MagneMotion Urban Maglev - Final Report

Project MA-26-7077-02.04.2

November, 2004

REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. AGENCY USE ONLY *(Leave blank)*	2. REPORT DATE November 2004	3. REPORT TYPE AND DATES COVERED Final Report

4. TITLE AND SUBTITLE M3 Urban Maglev – FTA Final Report	5. FUNDING NUMBERS
6. AUTHOR(S) R. Thornton, T Clark, J Wieler, B. Perreault, Walter Eggers	
7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES) MagneMotion 20 Sudbury Road, Acton, MA 01720	8. PERFORMING ORGANIZATION REPORT NUMBER TBP-10
9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES) U.S. Department of Transportation , Federal Transit Administration, Office of Technology, 400 Seventh Street, SW, Washington, DC 20590	10. SPONSORING/MONITORING AGENCY REPORT NUMBER MA-26-7077-02.04.2
11. SUPPLEMENTARY NOTES COTR: Mary Anderson, FTA	
12a. DISTRIBUTION/AVAILABILITY STATEMENT Limited Distribution Available From: National Technical Information Service, 5285 Port Royal Road, Springfield, VA 22161. Phone: 703-605-6000; Fax: 703-605-6900 Email: orders@ntis.gov	12b. DISTRIBUTION CODE N/A

13. ABSTRACT *(Maximum 200 words)*

The MagneMotion Urban Maglev System, called *M3*, is designed as an alternative to all conventional guided transportation systems. Advantages include major reductions in travel time, operating cost, capital cost, noise, and energy consumption. Small vehicles operating automatically with headways of only a few seconds can be safely operated in clusters to achieve capacities of more than 12,000 passengers per hour per direction. Small vehicles lead to lighter guideways, shorter wait time for passengers, lower power requirements for wayside inverters, more effective regenerative braking and reduced station size. The result of the design is a system that can be built for about $25M per mile, excluding vehicles, stations and land acquisition.

The vehicles have arrays of permanent magnets to provide suspension and guidance forces and also provide the field for the Linear Synchronous Motor (LSM) propulsion system. Feedback-controlled currents in control coils wound around the magnets stabilize the suspension and provide active lateral damping. The LSM windings are integrated with the suspension rails and excited by inverters located along the guideway. The vehicle size and maximum speed can be varied over a wide range to accommodate virtually all applications served today by other guided transportation systems.

14. SUBJECT TERMS Maglev, Magnetic Levitation, Propulsion, Urban, Requirements, Performance, Safety, Design, Analysis, Cost	15. NUMBER OF PAGES 83
	16. PRICE CODE N/A

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
Unclassified	Unclassified	Unclassified	

NSN 7540-01-280-5500

Standard Form 298 (Rev. 2-89)
Prescribed by ANSI Std. 239-18
298-102

Foreword

This Final Report is written in fulfillment of FTA Project MA-26-7077-02. As part of this project we have completed investigations on critical aspects of the Urban Maglev system called *M3*, MagneMotion Maglev. This report documents the maglev system architecture, provides a high level description of all components and their performance characteristics, presents the findings of our initial cost analysis, and outlines our development plan.

Our intended audience is managers and engineers with working knowledge of transportation systems and technology. The report supersedes the final report generated as part of FTA Project MA-26-7077-01.

Abbreviations and Acronyms

AASHTO	American Association of State Highway and Transportation Officials
APM	Automated People Mover
DSP	Digital Signal Processor
EDS	ElectroDynamic Suspension
EMS	ElecctroMagnetic Suspension
FTA	Federal Transit Administration
HSST	High Speed Surface Transportation
IGBT	Insulated Gate Bipolar Transistor
IPT	Inductive Power Transfer
ISO	International Standards Organization
LRFD	Load and Resistance Factor Design
LIM	Linear Induction Motor
LSM	Linear Synchronous Motor
NCHRP	National Cooperative Highway Research Program
NdFeB	Neodymium-Iron-Boron magnet material
pphpd	passengers per hour per direction

Metric Conversion Factors

The following Table gives factors for converting units used in this report. To convert from English to Metric multiply by the E→M factor, to convert from Metric to English multiply by the M→E factor.

	E unit	*M unit*	*E→M*	*M→E*
Distance	inch	mm	25.4	0.0394
	foot	m	3.2808	0.3048
	mile	km	1.6093	0.6214
Mass	lb	kg	0.4536	2.2046
	ton	Mg	0.9072	1.1023
Speed	mph	m/s	0.4470	2.2369
	mph	km/h	1.6093	0.6214
Energy	BTU	kJ	1.0544	0.9484
	BTU	Wh	0.2929	3.4149

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November, 2004

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Abstract

The MagneMotion Urban Maglev System, called *M3*, is designed as an alternative to all conventional guided transportation systems. Advantages include major reductions in travel time, operating cost, capital cost, noise, and energy consumption. Bus size vehicles operating automatically with headways of only a few seconds can be safely operated in clusters to achieve capacities of more than 12,000 passengers per hour per direction. Small vehicles lead to lighter guideways, shorter wait time for passengers, lower power requirements for wayside inverters, more effective regenerative braking and reduced station size. The result of the design is a system that can be built for about $25M per mile, excluding vehicles, stations and land acquisition.

The vehicles have arrays of permanent magnets to provide suspension and guidance forces and also provide the field for the Linear Synchronous Motor (LSM) propulsion system. Feedback-controlled currents in control coils wound around the magnets stabilize the suspension and provide active lateral damping for a stable ride. The LSM windings are integrated with the suspension rails and excited by inverters located along the guideway. The vehicle size and maximum speed can be varied over a wide range to accommodate virtually all applications served today by other guided transportation systems.

Executive Summary

The U.S. DOT Federal Transit Administration (FTA) launched an Urban Maglev Project in 1999 in order to create designs suitable for urban transit. MagneMotion, EarthTech, TPI Composites and others formed a team and, with cost sharing support from FTA, have developed an urban maglev system called *M3* that is suitable for a Maglev People Mover (MPM). This is the Final Report for the work completed to date. It provides an overview of the system with detailed discussion of the design of the suspension, propulsion, guideway, control system, and vehicle. It also describes the experimental measurements conducted on the demonstration prototype and provides applications examples, cost estimates and ongoing development plans.

The *M3* system was designed with a system approach focused on providing cost effective, fast, safe and environmentally friendly travel in an urban environment. Features of *M3* are:

- Small, lightweight vehicles operate at speeds up to 60 m/s (134 mph) with accelerations up to 0.2 *g*;
- A guideway based control system with Linear Synchronous Motor (LSM) propulsion allows short headway for capacities to 12,000 passengers per hour per direction;
- Permanent magnets provide key magnetic forces with high efficiency and a magnetic gap of 20 mm;
- Lightweight guideways provide good ride quality at minimum cost.

Conventional high speed maglev designs, such as Transrapid with its magnetic gap of 10 mm and top speed of 120 m/s (268 mph), require tight dimensional tolerance and stiffness for the supporting beams in order to achieve good ride quality. The use of a larger gap reduces guideway tolerance requirements and a unique suspension design allows a minimum horizontal turn radius of 18.3 m (60'). The permanent magnets provide passive lateral guidance with active damping used to improve ride quality. Onboard power requirements are so small that it is feasible to use inductive power transfer to provide onboard power at all speeds so there are no contacts between the vehicle and guideway and no exposed third rail or catenary wires.

Maglev suspension and Linear Synchronous Motor (LSM) propulsion technologies allow wayside communication and control and accurate position sensing that remove the technical and safety hurdles often associated with operating at short headways. All propulsion and control components are on the guideway so there is no need for radio communication of safety-critical information. Precise position sensing and frictionless propulsion and braking allow short headways and high capacity with small vehicles. The guideway is excited in short enough blocks to provide average propulsion efficiencies of over 80%, including all losses in the motor, electronics and power distribution system. Additional energy savings are possible because with closely spaced vehicles it is feasible to reuse energy regenerated by decelerating vehicles.

A two-span guideway design was developed for the *M3* system using traditional member geometry and materials (concrete and steel). Various designs based on different construction methods were developed. Dynamic vehicle-guideway interaction analyses were performed to assess ride quality. Guideway deflections from live load, thermal distortion and material creep and their effect on ride quality were considered. Excellent ride quality was achieved using very light beams. The estimated cost of the proposed *M3* guideway is less than the guideway cost for other APM systems and much less than the cost of competing maglev designs. Detailed cost analysis indicates that a long dual guideway can be constructed for less than 10 million dollars per mile, exclusive of land and station cost.

Vehicle weight reduction is one of the top three priorities along with cost and reliability. The composite vehicle has a weight at least 30% less than for an aluminum body vehicle. Manufacturing processes allow for a 100% composite body that eliminates any composite/steel interface typically found in other vehicles. The composite sandwich construction also provides good insulation properties and is superior to metal construction at damping vibration and noise. The baseline vehicle is 10 meters long and capable of carrying 24 people seated and 8 standees *or* 20 people seated and 20 standees *or* 32 people seated with no standees. The empty vehicle weight is 5.5 tonnes and the maximum loaded vehicle weight is 8.5 tonnes. The use of light vehicles has a major impact on the cost of the guideway and propulsion system and leads to reduced energy consumption.

The *M3* system design achieves high throughput with many small lightweight vehicles operating in clusters. The vehicles within a cluster operate with headways as short as 4 seconds but inter-cluster headway is large enough to allow stopping in the clear distance ahead. This cluster-based operation is the key to achieving high throughput with small vehicles, analogous to the way highways provide high throughput with small vehicles. The use of smaller vehicles results in shorter wait times than larger vehicles at equivalent throughput, especially at times of low demand. It also allows more flexible scheduling without the need for all vehicles to stop at all stations.

An accurate, inductive position-sensing system coupled with a wayside-based control system allows for the sharing of vehicle information over a reliable, hard-wired link. Knowledge of the precise position of all vehicles and the ability to apply consistent thrust in all weather conditions greatly facilitates the implementation of a short-headway control system. The control system for *M3* was developed to meet the goals of safety, reliability, efficiency, flexibility, and effective fault handling. It uses the concept of a 'target', or planned stopping point. Each motor controller is responsible for allocating and releasing movement permissions for targets within its borders, and utilizes 'brick-wall' inter-cluster headways and 'safe-follower' intra-cluster headways. This kind of operation allows for a level of safety similar to that of larger trains operating with brick-wall control. Use of small vehicles and systems that employ precise position sensing can reduce transportation capital costs and energy consumption simultaneous with decreases in average travel time and life cycle cost.

A demonstration prototype was constructed and detailed measurements confirmed that our computer modeling and simulation provide accurate predictions of performance. The vehicle required only a few watts for levitation and the propulsion system is more efficient than most rotary motor propulsion systems.

Four application examples were worked out in enough detail to confirm that for a variety of applications it is possible to achieve average travel times much lower than with conventional transit and in many cases the advantage is more than a factor of two. Detailed cost calculations confirm that even with the improved performance the installation and operating cost is less than half of that for typical wheel-based designs.

1 Introduction

The design of the *M3* system has focused on the components that contribute most to performance and cost. *M3* was designed with specific objectives for improving on conventional guideway-based transportation technologies and strategies for realizing the objectives.

- *Decrease travel time by a substantial amount.*
  Allow speeds up to 605 m/s (134 mph), acceleration and braking up to 2 m/s^2, short average waiting time and reduced dwell time.
- *Decrease operating cost by at least a factor of 2.*
  Reduce labor and maintenance costs and use less energy.
- *Reduce guideway construction cost*.
  Reduce guideway weight by reducing vehicle weight and matching the guideway to the vehicle.
- *Reduce environmental impact.*
  Reduce noise, guideway size and energy consumption.
- *Create an improved ElectroMagnetic Suspension (EMS).*
  Use permanent magnets with a 20 mm magnetic gap and make each magnet contribute to lift, guidance and Linear Synchronous Motor (LSM) propulsion.
- *Provide excellent ride quality.*
  Pay careful attention to guideway and vehicle design and take advantage of the distributed and non-contacting nature of maglev forces.
- *Create a very safe transportation system.*
  Use a dedicated guideway, vehicles that cannot derail, linear motor propulsion that does not depend on friction, and totally automated operation.

A key feature that drives the *M3* system concept is the use of small and light vehicles operating with short headway. The light weight contributes to reduced guideway cost and the small size, in conjunction with short headway, reduces wait time and allows station-skipping operation. MagneMotion's objective is to develop and demonstrate an urban maglev system with the following attributes:

- Electromagnetic suspension (EMS) system that uses the same permanent magnet field for propulsion, suspension, braking and guidance.
- Goal of $25M per mile of dual guideway excluding vehicles and stations
- Multi-vehicle control with headways as short as 4 seconds.
- Operational speeds up to 60 m/s (134 mph).
- Horizontal turns with a minimum radius of 18.3 m (60').
- Vertical turns with a minimum radius of 300 m (1000').
- Ability to climb and stop on grades of up to 18%.
- Vehicle bank angles of 18º using a combination of guideway superelevation and vehicle tilt mechanisms.
- Mechanical switching.

This report is organized as follows:

- Introduction
- Overview of system design
- Discussion of the major subsystems
    a. Magnetic suspension and guidance
    b. Linear synchronous motor propulsion
    c. Guideway design and manufacture
    d. Vehicle attributes
    e. Control system design and operation
- Description of a demonstration prototype and its performance
- Examples of four applications
- Cost estimates for a baseline design
- Continuing development plans and activities
- Conclusion

2 Overview

In the last 35 years maglev has changed from an engineering curiosity to the basis for commercial installations, one now operational in China and one soon to be operational in Japan. German and Japanese efforts over many years have demonstrated maglev's potential for safe, fast and economically viable transportation but potential users have not been impressed enough to install a major commercial system until very recently. The lack of commercial support has been partly due to emphatic statements by critics (there are also many supporters) from academia, industry and the government that maglev is too expensive in comparison with other types of guided transportation. These criticisms are not based on valid technical arguments but are akin to the criticisms of railroads that were made in the early 1800s when the "smart money" was being invested in canals. Unfortunately, maglev enthusiasts have not helped the cause by often focusing more on the technology than on what it can deliver to the user.

A principal problem with past maglev efforts has been an excessive emphasis on speed and technology without taking a system approach to solving a transportation problem. With this in mind, MagneMotion has stressed the system approach and examined all aspects of the problem of providing high quality and cost effective transportation with maglev by taking advantage of recent advancements in enabling technologies. For U.S. applications MagneMotion believes a key market for maglev today is in the low and middle speed region now dominated by light rail, rapid transit, commuter rail and all versions of Automated People Movers (APM). The MagneMotion Maglev system, called *M3*, is currently focused on speeds up to 60 m/s (134 mph) but with minor modifications the system could compete with any guided system including ones with both lower and higher speed capability.

A fundamental property of magnetic structures, called Earnshaw's Theorem, is that no static configuration of magnets can be levitated so as to be stable in all degrees of freedom. It is possible to be stable in all but one dimension, so it is possible to have a magnetic suspension stable in the vertical direction but then it must be unstable in a lateral direction. Such structures have been proposed but they tend to be heavier and more complex than if electronic control is used to stabilize the suspension in the vertical direction. The vertical stabilization approach to ElectroMagnetic Suspension (EMS) has now been proven to be suitable for operation over a wide range of speeds. For example, the new Shanghai Transrapid maglev installation uses this approach and is now carrying passengers at speeds up to 430 km/h (267 mph), 34% faster than the fastest high-speed trains in operation today.

Historically, the major disadvantage of EMS is the need to use magnetic gaps no larger than about 10 mm. MagneMotion has overcome this disadvantage by using permanent magnets in conjunction with control coils to allow a magnetic gap of 20 mm. Although some ElectroDynamic Suspension (EDS) designs feature larger gaps, it is not clear there is a need for gaps greater than about 20 mm at speeds that are of interest for urban transportation.

This report gives a description of *M3* as it exists at this time. Although the design is expected to evolve over the next few years it is unlikely to change in any major way. This report can be used to assess the potential merits of *M3* for specific applications.

