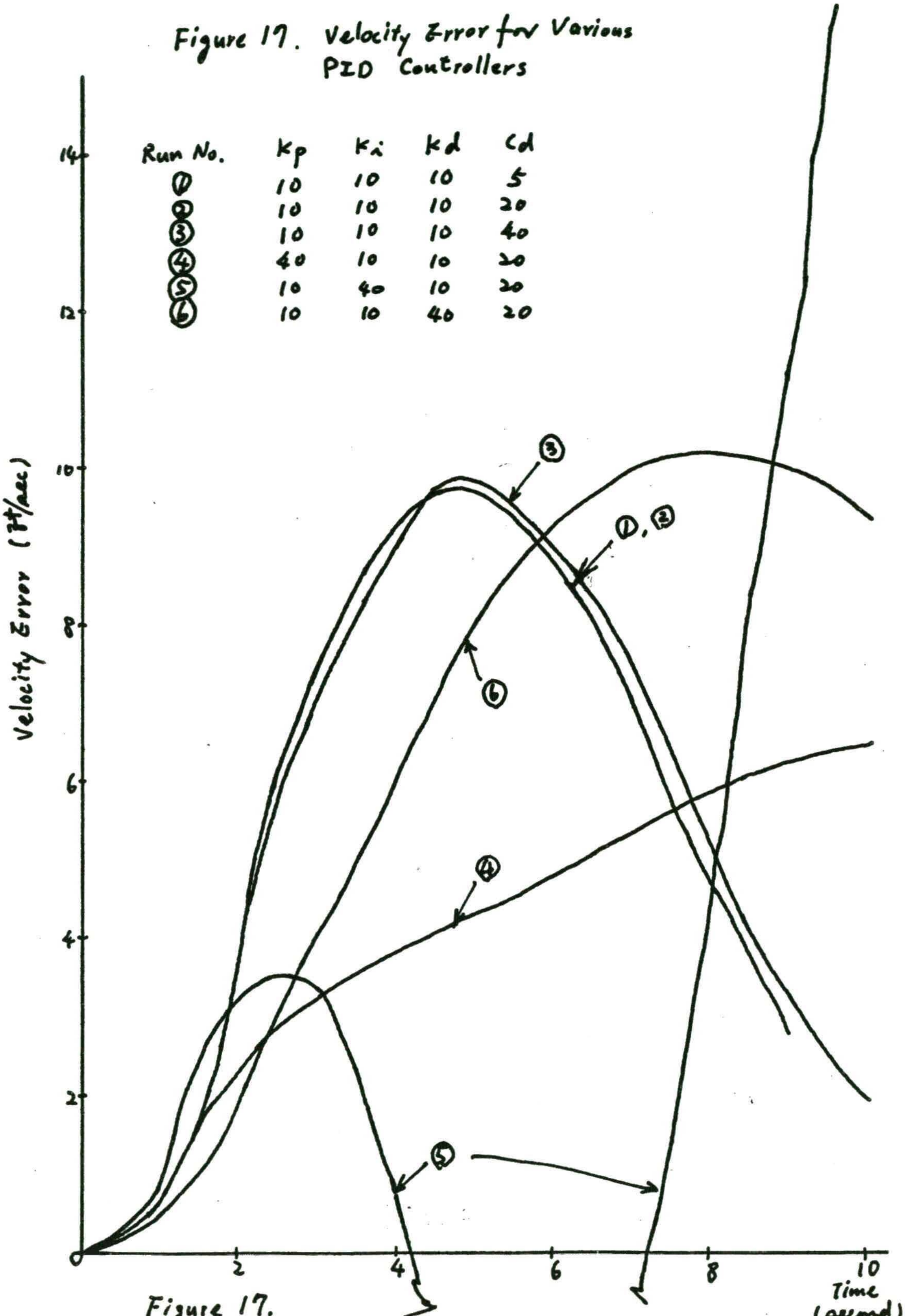

Figure 17.

oscillation becomes much higher and tends to go unstable very rapidly. This agrees well with the linear model. In general, integral control often tends to make systems unstable as it did in this problem.