Figure 2.1 shows a 3D view of a preliminary version of the baseline vehicle and some of its features.

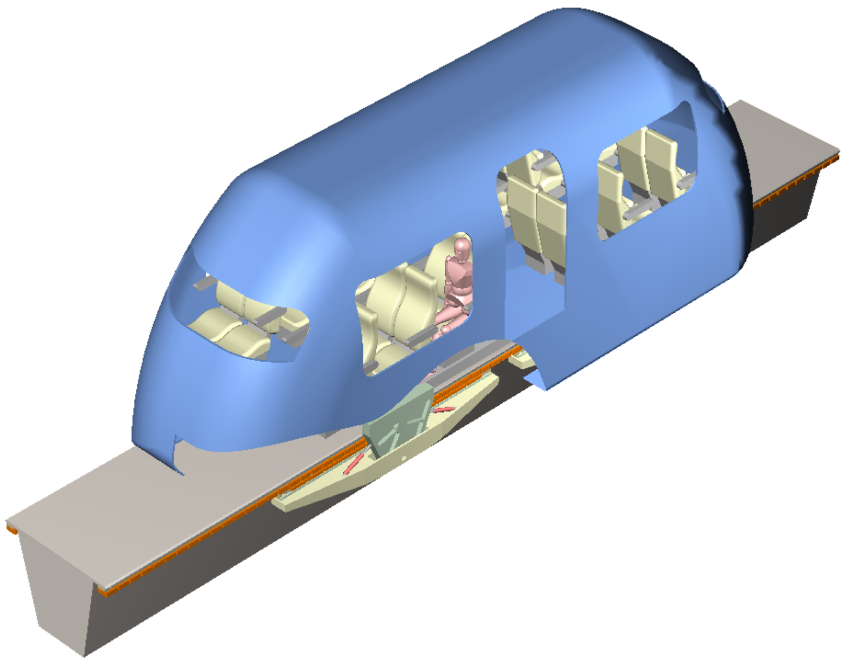


Fig. 2.1. Preliminary vehicle and guideway design.

Following are the key performance specifications that were the basis of the design:

- Speeds up to 60 m/s (216 km/h, 134 mph).
- Acceleration and braking up to 2 m/s^2 (4.4 mph/s).
- Headways as short as 4 seconds when operated in clusters.
- Capacity up to 12,000 passengers per hour per direction (pphpd).
- Horizontal turn radii of 18.3 m (60') and vertical radii of 300 m (984').
- Target cost of $25 million per mile excluding vehicles, stations and land.
- Minimum environmental impact with reduced noise and guideway size.

Figure 2.2 shows a cross-section of the guideway beam and the vehicle. The permanent magnets on the vehicle provide lift, guidance and act as the field for Linear Synchronous Motor (LSM) propulsion. Pivoting magnet pods allow short turning radii in both horizontal and vertical directions. Control coils wound around the magnets stabilize the suspension and adjust the nominal magnetic gap to the value that minimizes power requirements for the control. Windings in the guideway are excited by inverters located along the guideway and provide controllable thrust for acceleration, cruise and braking. The secondary suspension on the vehicle provides improved ride quality but can be omitted for lower speed operation.

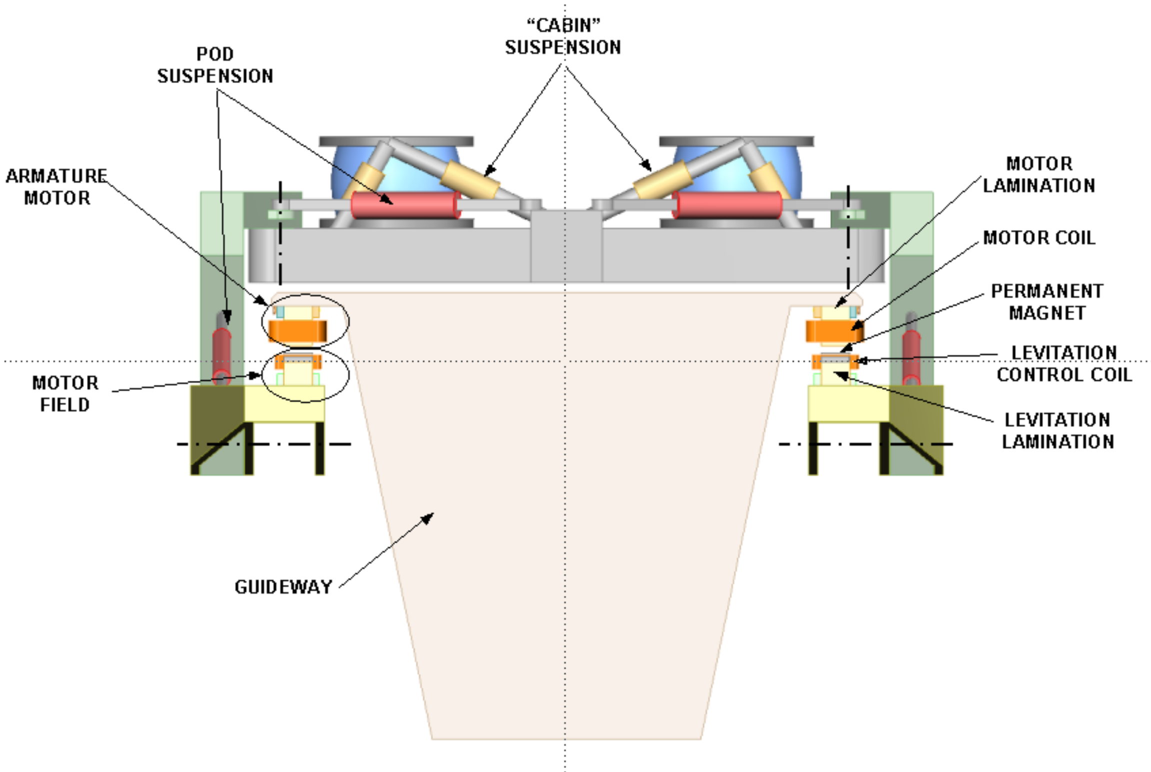


Fig. 2.2. Cross-section of guideway beam and preliminary vehicle suspension.

The design of the *M3* system has focused on the components that contribute most to performance and cost with a particular focus on subsystems that have unique features: permanent magnet EMS suspension, LSM design and manufacture, guideway beams, vehicle suspension and control systems. In order to create confidence in the basic design a demonstration prototype has been constructed and tests to date are very encouraging. Future plans call for extending the test track and building a high-speed test track as a prelude to installing a commercial system. A description of our prototype is given in Section 9 and a summary of our continuing development plan is given in Section 11.

3 Electromagnetic suspension and guidance

A key design objective was to create a suspension that is suitable for low to moderate speeds with frequent station stops, allows vehicles to make small radius turns in both the horizontal and vertical directions, and is suitable for use with small vehicles. Members of the MagneMotion maglev team have had considerable experience with both ElectroMagnetic Suspension (EMS) and ElectroDynamic Suspension (EDS). A careful review of the merits of each led us to pick the EMS design for the following reasons:

- Ability to levitate at all speeds.
- No need for high propulsive force at low speed to overcome magnetic drag.
- No need to shield the passengers from unacceptably high magnetic fields.
- Reduced cost for a complete system.

Following is a discussion of the *M3* features that contribute to decreasing cost and increasing performance.

3.1 Permanent magnet EMS

A key feature of the *M3* suspension is that every permanent magnet on the vehicle contributes to suspension, guidance and propulsion. This is analogous to the way every railroad wheel provides suspension and guidance and plays a key role in propulsion and braking. Without this 3-way combination there is added cost and complexity. For example, Transrapid uses one set of electromagnets to provide both lift and a field for an LSM but requires separate steel rails on the guideway and a separate set of feedback controlled electromagnets on the vehicle to provide guidance. The Japanese low speed HSST and Korean Maglev designs provide lift and guidance with a single electromagnetic structure but require a separate aluminum reaction rail on the guideway and Linear Induction Motor (LIM) primary on the vehicle to provide propulsion. For *M3* the integration of these three functions allows the vehicle magnet arrays to be mounted on pods that can rotate like wheel bogeys to allow sharp turns in both the horizontal and vertical directions.

Figure 3.1 shows a pod with permanent magnets attracted upward to a laminated steel suspension rail. Control coils around the magnets are used for stabilization and windings integrated into the suspension rails are excited by wayside power inverters to provide propulsion. Half-length magnets at the ends of the pod equalize magnetic flux and mitigate cogging. This drawing shows propulsion windings wound on teeth on a guideway rail and suspension control coils wound around permanent magnets on a vehicle pod.

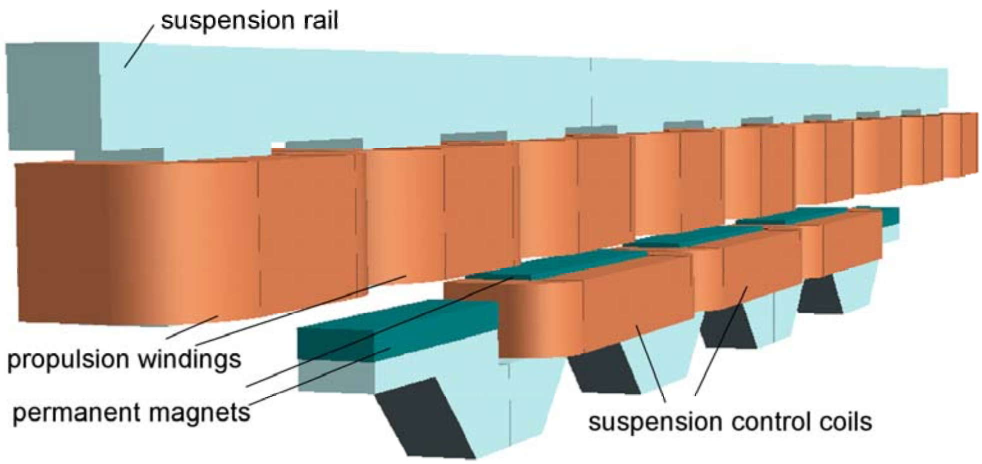


Fig. 3.1. A vehicle's magnet pod attracted upwards to a suspension rail.

Coils wound around the magnets are excited from a controller that uses gap and acceleration sensors to control current in the coils. This stabilizes the magnetic gap at that value which provides a match between vehicle weight and permanent magnet force. Ideally it would take negligible power to stabilize the suspension and in practice the power requirement is dramatically less than it would be if the entire suspension force were provided by electromagnets alone. When the vehicle is stationary the required control power will be only a few watts and at operational speeds simulations show it to be on the order of 100 W per Mg of vehicle mass. For comparison, Transrapid uses electromagnets for suspension and they require 1,000 W per Mg of vehicle mass for a suspension with a magnetic gap of only 10 mm, and require substantial additional power for guidance.

The use of permanent magnets allows the use of a magnetic gap of 20 mm with a corresponding reduction in guideway tolerance requirements. The vehicle mass is estimated to be 5.5 Mg empty with a load of up to 3 Mg depending on the number of passengers. The suspension controller will adjust the magnetic gap to minimize control power and thus the gap will vary 23 mm for an empty vehicle to 17 mm for a fully loaded vehicle.

3.2 Lateral guidance and damping

The suspension system must also provide lateral forces to guide the vehicle and resist lateral forces due to turns and wind. An important feature of *M3* is the way the magnets that provide suspension forces also provide guidance forces. If the vehicle is displaced laterally there will be strong restoring forces created by the tendency of the magnets to align themselves with the steel suspension rails on the guideway. By using a magnetic gap that is 25% the width of the suspension rails it is possible to provide passive guidance with a lateral guidance force up to 40 % of the vertical lift force. Figure 3.2 shows the results of a 3D Finite Element Analysis (FEA) of a 3.25 meter long pod (12 full size magnets and 2 half-magnets at the ends) designed to support 25% of a baseline vehicle. The plot shows the magnetic gap and lateral guidance force F_z as a function of lateral displacement z_d for a nominal load. With a 20 mm gap there is 16.7 kN of lift force when there is no lateral displacement. With a lateral displacement of 40 mm the magnetic gap would drop to 14.7 mm (to maintain the vertical force) and there would be a lateral restoring force of 0.33 g = 5.5 kN. Vertical and lateral skid-pads will be provided to deal with extreme forces, such as might happen during an earthquake.

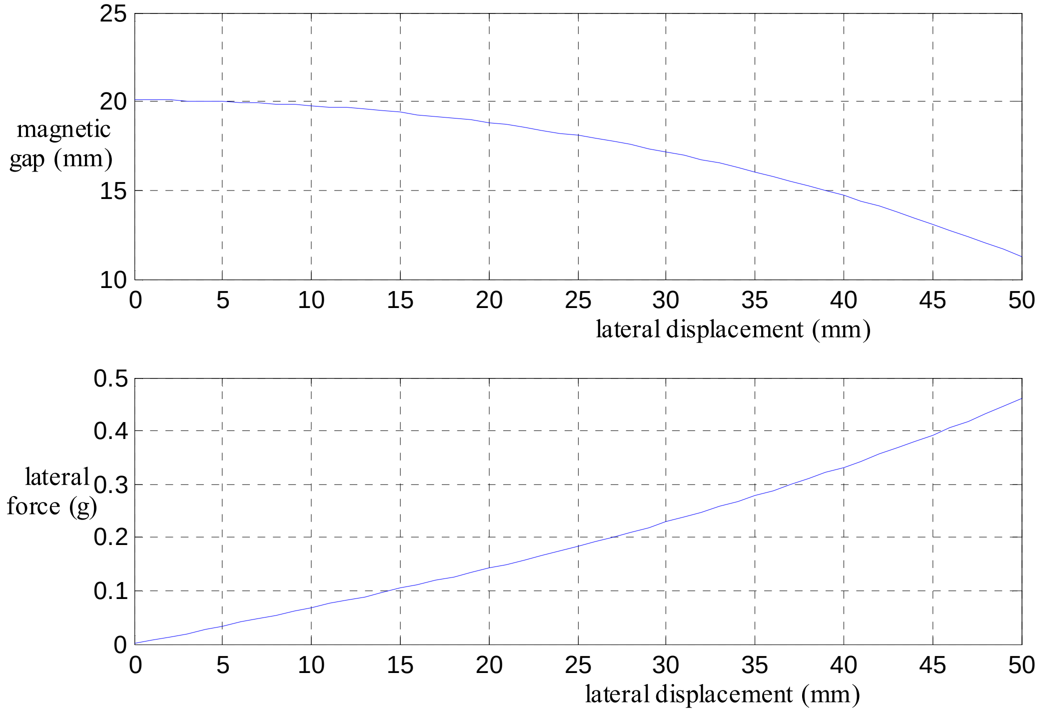


Fig. 3.2. Vertical gap and lateral force vs. lateral displacement.

Although the suspension is passively stable for lateral motion, there is very little damping so other means must be provided to prevent excessive lateral motion. We have achieved active lateral damping and this is described in Section 8 as part of the discussion of our demonstration prototype.

3.3 Horizontal and vertical turns

Creating a maglev system that can negotiate tight turns has been a challenge to all maglev designers. In a cost-effective design the magnetic force must be distributed over a large area but for making tight turns the suspension magnets must be articulated so that they follow the turn. The *M3* mechanism for doing this is shown in Figs 3.3-4. This preliminary design is for a 36-passenger (24 seated) vehicle that can negotiate horizontal turn radii of 18.3 m (60') and vertical turn radii of 300 m (984'). Improved designs are being studied.

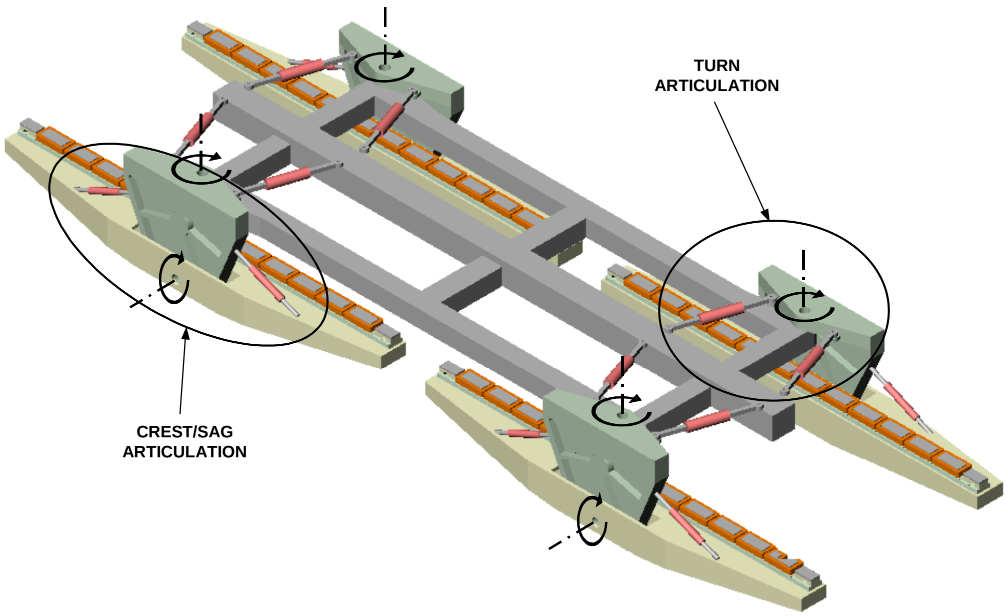


Fig. 3.3. Suspension mechanism showing pod pivoting for turns.

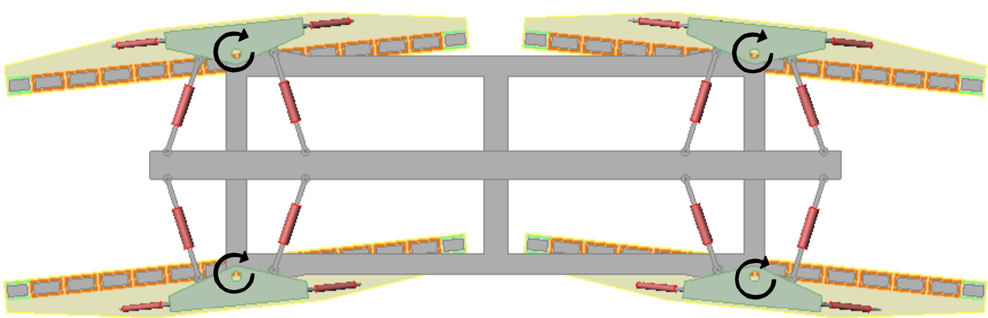


Fig. 3.4. Top view of suspension system.

4 Linear Motor Propulsion

4.1 Comparison of synchronous and asynchronous linear motors

Maglev developers have universally adopted the linear electric motor as the propulsion system of choice for maglev. There are two types of linear motor that are currently being used for commercial designs: Linear Induction Motor (LIM), also called a linear asynchronous motor) and Linear Synchronous Motor (LSM).The only practical version of the LIM is one that has the propulsion windings on the vehicle, the "short primary" design. This design has some advantages:

- A power inverter is required for each vehicle motor, but the total cost of inverters for a complete system is reduced.
- The guideway portion of the LIM consists of an aluminum sheet on steel backing and this is less expensive than an LSM stator.

The disadvantages of the LIM include:

- The vehicle weight is increased by at least 20% because of the onboard propulsion equipment.
- It is very costly in weight and efficiency to operate with a LIM magnetic gap more than about 10 mm and thus guideway tolerances are more critical.
- It is necessary to use sliding contacts to transfer all of the propulsion power to the vehicle or, at much greater cost, to use inductive power transfer.
- The motor efficiency is reduced, both because the motor is less efficient and because the vehicle is heavier and requires more propulsive thrust.

The only practical version of an LSM is one that has the propulsion winding on the guideway, the "long primary" design. This has a number of important advantages:

- The motor can use the same magnets as the suspension and thereby reduce vehicle cost and weight and increase efficiency.
- The magnetic gap can be larger.
- The vehicles are lighter so less propulsive power is required.
- There is no need to transmit propulsive power to the vehicle.
- The propulsion and control equipment is all on the guideway so communication is more robust, control is simplified and regenerative braking is easier to achieve.

The disadvantages of an LSM include:

- Higher cost for guideway-mounted LSM motor windings and wayside power inverters.
- Precise position sensing is required.

Virtually all high-speed maglev designs use an LSM for propulsion. Early versions of Transrapid used the LIM but starting with TR05 in 1975 they switched to the LSM. The Japanese high-speed maglev developers have always used an LSM. The Japanese HSST and Korean designs use a LIM but they have limited speed capability. A superficial analysis of cost might suggest that LIM propulsion is less expensive but when all of the costs associated with the negative aspects are considered it is likely to be more expensive for a complete system. The dramatic reduction in the onboard power requirements is also a strong incentive for using an LSM. For *M3* with light vehicles and a 20 mm gap the LIM is not a viable alternative. Details of the *M3* LSM design are discussed in this section.

4.2 The tradeoff between cost and performance

An LSM can be designed to give almost any desired performance, but increased performance implies increased cost. The design problem is to find that level of performance that is most cost effective. For example, we could use a smaller LSM that produces less thrust and is less expensive, but then vehicle acceleration is reduced so travel time is increased and we lose many of the advantages of a higher top speed.

For standing passengers it has been generally accepted that an acceleration of 1.6 m/s^2 is an upper limit for safe operation. Since an urban vehicle stops frequently and often has standees it was decided to limit acceleration to this value. In order to be able to accelerate a fully loaded vehicle at 1.6 m/s^2 it is necessary to have more thrust than is necessary for the same acceleration for a nominal load. Thus the *M3* design calls for an acceleration capability of 2 $m/s^2$with a nominal load. This also allows a maximally loaded vehicle to maintain an acceleration of 1.6 m/s^2 up to a higher speed.

In many examples that have been considered there are sizable regions where it is not necessary to provide rapid acceleration or deceleration and in these regions it is possible to reduce the propulsive power with a resulting saving in cost. The reduced propulsive power also implies reduced braking capability from the LSM, but this can be made up by other means, as will be discussed later.

4.3 Block length optimization

An LSM winding on the guideway is divided into sections called blocks and each block is excited by a wayside inverter to provide thrust. An important constraint is that only one vehicle can be in one block at one time. At low speeds and high acceleration we would like to have low winding resistance in order to have high efficiency with minimum inverter rating. At high speeds and moderate acceleration, the power loss in winding resistance is relatively low, but the winding inductance plays a major role in limiting performance and leads to a higher VA rating on the inverters. In order to achieve acceptable values of winding resistance and inductance it is necessary to limit the length of propulsion winding that is excited at any time. For *M3* a good choice of block length is in the range 12 to 72 meters depending on speed and acceleration requirements. In order to simplify installation it is convenient to match the block length to the guideway pier spacing. In a later section it will be shown that a good pier-to-pier spacing is 36 meters, so this has been chosen as the nominal block length. Near and in stations a 12 or 18 meter block length will be used and in regions where constant speed is the norm the block length may be as long as 72 meters. The combination of block length and inverter size should be carefully chosen so as to provide desired performance at the lowest cost.

4.4 Propulsion power system and inverter design

The inverter changes the DC bus voltage into variable amplitude, variable frequency, 3-phase AC for activating the windings on the guideway. The inverter is the key component in the power system and there are a number of issues that must be addressed if reliable and cost effective operation is to be maintained over the complete speed and force range.

4.4.1 Cost reduction techniques

The principal means of reducing power system cost are:

- Develop an inverter that uses state-of-art design and is matched to the application
- Use sub-block switching with longer LSM blocks so as to reduce the number of inverters required
- Adapt the DC power source and power distribution to the inverter input and output requirements
- Develop tooling and procedures to reduce installation cost.

The optimum design for an application will depend upon many things, including design considerations discussed here and the latest evolution in materials, components and design methodology. Here are some factors to keep in mind when adapting *M3*;

- The requirement for high thrust at low speeds is the operating condition that dominates the inverter design problem. Low speed operation puts maximum thermal stress on the components and necessitates using larger components than might otherwise be needed. The inverter must be sized for high-speed operation, but at high speeds the currents and switching losses are lower so there is less thermal stress on components.
- Inverters supply AC currents to the windings and for high power inverters it is essential to use switching frequencies in the middle of the audio spectrum. It may turn out that special features must be added in order to reduce audio noise to acceptable levels but these features can be added later if the need arises.
- Power system cost is expected to be 5 to 5.5 million dollars per dual-guideway mile with the higher value for systems with higher capacity. Increasing the maximum speed has relatively little impact on system cost.
- A major component of the power system cost is for installation of inverters, switches and cables. A major effort will be made to simplify the installation.
- The *M*3 propulsion system consumes substantially less energy per passenger mile than any conventional transit system in spite of the higher operating speed.

At speeds above about 23 m/s (51 mph) the inverter can be operated at maximum output with relatively low temperature rise for the key components. At speeds below 1 m/s (2.2 mph) the inverter output frequency is less than 2 Hz and it is necessary to use relatively rapid switching to create an approximation to the desired sinusoidal current waveforms. The thermal time constants of the semiconductor devices are so short that for these speeds the currents must be limited because of instantaneous power dissipation constraints, which limit the ability to distribute the dissipation equally over all modules. Perhaps more important, at these very low speeds the semiconductor devices are subject to thermal cycling that is known to have a deleterious effect on device longevity. At speeds between 1 and 23 m/s the devices are subject to substantial thermal stress but with a well-designed inverter the device junction temperatures are well below maximum limits.

The cost of a propulsion system can be reduced by using one inverter to power several sub-blocks with relatively simple thyristor switches used to determine which sub-block is activated. Several sub-block switching strategies were considered and it is possible to reduce inverter-related cost by up to a factor of two with the design details depending on the application. For the baseline design we assume there is an average of 1 inverter for every 2 blocks and each inverter includes a 2-way, 3-phase thyristor switch.

4.4.2 Inverter design and performance

A number of manufacturers sell relatively low cost IGBT (Insulated Gate Bipolar Transistor) power modules that contain most of the high power components for a 3-phase inverter. Use of these standard modules reduces system cost while allowing wide flexibility in the control. MagneMotion has spent many years refining the design of LSM inverter controllers and the use of these controllers in conjunction with standard power modules has led to a dramatic reduction in inverter cost as compared with the use of commercially available inverters.

The *M3* controllers use Digital Signal Processors (DSP) to process signals from the position sensors and control the power device switching according to the required output with the highest possible efficiency. At low speed the power devices switch at a frequency of a few kHz in such a way as to produce nearly sinusoidal currents with the correct frequency, phase and current for the required force and speed. At high speeds the switching is synchronized with the propulsion frequency and phase control is used to control thrust. This combination of asynchronous and synchronous switching allows the controller to drive a highly inductive load over a wide speed and force range without requiring an excessive Volt-Ampere rating for the power modules.

Figure 4.1 shows contours of constant inefficiency in the force vs. speed domain for a typical *M3* design. For this example the maximum force is 7 kN up to a speed of 27 m/s (60 mph) and then the power is limited to 189 kW for speeds up to 45 m/s (101 mph). The dissipation is primarily due to LSM winding resistance loss and the maximum power limit is primarily due to winding inductance. For this example the full 189 kW output is available up to 60 m/s (134 mph) and this is the maximum operational speed without any changes in inverter design. The inefficiency in the plot is defined as the ratio of power dissipated to power delivered and is only for the motor. For a typical vehicle mission, including the losses in inverters and the DC power system, the average inefficiency is about 12 % giving a propulsion efficiency of 88%. The propulsion efficiency is higher than for most wheel-based transit systems and, as discussed elsewhere, the lighter vehicles leads to additional energy savings.

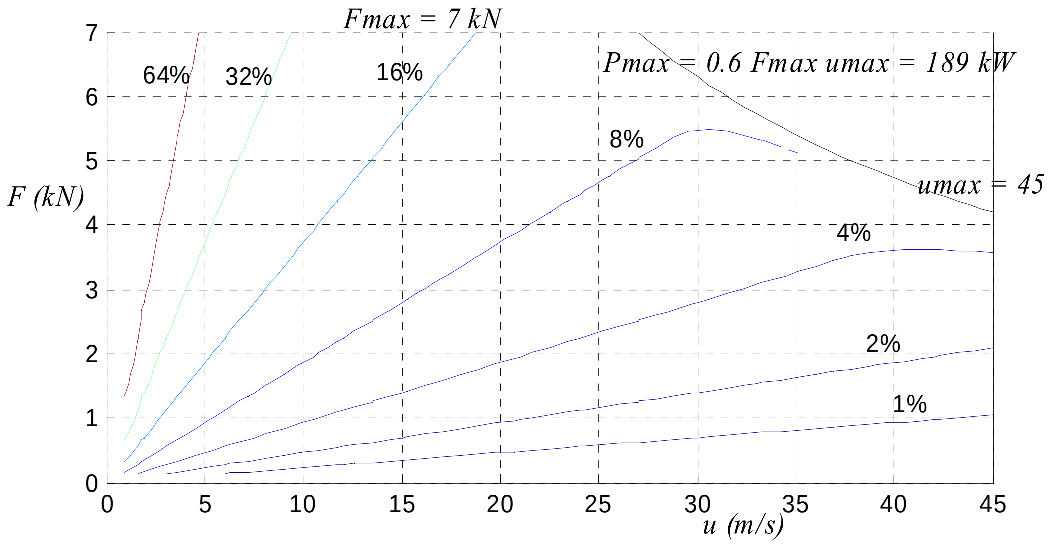


Fig. 4.1. Contours of constant inefficiency in the force vs. speed domain.

4.4.3 Acoustic noise

If the LSM propulsion were for a steel-wheeled vehicle the noise produced by electromagnetic forces would be masked by wheel and gear noise, but for maglev there is a much higher expectancy that noise be minimal. Some audible noise will always be produced by an LSM even if the waveforms were perfect sinusoids at the synchronous speed. With a switch-mode controller other frequencies are created, most notably harmonics of the propulsion frequency and frequencies generated by switching that is not synchronous with the propulsion frequency. In order to control thrust at low speed it is necessary to use switching at frequencies on the order of 1 to 3 kHz, frequencies that are in a sensitive region of the audio spectrum. There are steps that can be taken in the mechanical construction to minimize audible noise, but ultimately it may be necessary to use additional electronic hardware in order to achieve acceptable levels of noise. More work is needed to quantify this problem.

4.5 Power distribution and control

Figure 4.2 shows a typical power distribution design. The electric utility provides 3-phase power to a rectifier station that then delivers DC power to a bus connecting the wayside inverters. The spacing between Rectifier Stations is determined by vehicle density and acceleration profiles but will typically be in the range 5 to 10 km (3.2 to 6.4 miles). For the baseline design a spacing of 8 km (4.97 miles) is assumed.

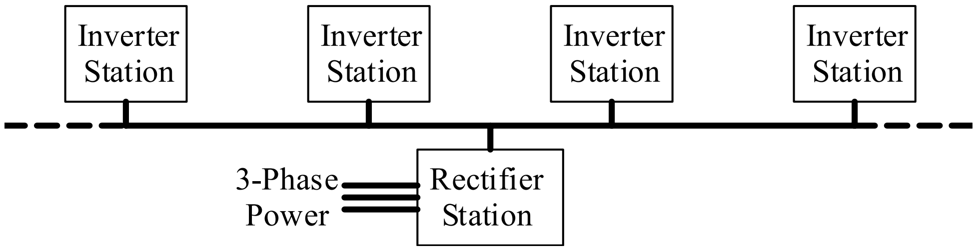


Fig. 4.2. Power distribution system.