Finally, a comparison of runs 2 and 6 shows that increasing K_d slightly decreases the damping and decreases the natural frequency. Again this is in accordance with the linear model.

The best of the above runs is rather difficult to determine, but the system having the lowest acceleration error (not shown) over the ten second duration was run 4, which had a large amount of proportional control. It's acceleration error was approximately 10.82% error from the desired value of .5 g.

In order to reduce this acceleration error other combinations of the coefficients were tried. In all subsequent runs C_d was set equal to 0.1, a value more closely related to that of the actual system. The best results were obtained for $K_p = K_i = K_d = 100$, where the maximum acceleration error was only 2.21%. Plots of the velocity error, vehicle acceleration, and area of the restriction are shown in Figure 18. It is interesting to note that Ar changes in a linear fashion which is very close to that predicted by the first method in section 5.8.4.1. One problem which may arise with these large coefficient values is that the hardware used may saturate before it can achieve gains of this order of magnitude.

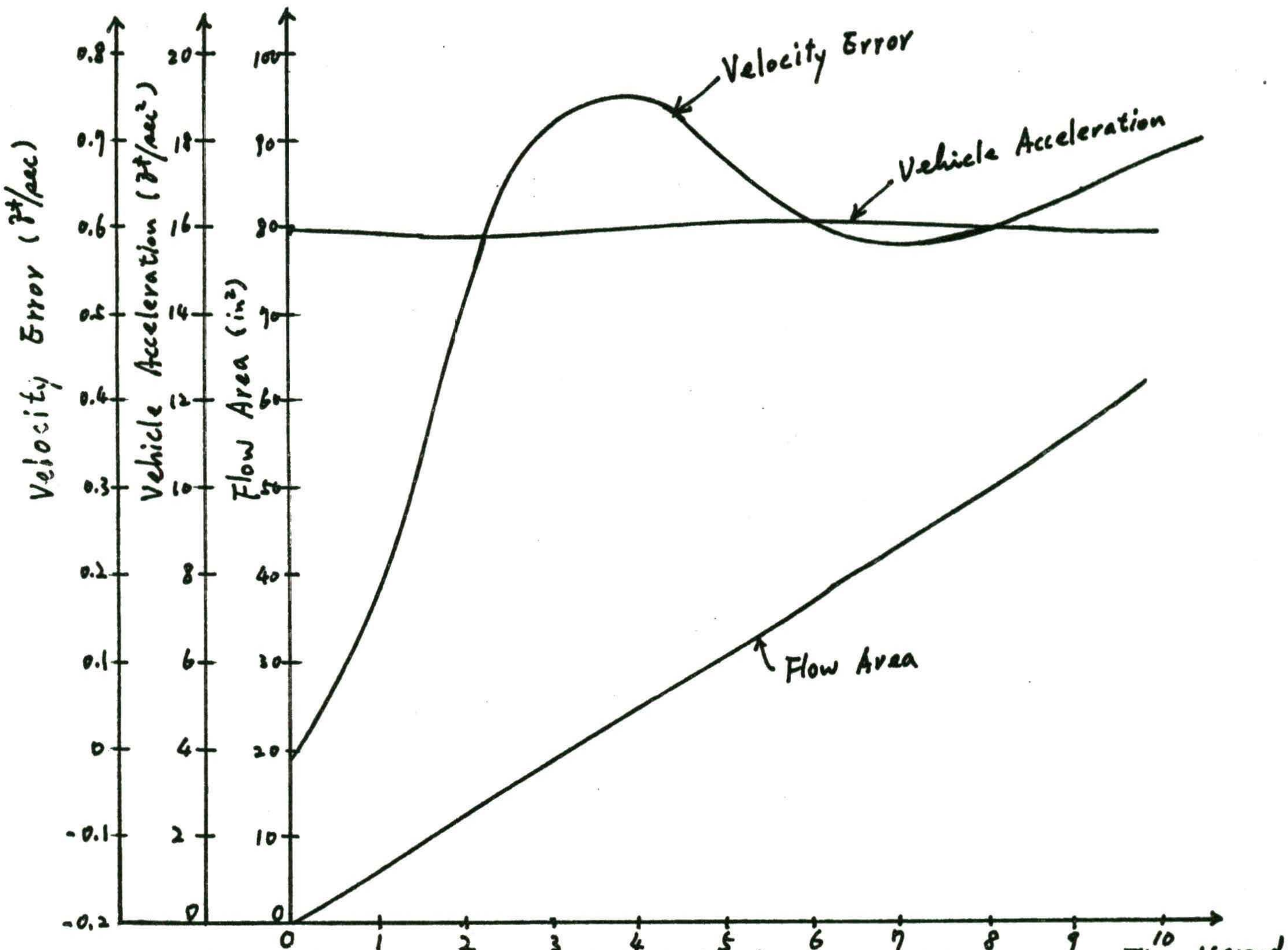

Figure 18. Velocity Error, Vehicle Acceleration, and Flow Area

5.8.4.2.4. Effects of Time Lag

One problem which was not considered in the above analysis is the fact that the variable restriction cannot change position instantaneously. To model the motion of the restriction, a first order lag was used. The resulting block diagram is shown in Figure 19.

As would be expected, the error was large after introducing the time delay. For a time constant τ = .1 sec, the acceleration error was very close to those of run 7. However, for τ = 1.0 sec the error is considerably larger.

5.8.4.2.5. Effects of the Tube Pressure Drop on Equation