The baseline design uses a 3-wire DC bus to transmit power at +900 VDC and -900 VDC with a 0 VDC wire connected to ground at frequent intervals. The DC bus and communication cables are installed in the beams on one side of the guideway. The port motor is powered from +900 V and the starboard motor is powered from -900 V, or vice versa. Since the currents in the two motors are nearly equal the majority of the power is distributed at the 1,800 V level; this high voltage allows longer distribution distances than are commonly used for rapid transit or light-rail. Relatively inexpensive 1,700 V IGBT power devices are now available and these will operate reliably on a 900 VDC bus. The DC bus is designed to carry 750 kW up to 4 km (2.49 miles) in each direction with a typical distribution efficiency of greater than 97% at full load. The rectifier station and cables are designed to provide 50% over-capacity for several minutes in order to deal with fluctuations in power demand. This baseline design is based on 4 vehicles per mile of dual guideway, each consuming 75 kW average, including onboard power which is derived from the same DC bus. In order to increase the number of vehicles we could add a second set of DC power cables in the second guideway, with the two sets of DC cables interconnected at the inverter stations, and increase the rectifier power rating.

The inverters not only provide power for accelerating vehicles, they are also used to decelerate the vehicles and deliver the vehicle kinetic energy back to the DC bus so it can be used elsewhere in the system. The use of regeneration can reduce total energy consumption by up to 40 % for a typical urban application, as will be seen in the next section.

Since only one vehicle can be present in a block at a time, inverter spacing must be short enough to deal with the minimum expected headway. For 4-second headway at 45 m/s we could, in principle, use one inverter to power a block that is almost 180 meters long. In a typical design we will use an inverter for every block near and in stations and an inverter for every other block between stations. This is done by means of sub-block switching with 3-phase thyristor switches used to determine which block is excited. With this scheme, and two blocks per inverter, vehicle headway can be as short as 120 meters when the block length is 36 meters. After the system is constructed if higher capacity is required the switches can be removed and a separate inverter used for each block.

4.6 Performance simulation

Figure 4.3 shows distance, velocity and power plots for a trip of 3.2 km (2 miles). It is assumed that the vehicle accelerates at a rate that is the minimum of 1.6 m/s^2 and a rate limited by the maximum available thrust from the motor for a nominal vehicle mass of 7

Mg. For deceleration the LSM is able to sustain a uniform 1.6 m/s^2 for almost the entire stopping time. It is assumed that there are no grades that prevent the acceleration or speed from being sustained. The model used for the plots in Fig. 4.3 includes the effects of aerodynamic drag, winding resistance and power system inefficiency.

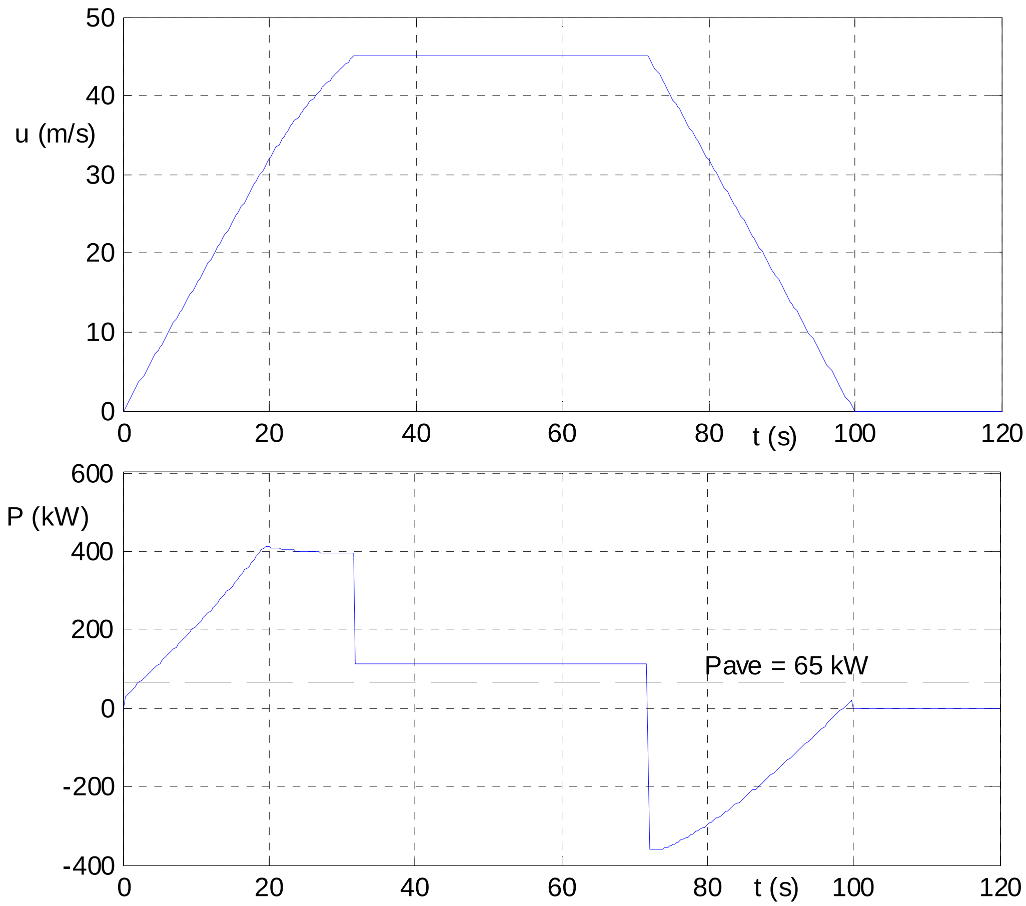


Fig. 4.3. Distance, speed and power profiles for a 3.2 km (2 mi) trip.

For the 3.2 km trip in Fig. 4.3 if the dwell time is 20 seconds the average speed is 27 m/s (60 mph). In order to estimate travel times for other trip lengths assume the speed is 45 m/s for the entire trip but with a time penalty of 30 seconds for every stop plus a dwell time, typically 20 to 30 seconds, for every stop. If the trip length is less than 1.6 km (1 mile) then the vehicle never reaches maximum speed so extra time is required.

Power consumption is: 412 kW peak; 113 kW cruise; 65 kW average with full use of regenerated power and 112 kW if braking energy is dumped in resistors without reuse. For this example regeneration provides a 42% saving in propulsion power cost. The savings would be less for a longer trip and more for a shorter trip.

In some cases more energy is being regenerated than can be used in a useful way and in this case the power must be dumped into resistors. It is possible to add a power dumping facility to each rectifier station, but this is unnecessary. There is always at least one inverter not being used for every inverter that is being used and these unused inverters can be used to dissipate power in the propulsion windings where there is no vehicle. This method of braking is particularly useful when, due to an emergency, it is desirable to stop every vehicle in the system in the shortest possible time. Preliminary calculations show that this can be done without the need for separate braking resistors. For emergency use braking resistors will be installed in rectifier stations but they do not need to be rated for continual use.

All electrically propelled transit systems create problems for the electric utilities because of the large and rapid fluctuation in power consumption. One advantage of using small, closely spaced vehicles is that starting times can be controlled to minimize the peak excursions. Simulations show that with only minor control of when a vehicle leaves a station it is possible to make full use of regenerated energy and reduce the peak power excursions by a large factor. More detailed simulations are planned when the system design is complete.

A problem all transit systems must address is the need to deal with electric power failures in a safe and effective way. Using battery backup of the inverter controllers so that they can maintain control solves the safety aspects of power failure. The controllers would cause the vehicles to brake and the braking energy would maintain the DC bus voltage until nearly all vehicles are stopped, at which point eddy-current and/or mechanical brakes would be used. The least expensive way to provide passenger egress in the event of power failure is to install a modest size standby power source in every rectifier station. A 100 kW DC generator, or possibly a standby battery, is adequate to provide some power for onboard vehicle use while also moving the vehicles, one by one, to a station where people can be unloaded. Such an emergency power source would add very little to the system cost but provide an important safeguard.

It is important that the complete power system be simulated in substantial detail before the design is finalized in order to identify potential problems with electrical transients and thermal behavior. Simulations should also be done to ensure that normal component or power system failure will not cause unreasonable consequences.

4.7 Efficiency and energy consumption

For a transit system energy cost is quite significant, typically more than 10% of operating cost. For *M3* it is estimated that for typical usage the propulsion efficiency will be in the range 80-85%. This includes all loss in the stator, inverter and power distribution system. In evaluating efficiency there are several points to keep in mind:

- The efficiency of the motor is very dependent on thrust and speed.
- The ability to use regenerated energy can reduce energy costs by a large factor.
- For urban applications a large fraction of the energy consumed is related to acceleration and deceleration, not cruising at constant speed.
- The use of light vehicles and station-skipping control strategies can greatly reduce energy usage.

One of the best measures of efficiency is the energy usage per passenger-mile of travel. From the plots of Fig. 4.3 we see that the average power required is 65 kW for an average speed of 60 mph or 3.9 MJ/mile. Assuming a nominal load of 18 passengers this implies a consumption of 217 kJ/pas-mi = 60 W-hr/pas-mi. In order to account for energy for

HVAC power usage and other factors, assume the actual consumption is 100 W-hrs/pas-mi. Assuming electric power cost of $0.12 per kW-hr, the energy cost is $0.012 per passenger mile. For continuous cruising at maximum speed the energy consumption is 62 W-hr/pas-mi, almost the same as with the stop.

In order to compare *M3* energy consumption with that of other transit systems we need to convert the energy to BTU/pas-mile. The theoretical conversion is 1055 J/BTU or 3.412 BTU/W-hr. For comparison with other modes we need to account for the 29% average efficiency of electricity generation and distribution (see Table B.3 in the Transportation Energy Data Book in the reference given following Table 4.1) so the appropriate conversion factor is 11.8 BTU/W-hr. This example shows an energy intensity of 1180 BTU/pas-mile and Table 4.1 shows how this compares with energy consumption for various rail and bus modes.

Table 4.1. Comparison of energy usage of various transportation modes.

	Energy usage, 10^{12} BTU	*Average trip length, miles*	*Energy Intensity, BTU/pas-mi*
M3			1,180
Intercity rail	23.0	238	4,137
Commuter rail	25.9	22.8	2,717
Rail Transit	48.6	5.1	3,114
Intercity bus	32.3		932
Transit bus	91.6		4,125
Certified air carriers	2,599	842	3,968
Autos	9,100	9.1	3,588

Data is taken from Tables 2.5,2.12-13, 9.2,9.12-14 in Transportation Energy Data Book: Edition 23 (Oct. 2003, Oak Ridge National Laboratory, available at www-cta.ornl.gov/data). Data is for 2001 except intercity bus mode data is for 2000.

Care must be taken in interpreting Table 4.1 because of wide variations within any mode. For example, the energy intensity for light rail varies from less than 2,000 BTU/pas-mi for Newark to more than 8,000 BTU/pas-mi for Cleveland. The conclusion is that *M3* has the potential to reduce energy consumption to less than half that of other modes and, if people will use maglev instead of a car or airplane the savings are even greater. If an *M3* system were operated with the same maximum speed and stopping frequency as intercity bus, *M3* would have essentially the same energy intensity. In summary, maglev can offer significant energy savings, particularly in comparison with the modes most used in the U.S. today.

4.8 LSM manufacturing

The design of the LSM suspension rails and winding has been completed and we have studied several methods for installing the components at the lowest possible cost. It is anticipated that we will manufacture the lamination stack and coils for both straight and curved (i.e. standard radius) sections of guideway as an environmentally sealed unit. The stator will be mounted under the suspension rails. We are developing improved ways of mounting the LSM stator to the guideway. Our effort has concentrated on reducing the amount of alignment that has to be done in the field.

5 Guideway

The focus of the *M3* design effort was to keep the guideway beams as small and light as possible without jeopardizing ride quality. The resulting design is based on deflection considerations, and the strength of the structures is far greater than is necessary so there is no compromise with safety. The relatively small size of the guideway is evident in the artist's rendition on the cover of this report. The pier spacing is large and the beam cross-section small when compared with virtually all other elevated transit systems. For new installation it is believed that most urban maglev systems will use elevated guideways to avoid the right-of-way access and safety problems of at-grade guideways or the cost of tunnels. Maglev vehicles make no wheel or engine noise and very little wind noise at speeds suitable for urban transportation. Many of the objections to elevated guideways are ameliorated by the *M3* design.

In some cases Urban Maglev will operate at-grade or in tunnels and in these cases the beams can have a smaller height with more frequent supports, but the design principles are the same. For example, a reduced height beam could be mounted directly on railroad ties to replace rails in a rapid transit retrofit.

Guideway cost is a dominant item so considerable effort has been made to reduce cost by reducing size and weight. The following sections discuss some of the key details of the guideway design.

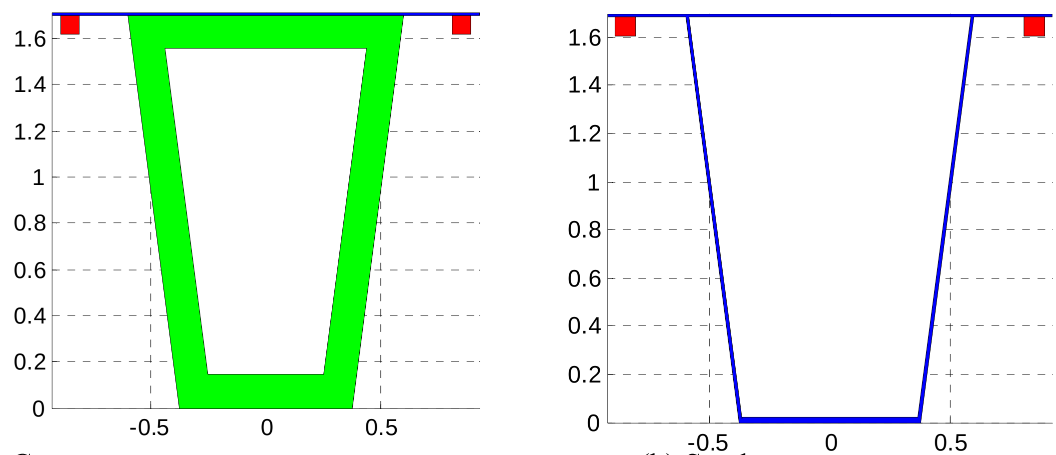
5.1 Beam design

With EMS designs the vehicles must either be supported by an overhead rail or use a monorail type of construction with the vehicle wrapped around the beam and magnets moving under the suspension rails. The overhead design could be useful for indoor use, but is not considered desirable for outdoor use because of the high cost of a support structure and poor ride quality in the presence of high winds.

The MagneMotion guideway consists of beams mounted on piers spaced 36 meters apart. This spacing was determined by an iterative process that considered the tradeoff in cost between using more piers and a lighter beam vs. fewer piers and a heavier beam. An additional consideration is a preference to use longer pier spacing because then there is less visual impact. For comparison the new Shanghai Transrapid installation uses a pier spacing of 24.8 meters but even with this shorter spacing the beams have a mass of 190 Mg so a longer span is not practical. The New Millennium extension of the Vancouver Skytrain uses 37 meter spacing.

It would be possible to make the beam length equal to the pier spacing, but there are major advantages of using a double-span beam. In this case a double-length beam is supported in the middle with a rigid mount and at the ends with a sliding mount. When the temperature changes the beam will change length and slide on the end mounts and enough space is allowed so that adjacent beams never touch. The distributed nature of the suspension magnets allows gaps of 20 mm to be easily bridged. As compared with a single-span beam with the same pier spacing, the double-span beam offers a 30% reduction in static deflection as well as a reduction in dynamic deflection, even though the lowest resonant frequency is the same. In some cases it may be necessary to use single-span beams and then a somewhat large section will be used to maintain adequate stiffness.

Three alternate sections have been studied for the guideway beams: a steel box girder, a concrete box girder, and a hybrid design that uses a concrete box girder with a composite steel top plate. The sections for concrete and steel are shown in Figure 5.1 for the case of a 1.7 meter gauge or distance between support rail centers. The top plate used for supporting the rails is only schematic with details of the method of attachment still to be completed. Alternatives being studied include cross-ties, similar to a railroad, and a composite plate with integrated attachment points. The hybrid design is similar to the concrete design except that steel cross-ties used to support the suspension rail are replaced by a solid steel top plate bonded to the concrete beam. With the hybrid design the steel that supports the rails also contributes to reducing guideway deflection and increasing the resonant frequencies.



(a) Concrete (b) Steel

Fig. 5.1. Alternate beam section with dimensions in meters.

For all of the alternates a two-span continuous girder configuration is used. Horizontal restraint is provided at the interior pier with fixed bearings in the case of the steel alternate, and with a monolithic connection in the case of the concrete alternate. The monolithic connection will use the additional stiffness of the pier to increase the overall stiffness somewhat, and is an economical means of making the connection.

These sections were incorporated into single and double guideway designs. The geometry of the design is dictated by dimensional constraints on the beam, and its connection at the column, which are imposed by the attractive maglev system. A relatively narrow girder is required because of the need for the magnetic pods to wrap around and under its edge. In addition, ride-quality considerations and deflection tolerances suggest a relatively deep girder. Together these two requirements result in a fairly deep narrow girder for which stability must be provided by external diaphragms. Since 4 meters (i.e. from beam center to beam center) will separate the two double-track guideway beams, intermediate diaphragms between the girders are not desirable (though they may be necessary in seismic zones). Stability is therefore provided by "outrigger" diaphragms at the bearings, or by a monolithic connection to the column. Diaphragms must be kept out of a zone of roughly half a meter below the top of the girders in order to allow the magnetic pods to pass.

5.2 Beam statics and dynamics

Guideway beams are designed on the basis of stiffness, not strength. Almost any design that gives good ride quality will be capable of carrying much higher loads than the maglev vehicles will create. The extra strength means, for example, that heavy maintenance or rescue vehicles could safely operate on the guideway if they operated at reduced speed. Since the key issue is ride quality, the important parameters are the guideway deflection under static and dynamic loads and due to thermal deflection and creep. Since beam cost is very nearly proportional to weight, the problem is to design a beam that is as light as possible but provides good ride quality.

For this discussion static deflection is defined as the deflection of the center of the beam when vehicles move across it at a low speed. Dynamic deflection is defined to be the extra deflection that occurs because of resonances in the beam. Although the beam has an infinite number of resonant frequencies, only the first one or two contribute significantly to vehicle ride quality. The peak dynamic deflection can never exceed the peak static deflection but it can have a major effect on ride quality. We can use precamber of the beam to compensate for nominal vehicle mass but cannot use it to compensate for dynamic deflection.

5.3 Comparison of steel and concrete beams

This discussion assumes beam properties given in Table 5.1 for a 1.5 meter gauge. We now believe a 1.7 m gauge is a better choice and Fig. 5.1 is based on this wider gauge. The main effect of increasing the gauge is to make the beam stiffer and provide a wider base for supporting the vehicle on turns.

Table 5.1 Properties of guideway beams for a 1.5 meter gauge.

	Concrete	*Hybrid*	*Steel*
Density (kg/*M3*)	2,400		7,860
Elasticity (E, GPa)	30		207
Top thickness (mm)	145	15	15
Side wall thickness (mm)	145	145	13
Bottom wall thickness (mm)	145	45	19
Mass (kg/m)	1,767	1,804	751
EI (N-m^2)	5280	7140	5480
Area (m^2)	0.6149	0.6345	0.0765
I (m^4)	0.1755	0.2421	0.0293
Static deflection (mm)	8.84	6.3	8.51
Thermal gradient deflection (mm)	3	7	9
Creep deflection (mm)	2	2	0
f_1 (anti-symmetrical, Hz)	2.09	2.43	3.44
f_2 (symmetrical, Hz)	3.27	3.80	5.38

All beams are for a double-span beam 72 m (236’) long and 1.6 m (58”) high and 1.5 m (59”) gauge. For a single span beam the static deflection is 30% higher.

5.3.1 Live load deflections

Deflection and ride-quality considerations, rather than strength, governed the design of both the steel and concrete alternates. Both are operating well below their safe load-carrying capacity. The quantity EI (elasticity times moment of inertia) for the steel alternate is 5500 $MN{\cdot}m^2$, while that for the concrete alternate is 5300 $MN{\cdot}m^2$. The live load deflections for the two are 10 mm and 9 mm respectively. The live load deflection of the concrete alternate is slightly lower, even though it has smaller section stiffness, because of the monolithic connection at the interior pier. Preliminary estimates for the dynamic amplification of the vehicle loading were 20%. A value of 20% has been used in these deflection calculations, and that value will be updated as the design of the system progresses. Structural damping for either alternate will be very small, on the order of 1% or 2%, and will do little to reduce the immediate dynamic effects of vehicle loading, though it will have an important effect on the time the guideway continues to vibrate after the vehicle has passed.

The hybrid section is envisaged as the concrete section with a steel plate attached at the top with sufficient shear-flow capacity to make the plate act compositely with the concrete. The plate would be made thick enough to support the vertical load from the vehicle and windings in transverse bending to transfer it to the girder. It would not be considered for the strength design for longitudinal bending. The increase in stiffness would only be considered for the reduction in deflection that it would provide. The section stiffness EI for the hybrid section is about 7140 $MN{\cdot}m^2$, which would reduce the live load deflection to about 7 mm.

The live load performance of the steel and concrete alternates are very similar. The steel alternate exhibits a lower dynamic response because it has a lower mass for the same stiffness, which results in a fundamental period that is significantly shorter than the transit time of the vehicle. The monolithic connection that is possible for the concrete alternate helps to increase its stiffness and compensate for the fact that it's fundamental period is closer to the transit time and therefore increases its dynamic response. That is, even though its dynamic deflection is greater, its static deflection is less, such that the total deflection is about the same. It is important to note that it is the vertical acceleration of the vehicle that is important, and not the dynamic deflection of the guideway. The total live load deflection, static plus dynamic, is a good proxy for vehicle acceleration, since the vehicle has to travel vertically from zero to the full deflection and back in the time it takes to cross the span. Since the total live load deflections are very similar, the ride quality will be similar as well. The hybrid alternate will have the best live load performance because it has the smallest total live load deflection.

The live load deflections in curved spans will be greater than those in tangent spans because of the additional component from twisting. Moreover, the twist itself will be undesirable if it becomes too great. Deflections were computed for a 20 m span on an 18.3 m radius to determine the possible extent of this problem. A vertical deflection of about 6 mm was found, which is less than that for the typical tangent span, owing to the reduced span length. The maximum twist results in a difference in elevation between the inner and outer suspension rail of about 8 mm, which is within acceptable limits. For comparison, the difference in elevation from a 6° superelevation will be 157 mm.

5.3.2 Thermal deflections

The deflection under live load is not the only consideration for ride quality. Thermal gradients will also contribute to the total deflection. At this stage only vertical thermal gradients have been studied. The effects of horizontal thermal gradients will be considered at a later stage of the study. The thermal gradient for the steel cross section was taken from the Federal Railroad Administration Report No. DOT/FRA/ORD-94/10, Safety of High-speed Magnetic Levitation Transportation Systems. The thermal gradient for the concrete alternate was taken from the American Association of State Highway and Transportation Officials (AASHTO) Load and Resistance Factor Design (LRFD) Bridge Design Specifications, 2002.

Analyses of the sections for thermal gradients show that the steel alternate exhibits significantly higher deflections under this effect, with an upward deflection of about 9 mm. The upward thermal gradient deflection for the concrete alternate is about 3 mm. The peak temperature for the thermal gradient for the steel structure is higher than that for the concrete, as would be expected. However it is only slightly higher and the differences in temperature alone cannot explain the large difference in the thermal gradient deflection. The principal cause of the difference is the difference in section geometry. Since the temperature gradient is very steep at the top slab, the thickness of the top slab of the concrete alternate results in an average temperature in the slab much less than the peak temperature at the extreme fiber. In contrast, the entire thickness of the top plate of the steel alternate is effectively at the peak temperature, and therefore tends to cause a much greater curvature in the steel section.

Reflective coatings and/or insulation may be used to reduce the temperature peaks at the surface of the section to reduce deflections. Such treatment would be effective for both concrete and steel alternates, but would obviously be more worthwhile for the steel alternate. Of course the addition of insulation will have cost implications and perhaps maintenance implications as well.

The situation for the hybrid alternate is not as clear. The temperature in the top plate of this section will probably be higher than what was found for a hollow steel girder, since it will tend to be insulated below by the concrete. However it is probably also safe to assume that the average temperature in the concrete top slab will be less than it is in the concrete alternate. In the absence of any data on the temperatures in such a structure, we have assumed that the steel plate will see the same temperature as the top plate of the steel alternate, and that the temperature changes in the concrete portion of the section will be negligible. In response to such a loading, the deflection of the hybrid alternate is midway between those for the concrete and steel alternates.

5.3.3 Long term deflections

The effect of long-term concrete deformations must also be added to the deflections from thermal gradients in order to get a meaningful comparison of the deflections affecting ride quality for the three alternates.

The long-term deflections of concrete can be separated into two components: shrinkage and creep. Shrinkage occurs independently of the applied loads and it does not tend to cause deflections in the superstructure, except for the secondary effect that it has on pre-stressing loss and the effect it has on column shortening. Creep occurs in response to a sustained applied load and tends to increase deflections that exist in the structure from those loads.

Both steel and concrete alternates will experience deflections due to shrinkage shortening of the concrete column over time. The magnitude of the deflection will depend on the ambient humidity, curing practices and the height of the column. Strains can vary from about 200 to 500 microstrain. For "average" conditions, shrinkage strains that occur after erection of the superstructure on the order of 200 microstrain can be expected. For a 15 m high column (column + foundation shaft), a deflection of about 3 mm results. Column shortening, however, will not affect the ride quality except at abrupt changes in column height, such as stations and abutments where the difference in deflections between adjacent columns is large. Shrinkage will not otherwise affect the deflections in the superstructure, except that it will contribute to the loss of prestress in the concrete alternate, which will have some small effect of the pre-stressing deflection.

The creep strain is proportional to the stress in the structure under permanent loading. Permanent loading includes the girder self weight, the superimposed loads from the windings and their supports, and pre-stressing. Since pre-stressing will tend to cause curvatures in the opposite direction from dead load bending, the creep deflection from pre-stressing will counteract creep deflection from dead loads, just as elastic deflections from pre-stressing will counteract elastic deflections from dead load. Since the creep curvature will depend on the total moment on the section over time, it is convenient to think about a "creep-inducing moment" which is the difference between the pre-stressing moment and the dead load moment. If the pre-stressing moment is identically equal and opposite to the dead load moment along the entire structure, the creep-inducing moment will be zero and therefore the vertical creep deflection will be zero. A pre-stressing design that creates moments equal and opposite to the dead load moments is generally referred to as a "balanced" design, in other words the pre-stressing balances the dead load.

It is generally not practical or economical to exactly balance the dead load with pre-stressing. However, in the case of the *M3* system the unusually light live load and girder make it possible to design pre-stressing that is very close to balanced without an excessive economic impact.

5.3.4 Total deflections and camber

For both the steel and concrete alternates, it is possible – and necessary – to camber the girders in anticipation of service deflections. Both alternates will have to be cambered for dead load deflections. The concrete alternate would also have to be cambered for creep so that, with time, those deflections will tend to bring the riding surface closer to flat and level instead of tending to increase the deflections.

Typically for roadway, and even light rail bridges, the girders are cambered to end up "flat" under permanent load deflections. It is also possible, however, to consider cambering the beams for live and thermal deflections. This is especially interesting in the case of *M3*, since cambering to counteract live load deflections would reduce the vertical acceleration of the vehicle as it crosses a span and improve ride quality. Though such camber would be beneficial, it will take careful study to determine what the actual optimum camber would be, since it would be necessary to consider the transient nature of thermal deflections, and the fact that the weight of the vehicle is not constant.

The worst-case deflection for ride quality could be some combination of all of the above-mentioned deflections – or no net deflection at all. For example, in the case of the steel alternate, the thermal gradient deflection is a positive 9 mm, while the live load deflection is about equal and opposite to that value. If the vehicle should pass at a time when the full thermal deflection is present, the net result would be that the total deflection due to live load and thermal gradients would be zero, which would be beneficial to ride quality. In this scenario, it is clear that cambering the girder upward to completely counteract live load deflections would be counterproductive.

Note that negative thermal gradients can exist in bridge girders. Such negative gradients for concrete bridges are presented in the National Cooperative Highway Research Program (NCHRP) Report 276, Thermal Effects in Concrete Bridge Structures. Though there is no similar report for all steel box girders to the authors' knowledge, certainly such gradients exist, and present a topic for further study.

Assuming for the moment that the deflection due to a negative thermal gradient is equal to the opposite of half the deflection from positive gradient, it may be that the optimum camber for the steel alternate is some compromise value. A value of about half of the live load deflection (plus the dead load deflection) would be appropriate as a first estimate. This value should be adjusted based on the expected fraction of the time that the thermal gradient deflections exist and their correlation with the operating hours of the system and the expected total vehicle weight during those hours.

In the case of the concrete alternate, creep and thermal deflections would have to be considered as well when figuring live load camber. The creep deflection should be considered from the time that the suspension rail is installed, because the installation will account for creep occurring before that time. In our study this so-called "service creep" results in sag, though that would not necessarily always be the case. Typically the girder would be cambered to arrive at a "flat" condition late in service when creep has run its course. However since the creep deflection could potentially be positive or negative and it has to be considered in conjunction with the thermal deflections, it is not immediately clear that the same approach would be appropriate for the *M3* system, and the question needs further study. Assuming that service creep results in a sag of 3 mm, and positive thermal gradients result in a hogging that is roughly equal and opposite, something close to full camber to compensate for live load deflections would be appropriate. Table 6.2 shows deflections due to creep, temperature and live loads.

Table 6.2. Deflections due to creep, temperature and live loads.

	Creep Deflection	*Peak Temperature*	*Thermal Gradient Peak Deflection*	*Live Load Deflection*
Concrete	-3 mm	23 °C	3 mm	-9 mm
Steel	N/A	30 °C	9 mm	-10 mm
Hybrid	-2 mm	30 °C	7 mm	-7 mm

5.3.5 Horizontal deflections

The guideway will be subject to horizontal deflections during operations from wind, live loads and thermal loads, which will also affect the ride quality. Horizontal thermal gradients require further study, as discussed above. Horizontal deflections from wind and live loads can be broken down into a guideway-beam component, which comes from horizontal bending in the beams, and a pier component, which is the result of bending in the pier and rotations in the foundation.

In the case of wind, there will be a dynamic structure response, and it will depend on the wind speed, gust characteristics, and the geometry (drag coefficient and natural frequencies) of the guideway. A complete analysis that considers all of these factors is beyond the scope of the current work; however, as a first approximation it is possible to calculate static deflection based on the wind loads given in the AASHTO code. For a 100 mph wind, AASHTO gives a net total pressure of 50 psf (2.4 kPa). Applying this load to the windward beam only, a total deflection of 60 mm is found. Approximately 40 mm of that deflection comes from column bending. There will also be a load on the leeward beam, which is not considered in the above numbers. AASHTO stipulates a value of half that for the windward chord for leeward truss chords. If that loading is used for the leeward beam, the pier deflection will increase by about 50% to 60 mm. This is a relatively large deflection, and it may dictate the use of stiffer substructure elements, though horizontal deflections at the top of the pier of up to 100 mm have been allowed for some light rail systems.

Assuming that the vehicle will only operate at full speed in winds of 50 mph or less, the deflection used for assessing the ride quality can be reduced. Since the wind pressure is proportional to the square of the wind velocity, the deflections will be one quarter of those given above, i.e. 15 mm for the piers and 5 mm for the beam.