Finally, a consideration which is not investigated in detail here, but which should be considered in further investigations, is the effect of pressure drops along the tube on the vehicle's motion. Briefly, the effects of this pressure drop on the linear analysis will be checked out. Figure 20 shows the new physical model. Pi is the pressure at the restriction outlet, and p is the pressure at the

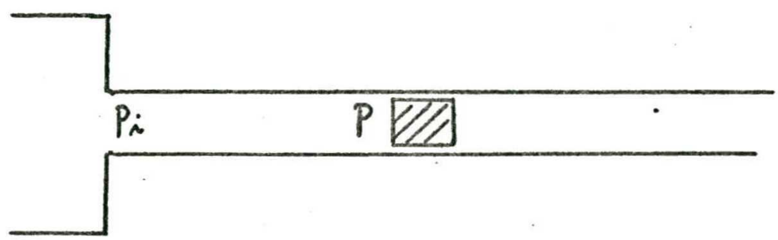

Figure 20. Pneumatic Propulsion with Tube Pressure Drop

vehicle. The difference Pi-P is the pressure drop in the tube. Here, the pressure p is a function of Pi, x, and $\dot{x}$. The linearized relation

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for p can be expressed as $p \approx p_i - k_1 x - k_2 \dot{x}$ where K_1 and K_2 are positive. When this is plugged into the block diagram of Figure 15, a new block diagram is shown in Figure 21. The corresponding characteristic equation is

$$\left(\frac{M_v}{A_v} + C_4 K_d\right) D^2 + \left(\frac{C_d}{A_v} + C_4 k_p + C_3 + k_2\right) D + \left(C_4 k_i + k_1\right) = 0$$

Thus the increase of K_1 causes an increase in natural frequency and the decrease in damping ratio. Therefore, this increases the speed of response. But the increase of K_2 causes the increase of the damping ratio; therefore this decreases the oscillation of the system.

5.8.4.2.6. Other Considerations

After the pneumatic propulsion accelerates the vehicle to 100 mph, gravity takes over to continue accelerating the vehicle to 300 mph. At this point where pneumatic propulsion is no longer used, it would be most efficient to close off the 4 psi air behind the vehicle so that it may be evacuated before the next vehicle leaves. This would prevent the 4 psi air from diffusing to a lower pressure in the tube and would facilitate the evacuation of this air from the tube. Otherwise, the vacuum pumps would be required to pull a larger mass of low pressure (1 psi) air from the tube.