5.3.6 Ride quality considerations

The static and dynamic deflection under live load are major considerations for ride quality, but they are not the only ones. Thermal gradients and long-term material deformations (creep and shrinkage) will also contribute to the total deflection. At this stage only vertical thermal gradients have been studied. The effects of horizontal thermal gradients will be considered at a later stage of the study. The thermal gradient for the steel cross section was taken from the Federal Railroad Administration Report No. DOT/FRA/ORD-94/10, "Safety of High-speed Magnetic Levitation Transportation Systems…" and is given in the appendix. The thermal gradient for the concrete alternate was taken from the AASHTO LRFD Bridge Design Specifications, 2002.

In the concrete alternate, the combination of deflections that would cause the worst ride quality is probably creep and live load deflection. The total deflection of live load alone is greater than the combination, since the creep deflection is positive at the point of maximum live load deflection; however, the creep deflection causes double curvature in the span, which would be more unfavorable from a ride-quality perspective. The maximum change in deflection for this combination is about 6 mm and it occurs over about half a span length. Again, efforts would be made to camber the girder to help reduce this effect. A reduction of half of the creep deflection can be reasonably expected.

For both the steel and concrete alternates, it is possible to camber the girders in anticipation of shrinkage and creep deflections. That way, with time, those deflections will tend to bring the riding surface closer to flat and level instead of tending to increase the deflections. Precamber can also be used to compensate for normal live load, but variations in load and dynamic behavior cannot be compensated.

The seismic design requirements for the guideway for the *M3* system are not fundamentally different from those for other bridge structures. Ideally the foundation and superstructure will be designed to remain elastic, and the columns detailed to respond in a ductile manner, though that approach may change depending on the location of the site and the local seismic risks. Further requirements unique to the *M3* system could include maximum tolerable deflection limits during seismic response (for example angle deviations at expansion joints) or buffers to prevent the vehicle from locking up and stopping too suddenly if it bumps up against the guideway. Such additional safety considerations will have to be addressed as the mechanical systems for the vehicle and suspension rail are developed further.

Ride quality is often measured by plotting a spectrum of vertical acceleration vs. frequency (for a vehicle moving along the guideways) and comparing that with an empirically derived limit, such as the International Standards Organization ride quality standard shown in Fig. 5.2.

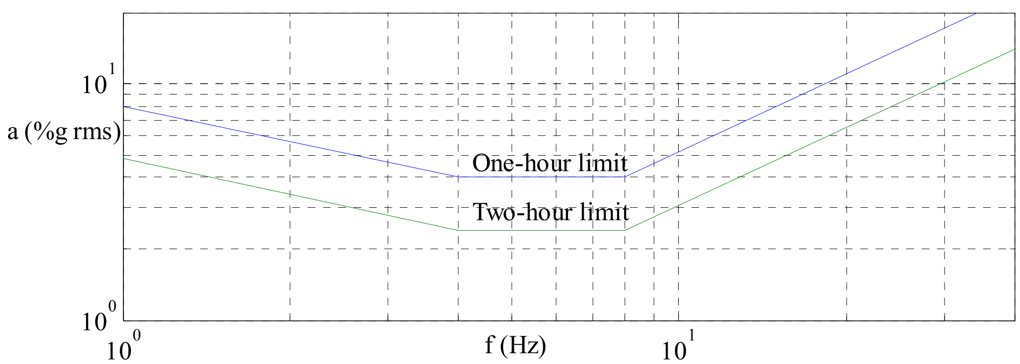


Fig. 5.2. ISO Standard for acceptable vertical acceleration for good ride quality.

The problem of achieving good ride quality is particularly difficult at high speed because the vertical acceleration tends to increase as the square of the speed. Figure 5.3 shows the computed vertical acceleration spectrum for the baseline vehicle traveling along the concrete guideway at 45 m/s. This simulation assumes full live load deflection with no precamber and no deflection reduction due to the attachment of the beam to the middle pier, but it neglects deflection due to creep and thermal effects. The blue (left) spectrum is for the front of the vehicle and the red is for the rear.

The vertical spectrum is dominated by the pier-crossing frequency, 1.25 Hz, and the lowest resonant frequency of the beam, 2.4 Hz, modulated by the beam crossing frequency. There is also some response near the higher frequency beam resonances: 3.8 and 9.8 Hz. Particularly noteworthy is the low amplitude of high frequency components, a result of the distributed nature of the magnetic suspension. More detailed and accurate simulations will be done in later phases of *M3* development, but it appears that the *M3* vehicle can have dramatically better ride quality than the ISO limits given in Fig. 5.2.

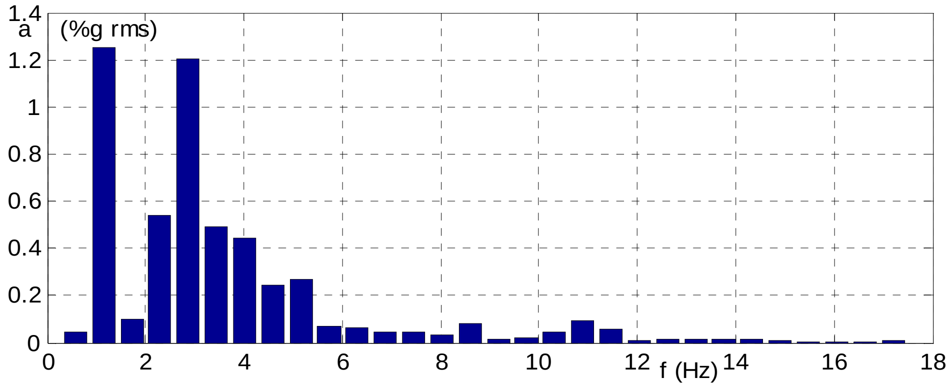


Fig. 5.3. Vertical acceleration spectrum at 45 m/s on concrete guideway.

5.3.7 Longitudinal design

Because the guideway design is controlled by deflections, the stresses from service loads are very minor. In the steel alternate the total service stress is less than 20% of yield, and the live load stresses are almost insignificant. The situation is much the same for the concrete alternate, though the stresses are somewhat higher in relation to allowable limits, at least for shear. In the case of a guideway on an 18 m horizontal curve, the shear stresses from shear and torsion are still well within acceptable limits. For example, in the case of the concrete alternate the principal tensile stress in the web during service loading is less than one MPa (a value of about 1.5 would be acceptable). The shear stress in the steel alternate is even less compared with the allowable value.

5.3.8 Seismic design

A seismic analysis of the system following the AASHTO LRFD specifications was performed on the two alternates to verify the preliminary member sizes. A peak rock acceleration of 0.4 *g* and a soil coefficient of 1.0 were assumed, indicating a high seismic zone on firm soil or rock. The foundation was assumed to be a 1.8-meter-diameter single drilled shaft. This foundation type has been gaining popularity in California because of its excellent seismic performance and simplicity of construction. Because continuous girders are restrained longitudinally at the center column and left free to expand at the outer columns, a single column resists the longitudinal seismic actions. During transverse seismic response, the girders span horizontally between columns so that all columns are acting. Therefore the longitudinal response controls the design of the column. At some point it may be worthwhile to consider pinning the superstructure at one outer column in high seismic zones to allow all of the columns to participate in the longitudinal seismic response, however it is beyond the scope of this report to develop the special details required for such a connection.

A 1.0-meter diameter column was chosen for the steel alternate. This alternates lightweight results in lower seismic demands and allows for the use of a smaller foundation and substructure elements. The results of the preliminary seismic analyses indicate that this column and foundation are adequate for the loads suggested by the AASHTO specification.

The key point to consider for the steel alternate superstructure with regards to seismic performance is that it will be supported on bearings. The tall narrow cross section will have to be stabilized either by providing tie-down bearings, diaphragms between the girders at the piers, or by devising a continuous connection between the girders and the hammerhead. Adequate bearing-seat width and restrainers will be provided to prevent loss-of-support failures.

A 1.2-meter diameter column was used for the concrete alternate. With the greater mass of this alternate, the larger stiffness keeps the displacement at acceptable levels. The same diameter drilled shaft is used, though it will require a greater steel content and penetration into the founding rock or soil.

Stability is provided by the monolithic connection at the interior pier. As with the steel alternate, shear keys, adequate seat width and restrainer cables will be required to maintain support at the expansion piers.

5.3.9 Conclusions and recommendations

All of the three alternates developed in this preliminary study would be acceptable for the guideway for the *M3* system. An effort was made to achieve approximately equal performance between the three alternates so that the cost comparisons would be meaningful. The authors believe that this has been achieved more or less, though the different characteristics of concrete and steel have made exact equality impossible. The one that could possibly be called an outlier is the hybrid alternate, which has significantly greater stiffness and therefore lower live load deflections. It was necessary to develop it in this way, however, due to the nature of the construction technique envisaged.

It is difficult to decide which alternate would be preferred and probably impossible without knowing the site, length of the project, and local construction conditions. In general it is reasonable to conclude that the concrete alternate will be the least expensive by a significant margin. It is also likely that it will require the least maintenance and have the lowest lifecycle cost. This will be borne out in most locations in the United States, though there may be some places where steel may be less expensive because of local contractor experience and availability of the materials.

It is difficult to say which alternate will perform the best in terms of ride-quality. The live load deflection response is similar for steel and concrete, with the concrete having a slightly greater dynamic response but a smaller static response. The total deflection of the hybrid is the lowest, giving it the best live load behavior. The price premium of the hybrid alternate over the concrete alternate is essentially paying for improved ride quality.

The thermal deflection of the steel alternate causes its greatest performance problem. Though the vertical gradient causes a significant deflection, the deflections from horizontal gradients are likely to be a worse problem and need further study. We believe that insulation and reflective coatings will solve this problem, but at some as yet unknown increased cost.

Creep deflections are the greatest concern for the concrete alternate, and although they could theoretically be limited to acceptable levels, uncertainties about correctly predicting them add a greater risk to this alternate. The hybrid alternate faces the same construction risk from creep, and the additional uncertainty about our ability to attach the top plate with sufficient stiffness and strength. It is expected that it will be possible to construct all of the alternates within adequate dimensional tolerances. Although past experience has shown that the tolerances actually achieved for both steel and concrete may be at the limit of what is needed, it should also be recognized that the current technology has been developed to deliver only the tolerances that have been required by road and rail bridges, and it should be expected that improvements for both materials can be realized if required for the *M3* system.

Based on our findings to date, there are potentially considerable advantages to the concrete alternate for its lower cost and its thermal-deflection characteristics. Its principal detractions are the uncertainties involved with creep deflections and the attendant construction risk. The advantages are significant enough though that it is advisable to construct a test segment to quantify and understand the risks. We recommend therefore that a prototype system of limited scale be built using concrete. Depending on the scale and end use of the system, it may be appropriate to build all of it, or only a portion of it in concrete, and the rest in steel. A cast-in-place structure would probably be the best choice, as it would be the most appropriate for a guideway of limited length and would still allow us to study the dimensional stability of concrete.

Several other important issues remain to be studied at this stage. Horizontal deflections from thermal gradients will likely cause deflections equal to or even larger than the deflections from vertical gradients. Since horizontal accelerations are more disturbing to passengers than vertical accelerations, this is an important area to study. Work needs to be done to determine the horizontal gradients that will exist in concrete and steel structures, and change the cross sections if necessary, to limit the deflections set up by these gradients. We recommend that both the steel and the concrete alternates be advanced through this next stage.

Likewise, it will be important to consider horizontal accelerations from wind and live loads. The effect on those accelerations from deformations in the substructure should be considered in evaluating these effects. Work to be done includes determining the aerodynamic properties of the cross section, the response of the structure to a generic wind climate, and the effect of wind oscillations on vehicle performance. For live loads, additional rolling-stock type analyses should be performed for various guideway parameters to determine acceptable limits for foundation stiffness, pier height, etc. Work to optimize the span lengths for curves of various radii would also be warranted.

The mounting hardware for the suspension rails is probably the most important item to develop. It needs to allow for easy adjustment both horizontally and vertically. If it were possible to develop hardware that allows for rapid and economical readjustment, then it would be possible to assume more risk in the dimensional stability of the initial construction, and would help greatly in developing the lowest cost guideway.

5.4 Ride quality analysis of baseline design

The concrete box beam guideway has been determined to be the lowest cost method of building a guideway that allows good ride quality for our baseline vehicle operating at speeds up to 45 m/s (101 mph). The focus of this Section is to discuss ride quality for vehicles moving on concrete guideway beams.

5.4.1 Concrete beam properties

Vehicle considerations have caused us to increase the gauge of the suspension rail from that of the original design. The new gauge is set at 1.7 m suggesting a somewhat wider beam. The ride-quality studies presented herein consider a beam with dimensions shown in Fig. 5.4. The width may change as details of the suspension-rail mounting hardware are finalized. The span has been kept at 36 m. Since the target of this analysis is a test track, the structure has been simplified to a two-span continuous single-track guideway with simple supports. No column is considered in the analyses since the test track will be supported on zero-height piers. Other beam data are included in Table 5.3. Note that a mass of 150 kg per meter has been added to account for the suspension rail and mounting hardware.

Table 5.3. Concrete beam parameters.

Concrete elastic modulus	30,000	MPa
Beam inertia	0.2067	m^4
Beam area	0.673	m^2
Concrete specific weight	23.56	kN/m^3
Total beam weight	17.3	kN/m
Beam natural frequencies	2.27, 3.55, 9.08	Hz

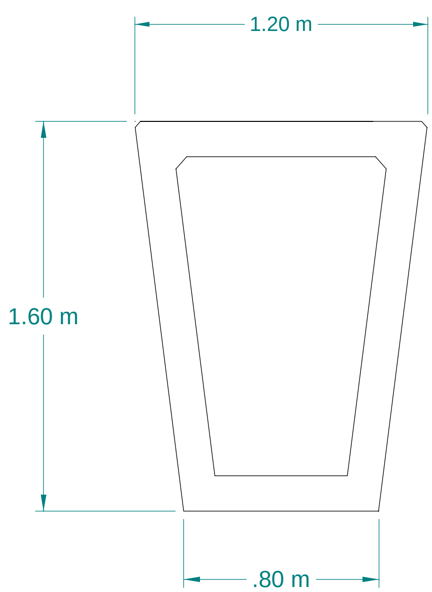


Fig. 5.4. New guideway beam for increased gauge.

5.4.2 Rigid vehicle analysis validation

Data for the *M3* vehicle at the target cruising speed are as follows:

Vehicle velocity………………...45 m/s
Vehicle mass…………………..……7 Mg
Vehicle length…………………..….10 meters
Pier-crossing frequency ……….1.25 “Hz” (45 m/s ÷ 36 m)
Beam natural frequencies……….2.27, 3.55, 9.08 Hz

These parameters are used throughout as the baseline case for ride-quality studies, as in the previous work.

Our previous analyses of ride quality have been done in the time domain, with the load of the vehicle traversing the beams represented by nodal loads along the beam. These nodal loads were assigned to time functions, which ramped them up when the vehicle approached a node and back down after it passed. The movement of the vehicle was thereby modeled as a wave of nodal loads along the span.

While this methodology worked very well for capturing the dynamic behavior of the beam, as would be needed for a stress analysis of the beam, it was not sufficient to directly describe the quantities needed for a ride-quality assessment. This is because the vertical acceleration of the car body, which is the quantity of interest for studying ride quality, is made up of several components. Among these are:

- The vertical acceleration of the supporting beam.
- The vertical movement of the vehicle as it follows the curve of the beam (deflection plus fabrication curve).
- The movement of the car body relative to the beam, i.e. compression and extension of the suspension.
- The vibration of the car body itself.

The results of ride quality analysis are shown in Figures 5.5-6.

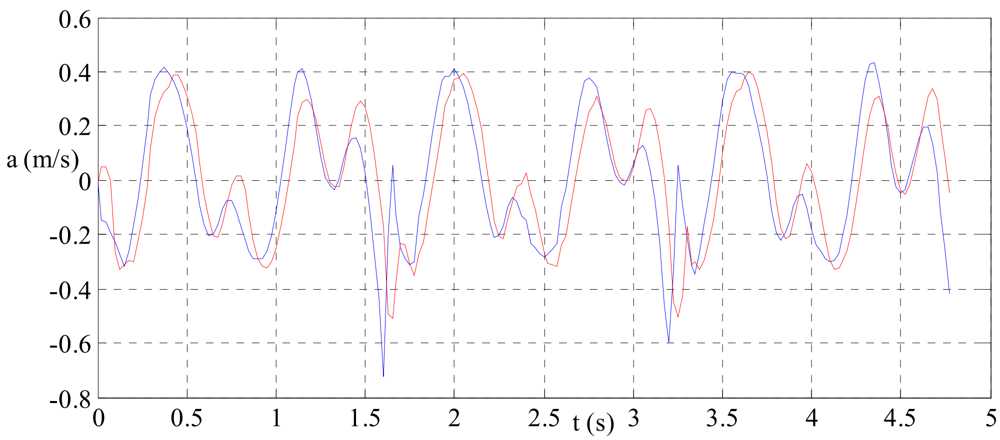


Fig. 5.5. Vertical acceleration of a rigid vehicle at 45 m/s.

The red curve is for the back of the vehicle and the blue for the front.

The corresponding acceleration spectrum is shown in Figure 5.10.

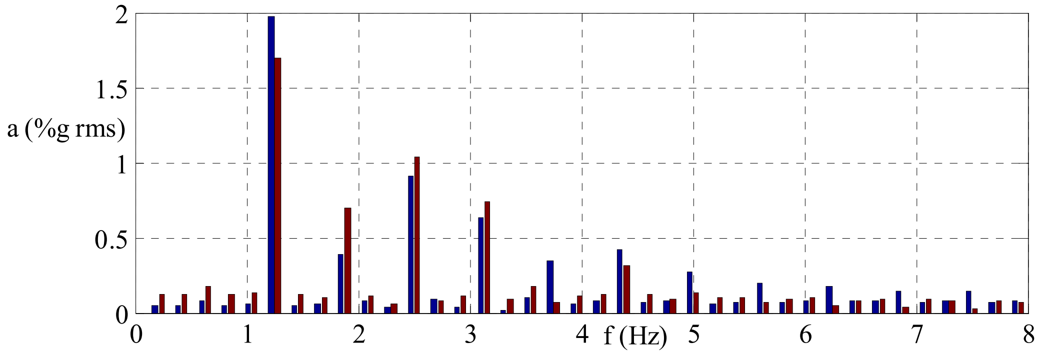


Fig. 5.6. Vertical acceleration spectrum for rigid vehicle at 45 m/s.

5.4.3 M3 *vehicle suspension*

The function of the suspension is to isolate the car body from the irregularities of the guideway. Ride comfort is a function of high frequency vibrations, vertical spring actions or "bounce", and body pitch and roll that the car body experiences in crossing irregularities in the guideway. High frequency vibrations relate to the vibration of the car body itself, which is beyond the scope of this study. Body roll involves 3D behavior of the vehicle through horizontal turns, which is not part of this study either. It is the vertical "bounce" and the pitch of the vehicle as they vary with time that will be considered here.

Vertical accelerations can produce rider discomfort in different ways, depending on their frequency content. Low frequency vibrations, in the range of 0.5 to 0.8 Hz are usually associated with motion sickness. Moderate frequency vibrations in the range of 5 to 6 Hz affect the visceral regions. Higher frequency vibrations of up to 20 Hz are bad for the head and neck. Most vehicle suspensions target a natural frequency of 1 to 2 Hz because that is roughly the frequency of walking, and is believed to be the frequency at which the ride will be most comfortable, though designers have started to question this approach.

In the current design of the *M3* system, the baseline vehicle velocity is 45 m/s and the beam span is 36 m, resulting in a pier-crossing frequency of 1.25 Hz. Since this essentially represents the frequency of a forcing function, it would be expected that much of the car body acceleration would occur at this frequency.

The fundamental natural frequency of the current beam design is 2.27 Hz. This corresponds to a two-span beam with three simple supports. (As indicated previously, a monolithic connection to the column is not considered, which is a departure from the previous study.)

The natural frequency of the vehicle would ideally be within the 1 to 2-Hz frequency range as well, but different from the pier-crossing frequency and the natural frequency of the guideway beam. The vehicle has therefore been sprung such that the frequencies of the first two modes are roughly 1.4 and 1.8 Hz. The rear spring has been made nearly twice as stiff as the front so that the vehicle is not symmetrical front to back. This avoids having pure bounce and pitching modes. Instead the first mode is a "Front-End-Bounce" (FEB) mode, with little movement in the rear suspension. The second conversely is a "Rear-End-Bounce" (REB) mode. Figures 5.7-8 show the key ideas.

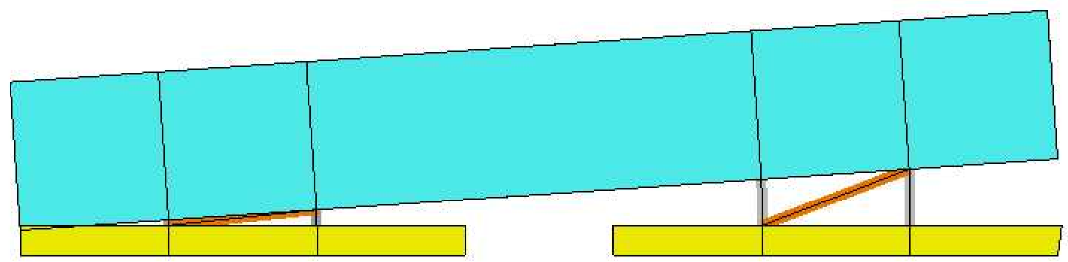


Fig. 5.7. Vehicle front-end-bounce mode – f = 1.41 Hz.

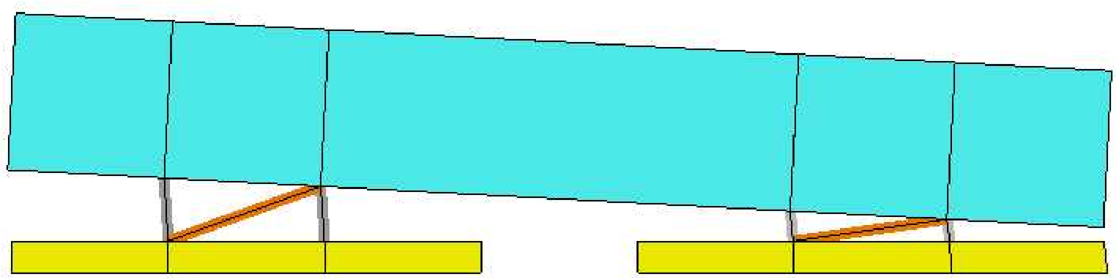


Fig. 5.8. Vehicle rear-end-bounce mode – f = 1.87 Hz.

The vehicle characteristics are preliminary at this stage. No car body flexibility is considered, the mass distribution is assumed to be uniform, and the un-sprung weight is assumed to be 20% of the total weight. A relatively high un-sprung weight is used because it was assumed that the mass of the magnets in the pods will be significant and an advanced lightweight car body will be used. The spring stiffness has been chosen to give natural frequencies comparable to those of automobiles and other transit vehicles. With the beam-crossing frequency and the vehicle natural frequencies and damping that we have, calculation of a simple single degree-of-freedom transmissibility based on the frequency ratio of the pier-crossing frequency and the vehicle natural frequency suggests that the deflections resulting from the vehicle following the sag of the beam will be amplified. This is corroborated by the numerical results, which show car body deflections greater than the pod deflections.

5.4.4 Ride quality criterion

The ride-quality standard of the International Standards Organization limits rms acceleration as a function of frequency, as illustrated in Figure 5.2. Acceleration spectra from the analysis results for zero to 15 mm camber have been compared with this standard. For cambers of 0 and 5 mm, the car body accelerations are below the limits set out in the ISO standard for two hours of exposure, indicating an excellent ride. When the camber is increased to 10 mm, the acceleration exceeds the two-hour standard at a frequency of 2.5 Hz, but is still below the one-hour-exposure standard. At 15 mm of camber, the one-hour standard is exceeded, again in the frequency range of 2.5 Hz. Above 15 mm of camber the ride is above the ISO limit, and not acceptable for long trips.

5.4.5 Increase in beam stiffness

It is tempting to increase the stiffness of the beam in an effort to improve the ride quality, since the vehicle accelerations are directly related to the beam deflections. However to do so may not necessarily be the best approach. Our current beam design delivers excellent ride quality for cambers in the range of zero to 5 mm at a maximum velocity of 45 m/s. It is the increase in camber that is causing the ride to deteriorate, not the flexibility of the beam. Since the uncontrollable camber from creep and thermal loading is not a function of beam stiffness, increasing the stiffness may not be a good way to increase the ride quality.

In the case of thermal gradients, the deflection will be relatively insensitive to increases in beam inertia, within small excursions from the current beam dimensions. Thermal deflections are more likely to be controlled effectively by providing an upward facing surface in the form of a protruding flange near the bottom slab to absorb radiation and increase the temperature at the bottom of the section, counteracting the expansion of the top slab, or to provide insulation or a reflective coating on the top slab. If the beam is made stiffer without reducing the hogging from thermal loading, the ride may actually become worse. That is because a flexible beam deflects to compensate somewhat for misalignments. This can be seen below in the results of analyses done on very stiff beams with non-zero camber. The accelerations increase with beam stiffness. This is an important topic for further study.

5.4.6 Higher speeds

The *M3* vehicle will be capable of speeds higher than 45 m/s, in the case where a particular urban alignment allows higher speeds. We have studied the vehicle response and Figs. 5.9-10 show the acceleration history and spectrum from this analysis for 60 m/s (134 mph) to see how the ride quality degenerates with speed.

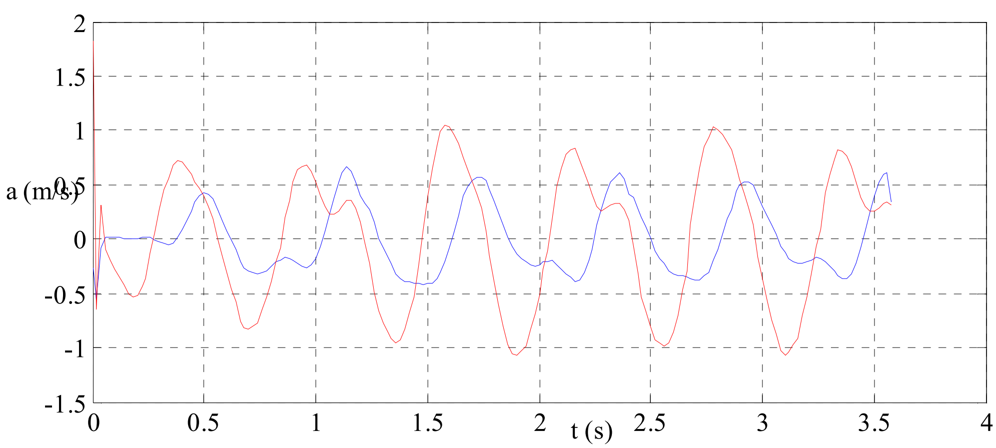


Fig. 5.9. Acceleration history for sprung vehicle at 60 m/s (134 mph).

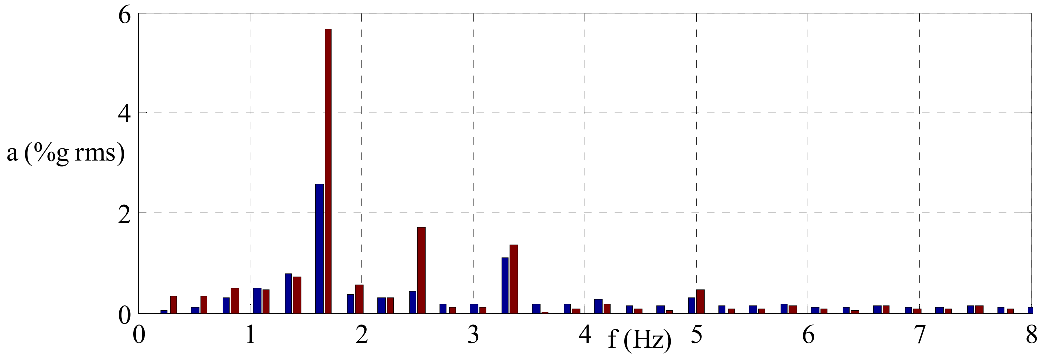


Fig. 5.10. Acceleration spectrum for sprung vehicle at 60 m/s (134 mph).

The rms acceleration at the beam-crossing frequency (now 1.67 Hz) has increased to 5.5% *g* at the rear of the vehicle. This is very near the one-hour limit of the ISO standard. For long distances, a stiffer beam and tighter suspension rail tolerances would likely be required. This could be achieved, for example, by shortening the span with the current cross section. It is unlikely, however, that trip length will exceed on hour for an urban system, and even less likely that long stretches of 45+ m/s could be achieved. It is therefore anticipated that higher speeds are possible with the current system, if careful attention is paid to the ride-quality requirements.

5.4.7 Ride quality conclusions for baseline beam

The conclusion to be drawn from this study is that the current beam design provides excellent ride quality for the vehicle speed considered, and increases in speed may be possible for short runs. It is also clear that controlling camber is as important as beam stiffness for insuring ride quality. Construction tolerances, thermal deflections, and long-term deflections are all important parameters that must be dealt with in order to achieve the ride quality desired. Improvements in the vehicle design, especially the possibility of the addition of an active suspension, will also lead to better ride quality.

5.5 Concrete beam creep considerations

As discussed in reference in the Ride Quality discussion above, camber is a very important aspect of ride quality. In the case of concrete beams, adverse camber due to creep could pose a problem, if it is not adequately accounted for in design. This task was intended to review the experience with the deflection of actual pre-stressed beams in service. Unfortunately, very little useful data have been found by the authors to date. The studies that they are aware of tend to be for bridge types that are too different in configuration and scale to be applicable to the *M3* guideway.

Nevertheless, some general information regarding the errors in deflection prediction in concrete bridges has been found. Stevula assessed the ramifications of several types of modeling inaccuracies on the predicted deflection of a large post-tensioned box-girder bridge. He found that neglecting shear deflections and shear lag in the computations was one of the most significant sources of inaccuracy. Neglect of cracking, both shear and flexural, was also cited as an important source of error. Finally, he advised careful assessment of the contribution of thermal gradients to long-term deflections. Thermal gradients contribute to long-term deflections by causing cyclical tensile strains that gradually reduce the tensile strength of concrete until flexural cracks are initiated, which then lead to greater deflection.

Favre et al. have shown that live load deflections will gradually increase with time because of degradation of the steel-concrete interface at flexural cracks. The magnitude of this degradation is a function of the amount by which these cracks move during loading, which depends on the level of pre-stressing. They report that higher levels of pre-stressing moderate this effect by keeping the cracks in compression and preventing a fatigue-like response.

Errors in the prediction of creep properties of the concrete were not cited.

We will now look at the impact of each of these issues in turn for the case of the *M3* system.

Shear Deflections are easily accounted for with modern bridge-design software, and their importance can easily be assessed for the *M3* system. With our current configuration they do not pose a significant problem.

Shear Lag is most important for box-girder structures with geometries appropriate for roadway bridges. The wide top slabs and long cantilever-slab overhangs present a longer load path between flexural normal stresses and the shear stresses at supports that generate them. Pre-stressing exacerbates this effect, as pre-stressing forces usually generate very little shear at a support and thus are usually well distributed over the section at a pier. Generally this would suggest the use of different cross-section properties for vertical loads and pre-stressing, which is seldom done in practice. In the case of the *M3* system, the width of the box is kept small by the gauge of the vehicle, and there are no long overhanging slabs, so the section will not be as sensitive to shear lag as a typical roadway-bridge cross section. Nevertheless, a careful assessment of the shear-lag effects will be made during the final design stage of an actual system.