Methods of making up for friction losses as the vehicle travels along the flat portion of the tube was not considered in detail in this study, but will be briefly discussed. Since the air bearing friction force will be approximately 100 lbs. at 300 mph (see Chapter 4), the simplest means of keeping a constant velocity would be

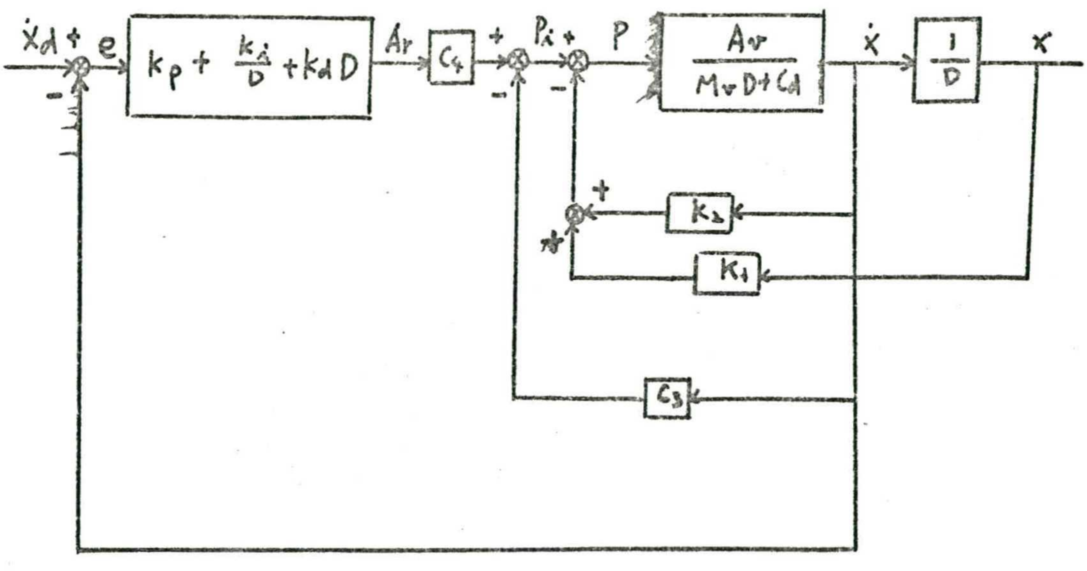

Figure 21
Linear System with a PID controller and with Pressure Drop

to provide a very slight slope to the tube track. Over a length of 100 miles the additional drop would only be 530 feet. Other schemes include the Linear Induction Motor boost-glide technique mentioned earlier in this chapter, periodic air pressure boosts behind the vehicle, or letting the vehicle's speed reduce while traveling horizontally. In the latter case, the vehicle would lose approximately 100 mph in a distance of 100 miles. This may be a little excessive for efficient operation of the system particularly if distances between stations are excessive.

5.9. Summary

In this chapter, a review of the potential propulsion systems which could be utilized in tube transit systems was presented. A combination pneumatic and gravity driven system was suggested. A detailed analysis of the controls necessary to yield a constant acceleration output for the pneumatic portion of the system was performed. This analysis showed that a PID controller could successfully be used to provide constant acceleration with an error of less than 3 %. The effects of restriction time lag and tube pressure drop were also investigated and shown to be detrimental to the control of the system if either the time lag or pressure drop are excessive.

Chapter 6. Conclusions and Recommendations

6.1. Conclusions

The conclusions obtained from this study are:

1. Tube transit systems are definitely feasible although the costs on present day standards may be exhorbitant.
2. The air bearing suspension, which was investigated in some detail, results in very low friction factors, but may be difficult to manufacture to the tolerances required. The low clearance requirements could definitely present operational difficulties. The required amount of air for the air bearing would be very low relative to other air cushion systems.
3. Pneumatic propulsion in combination with gravity seems to be very feasible. It is possible to supply adequate control to keep the vehicle's acceleration constant. A PID controller was used in the pneumatic control system to achieve this result. Advantages of pneumatic propulsion include it's simplicity and potential low cost.

6.2. Recommendations

1. Further investigations of the air bearing suspension should be performed to determine it's sensitivity to guideway disturbances and to determine modes of control for the bearing. Means of increasing the bearing gap without increasing the required air supply should be ascertained.

2. Gravity propulsion is definitely advantageous, but largely depends on the ability to excavate large, deep tunnels through a wide variety of materials. Therefore, tunneling methods must be improved such that these tunnels can be economically constructed.
3. The hardware necessary for the pneumatic propulsion system should be investigated to determine if the controller constants used in this study are feasible. If they are not, other modes of control such as acceleration feedback, feed forward control, and feedback controllers should be investigated.
4. Electromagnetic suspension should be investigated as an alternative to the air bearing suspension. Electromagnetic suspensions do not require any air and would be ideal for an evacuated system.
5. For an evacuated system, safety considerations such as protection from loss of pressure in the vehicle must be investigated.
6. Lateral support of the vehicle must be investigated along with the development of adequate seals for the pneumatic propulsion system.

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Appendix

Appendix A   Reynolds Number for Air Bearing Flow

Reynolds number $$Re = \frac{\rho V D}{\mu}$$

where D is called hydraulic diameter [34] for cross sections other than circular cross section and has the following relationship

$$D = \frac{4A}{C}$$

where A is the cross sectional area and C is the perimeter of the cross section

For the air-bearing case of Figure A-1

$$D = \frac{4A}{C} = \frac{4 \cdot \ell h}{2(\ell + h)} \approx \frac{4 \ell h}{2 \ell} = 2h$$

$$Re = \frac{\rho V D}{\mu} = \frac{2 \rho V h}{\mu}$$

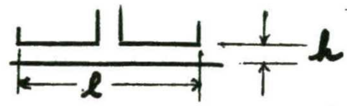

Figure A-1. Air Bearing

for the proposed system assume $T = 70°F = 530°R$ and consider the air-bearing film as a whole volume having an average pressure of 6 psi as explained in Chapter 4.

The equation of state

$$P = \rho R T$$

or $$\rho = \frac{P}{RT} = \frac{6}{53.35 \times 530 \times 12} \quad \frac{lbm}{in^3}$$

and $\mu\,(\text{air at } 70°F) = 0.026 \times 10^{-7} \ \frac{lbf\text{-}sec}{in^2}$

Then $$Re = \frac{2 \rho V h}{\mu} = 2220$$

Appendix B Gravity Propulsion Calculation

from Figure B-1, it is very clear that the forward acceleration

is $a = g \sin\theta$

$$V = V_i + a t$$

$$S = V_i t + \frac{1}{2} a t^2$$

$$V_f^2 = V_i^2 + 2aS$$

$$H = S \sin\theta$$

Therefore

$$V_f^2 = V_i^2 + 2a \cdot \frac{H}{\sin\theta}$$

$$= V_i^2 + 2 \cdot g \sin\theta \cdot \frac{H}{\sin\theta}$$

or $V_f^2 = V_i^2 + 2gH$

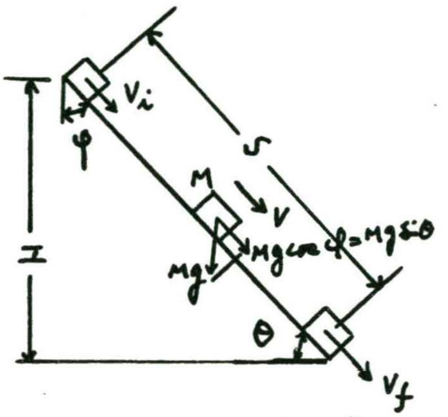

Figure B-1. Gravity Propulsion

Appendix C Restriction Mass Flow Rate Approximation Equation

Mass rate of flow through the flow restriction is

$$\dot{m} = \frac{A_r P_0}{\sqrt{T_0}} \left\{ \frac{2kg}{(k-1)R} \left[\left(\frac{P}{P_0}\right)^{2/k} - \left(\frac{P}{P_0}\right)^{k+1/k} \right] \right\}^{1/2} \qquad (C.1)$$

This equation is plotted in Figure C-1.

The tangent line to equation (C.1) for a specific value

of A_r is

$$\dot{m} = \dot{m}_{op} + \left.\frac{\partial \dot{m}}{\partial P}\right|_{P_{op}} (P - P_{op})$$

(see figure)

Now,

$$\dot{m} = \frac{A_r P_0}{\sqrt{T_0}} \left\{ \frac{2kg}{(k-1)R} \left[\left(\frac{P_{op}}{P_0}\right)^{2/k} - \left(\frac{P_{op}}{P_0}\right)^{k+1/k} \right] \right\}^{1/2} +$$

$$\frac{A_r P_0}{\sqrt{T_0}} \left[\frac{2kg}{(k-1)R} \right]^{1/2} \cdot \frac{1}{2} \left[\left(\frac{P_{op}}{P_0}\right)^{2/k} - \left(\frac{P_{op}}{P_0}\right)^{k+1/k} \right]^{-1/2} \times$$

$$\left\{ \frac{2}{k} \left(\frac{1}{P_0}\right)^{2/k} (P_{op})^{(2/k)-1} - \frac{k+1}{k} \left(\frac{1}{P_0}\right)^{\frac{k+1}{k}} (P_{op})^{\frac{k+1}{k}-1} \right\} \times$$

$$(P - P_{op})$$

$$\text{or}\quad \dot{m} = \frac{A_r P_o}{\sqrt{T_o}}\left\{\frac{2kg}{(k-1)R}\left[\left(\frac{P_{op}}{P_o}\right)^{2/k} - \left(\frac{P_{op}}{P_o}\right)^{\frac{k+1}{k}}\right]\right\}^{1/2} + \frac{1}{2}\,\frac{A_r P_o}{\sqrt{T_o}}\left\{\frac{2kg}{(k-1)R}\right\}^{1/2} \times$$

$$\frac{1}{P_{op}}\left[\left(\frac{P_{op}}{P_o}\right)^{2/k} - \left(\frac{P_{op}}{P_o}\right)^{\frac{k+1}{k}}\right]^{-1/2}\left[\frac{2}{k}\left(\frac{P_{op}}{P_o}\right)^{2/k} - \frac{k+1}{k}\left(\frac{P_{op}}{P_o}\right)^{\frac{k+1}{k}}\right] \times$$

$$(P - P_{op}) \qquad (C.2)$$

if $P_{op} = 4\,\text{psi}$, $P_o = 40\,\text{psi}$, and $T_o = 560°F$

then equation (C.2) becomes

$$\dot{m} = 0.464\,A_r + 0.0649\,A_r P - 0.2596\,A_r$$

$$\text{or}\quad \dot{m} = 0.2044\,A_r + 0.0649\,A_r P$$

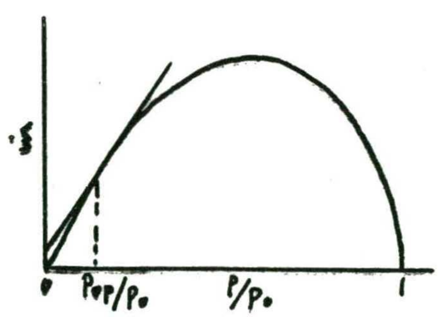

Figure C-1 Isentropic Flow Rate