Thermal Gradients are a cause for concern in and of themselves for the deflections they cause. The direct deflections and stresses from them have already been considered in the first phase of this project. Based on the temperature information we have, and the stresses we have calculated, there is little danger of tensile cracking from thermal gradients. The greatest concern would be that we have underestimated the temperatures in the structure from solar radiation. Our intent is to look at the applicability and efficacy of coatings and insulation as time and resources permit, as a possible way to eliminate this issue.

6 Vehicle

The MagneMotion Team has not yet developed a detailed vehicle design but the important parameters have been determined. This section discusses a baseline vehicle design that is similar to the vehicle depicted on the cover and in Fig. 2.1.

6.1 Size

European and Japanese maglev developers have always viewed maglev as a modern form of train travel with the potential for higher speeds, lower maintenance cost, etc. The German Transrapid and the Japanese and European high-speed designs all use multi-car trains with each train carrying several hundred passengers and train spacing of several minutes. In contrast, U.S. maglev developers have always thought of maglev as a form of bus or airplane with a preference for smaller vehicles operating more frequently. All 4 designs that resulted from the U.S. 1992 National Maglev Initiative recommended the use of individual vehicles with capacities less than a 100 passengers. Following are some of the advantages of each approach.

Larger vehicles or trains of vehicles operating with longer headway are most beneficial for high-speed intercity operation. Advantages include:

- Lower labor cost when operated manually;
- Higher capacity is possible;
- Lower aerodynamic drag per passenger;
- Vehicles can be less expensive per passenger.

Smaller vehicles operating with shorter headway are most beneficial for urban transit operation. Advantages include:

- Reduced cost for elevated guideways;
- Reduced waiting time for passengers;
- Reduced propulsion power per vehicle;
- Clusters of vehicles with spacing controlled electronically are more versatile than physically connected trains of vehicles;
- The number of stops can be reduced by using station skipping strategies;
- It is easier to reuse regenerated braking energy.

For *M3* there are major advantages of using small vehicles with short headway so that is the basis of the baseline design. The suspension and propulsion technology used for *M3* systems will work very well for larger vehicles or trains but the capital and operating cost will be greater. Our baseline vehicle can be configured in several ways: 40 passengers (12 seated), 36 passengers (24 seated), or 32 passengers (all seated). The key dimensions are given in Table 6.1.

6.2 Headway and capacity considerations

It is desirable to be able to operate with short headway so that high capacity can be realized with small vehicles. This is possible because accurate position sensing is inherent with LSM propulsion and braking does not depend on friction. The use of short headway allows a cluster of vehicles to act like a cluster of buses on a highway and thereby achieve high capacity. The vehicles can operate safely with only a few seconds headway just as cars operate on a highway with short headway, but there does need to be space between clusters in order to allow either in-line or off-line loading and unloading. Section 7 on Control discusses these issues in more detail but the effect of using clusters of vehicles with short headway capability means high capacity can be achieved with bus-size vehicles.

6.3 Weight

Vehicle weight is very important and there is an important tradeoff between using a more expensive but lighter vehicle and requiring heavier guideways and a more powerful linear motor. Steel-wheel suspended vehicles typically have an empty vehicle weight that is 3 to 6 times the maximum passenger weight. With careful design an LSM propelled vehicle constructed with composite materials can lead to empty vehicle weight less than 2 times the maximum total passenger weight. Since transit systems spend much of their energy budget on acceleration, weight savings have an important impact on capital and operating cost. It is easily justifiable to pay more for a lighter vehicle. Table 6.1 gives our best estimates of baseline vehicle overall dimensions and weight.

Table 6.1. Dimensions and weight of baseline vehicle.

	Metric		*English*	
Length	10	m	32.8	ft
Width	2.9	m	9.5	ft
Height	3.2	m	10.5	ft
Rail gauge	1.7	m	5.6	ft
Weight empty	5,500	kg	12,100	lb
Max. load	3,000	kg	6,600	lb

6.4 Secondary suspension

The primary suspension is provided by the magnets but there will be a secondary suspension that has two components.

1) The magnet pods have pivots with dampers so as to allow tight turning radii in both horizontal and vertical directions. These components are needed at all speeds.

2) Pneumatic or magnetic springs allow improved ride quality and can, if desired, provide active control of ride quality, including tilting. These components are only needed at speeds above about 20 m/s (45 mph).

6.5 Onboard power system

It is necessary to provide onboard power for Heating, Ventilating and Air Conditioning (HVAC) and this power requirement is much greater than the total of all other onboard power requirements. The total average requirements are estimated to be 8 kW for HVAC and 4 kW for everything else with significantly more than this for peak power. A major reason for the low total requirement is the use of a permanent magnet suspension that requires an order of magnitude less power per Mg of vehicle weight than with an electromagnetic suspension. This power will be provided by batteries plus a charging source.

For the purpose of the demonstration prototype conventional lead acid batteries were used to power the suspension controllers but the final design will use newer battery technology to save weight. The onboard batteries will be charged via Inductive Power Transfer (IPT) between a power source on the guideway and pickup coils on the vehicle. MagneMotion has constructed such systems in the past and is confidant that the required power can be transferred at all speeds at reasonable cost and efficiency.

Unlike Transrapid, which depends on a 1,000 VDC sliding-contact power pickup system at speeds below 80 km/h (50 mph), *M3* vehicles will never require contact with the guideway and this will be a major asset in improving reliability and decreasing cost.

6.6 Use of larger vehicles or trains

The *M3* suspension and propulsion system can be used with almost any size vehicle or train of vehicles. It could conceivably be used to replace suspension and propulsion systems in competing maglev designs with potentially lower total overall cost. However, the *M3* system design is based on optimizing system cost and MagneMotion does not believe the use of larger vehicles or trains has advantages except when higher capacity than 12,000 pphpd or higher speed than 60 m/s (134 mph) is required.

7 Control System

7.1 Introduction

For any modern mass transit system a digital control system is required. There are many ways to implement such a control system, and one concept is presented in this chapter. Only the high level concepts are described here – there are many details of implementation that are left out for the sake of a concise, readable document.

7.2 Objectives

Any control system for people movers should be designed with the following goals:

Safety
The most important goal of the control system must be safety. According to USDOT 2001 highway fatality statistics, more than 40,000 people died last year in automobile crashes, a fatality rate of 1.52 people per 100 million vehicle miles traveled. While the general public accepts this rate, they rightly hold mass transit to a much higher standard, with outrage expressed at any deaths on a public transportation system so safety must be the primary goal of the control system.

Reliability
The transit system should be reliable, such that avoidable traveler delays are minimized, and should be efficient, so that the transport resource is used to near its full potential and not significantly limited by the control system. This ensures maximum return on investment.

Ability to achieve high capacity
The choice of control system can have a major impact on system capacity. The objective is to achieve the highest possible capacity consistent with the available resources and the requirement for very safe operation.

Flexibility and expandability
The control system should also be flexible, to be easily used for transit systems of differing types (e.g. shuttle vs. network) and variations in demand. At the same time, it should be expandable so that the system can be upgraded with additional guideway with minimal impact on existing routes.

Effective fault handling
The system should have carefully planned fault handling to deal with problems as they occur, both expected and unexpected types of problems.

7.3 Architecture

The architecture for the *M3* control system is hierarchical and designed for expandability. As the system grows, more modules are added at each of the lower levels. The hierarchical system also minimizes communication and required processing power, as each function can be implemented at the appropriate level. Figure 7.1 illustrates a 3-level hierarchical system suitable for a complex system, but for a simple system 2 levels may suffice.

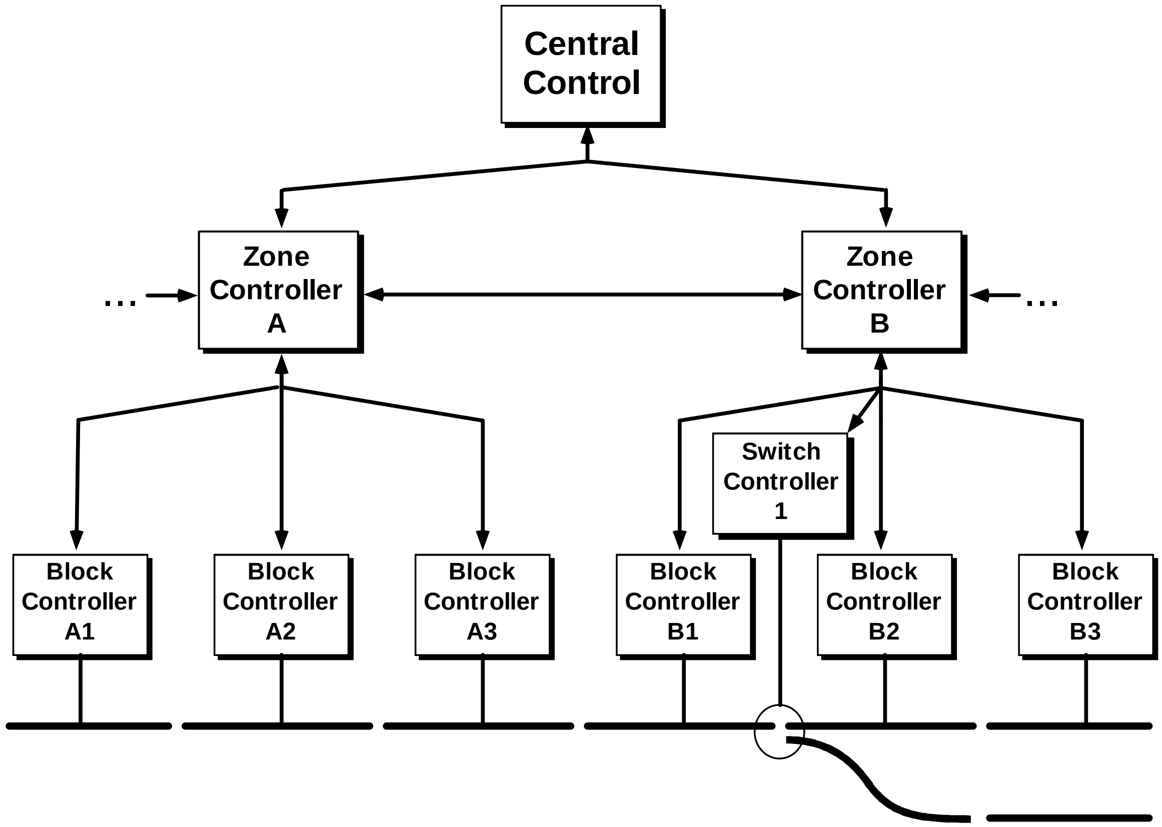


Fig. 7.1. Hierarchical control system.

Following are the functions of the different levels:

Block Controller:

- Constantly tracks position of one vehicle.
- Controls vehicle according to orders from zone controllers.
- Drives the linear motor via power electronics.
- Keeps track of vehicle state information, vehicle ID, position, velocity, etc.
- Communicates with adjacent block controllers and zone controllers.
- Provides motor synchronization, vehicle handoff, liftoff, E-stop, etc.
- Monitors status of block and inverter.

Zone Controller:

- Constantly tracks positions of vehicles.
- Monitors status of power system, block and switch controllers.
- Provides vehicle coordination: grants movement permissions to vehicles via block controllers; ensures adequate vehicle spacing; implements safe merge strategies; responsible for vehicle protection functions.
- Reports errors to central controller.
- Interfaces to station controllers.
- Tracks vehicle information, routing, etc.

Central Controller:

- Performs global optimizations; manages vehicles; performs vehicle routing and switching decisions.
- Displays system condition to operator.
- Records and reports fault conditions.
- Tracks network statistics.
- Communicates with all controllers and uploads software updates to them.
- Diagnoses system problems.

7.4 Proposed control scheme

The following control scheme is based upon a control scheme implemented by MagneMotion for a material handling system but with additional features to insure safety and fault tolerance.

7.4.1 Constraints

The control system design is strongly influenced by the system constraints. The limitation of one vehicle per LSM block imposes a significant constraint on how the system is operated. The system must ensure that under any normal set of operational circumstances each vehicle must be in a separate block. The stopping distance for a vehicle may be several blocks so each vehicle must also at all times have a dedicated block for the vehicle to stop, a place where no other vehicles are allowed.

Other constraints include the headway criteria, emergency egress points, stop exclusion areas (in switches, etc.), and emergency stop capabilities. A variety of headway constraints may be imposed on the system, including both ‘brick-wall’ and “safe-follower” types. A brick-wall headway is defined as the minimum headway between vehicles such that if a vehicle comes to an immediate stop (e.g. hits a brick wall), the following vehicle will be able to stop in the intervening distance. A safe-follower headway is defined as the minimum headway between vehicles such that if a vehicle applies maximum braking, the following vehicle will be able to stop without collision.

Finally, there are the ride quality and performance constraints of the system, described as jerk, acceleration and velocity limits.

7.4.2 System Characteristics

The propulsion system does not depend on friction to deliver thrust to the vehicle and provides consistent performance in any weather conditions. It has higher acceleration and deceleration capabilities than steel wheel on steel rail in even the best weather conditions. The *M3* system utilizes an inductive position-sensing system which is capable of constantly monitoring the precise position of all vehicles at all times. The propulsion control system is located on the wayside, so position information is easily shared for all vehicles over a hard-wired, low-latency communication system without the vagaries and susceptibilities of radio communication. Knowledge of precise position of all vehicles and the ability to apply consistent thrust in all weather conditions greatly facilitates the implementation of a short-headway control system.

7.4.3 Control system description

The control system developed for *M3* was tailored to meet the constraints of a long-stator LSM drive, specifically that only a single vehicle is allowed over any motor block at one time. The control system is based on the concept of a "target', or planned stopping point. This concept has been proven in a commercially operating material handling system.

In order to maximize throughput capability vehicles are operated in clusters. Each station is sized to handle a full cluster of vehicles. By starting and stopping a cluster of vehicles in a station, the line capacity through a station can be significantly increased over a single-vehicle station. The clusters act as virtual trains thereby allowing high capacity.

Each zone controller is responsible for allocating and releasing movement permissions for targets within its borders. The target "release" mechanism is set up with brick-wall inter-cluster headway and safe-follower intra-cluster headway, as illustrated in Fig. 7.2. This kind of operation allows for a level of safety similar to that of long trains. In such a cluster one can associate each vehicle with a car of a virtual train. If the highest level of throughput is not necessary, "brick-wall" headways may be used between all vehicles. The target concept is easily tailored to either headway type, and may even be changed dynamically for higher throughput during times of peak load.

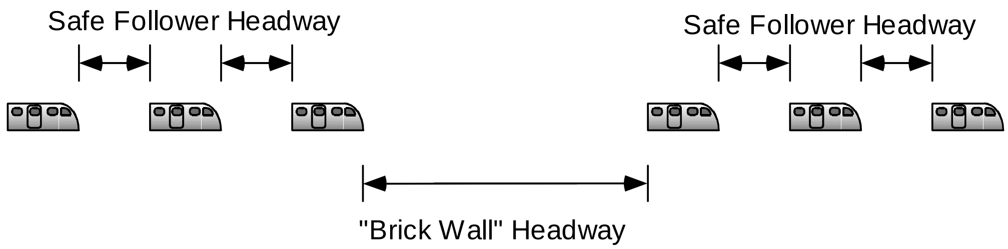


Fig. 7.2. Two clusters of vehicles.

7.4.4 Advantages of proposed design

The MagneMotion control scheme is able to meet high throughputs in a safe manner while respecting the one vehicle per block limitation. It is flexible in terms of the headway strategy used and is easily expandable. Through proper target placement the algorithm supports such design goals as sub-block switching and can limit the loading of support structures by limiting the number of vehicles on a particular track section. The scheme, through proper placement of targets, can also support stopping only at egress points along the guideway. This system is fail-safe in the sense that when a controller or a communication link is not operational, new targets are not granted and the vehicles come to a stop at their last acquired targets.

7.4.5 Advanced operational strategies

To exploit the full capability of the transport system, certain strategies may be used. In a high-speed urban system with on-line stations it will be necessary to use multi-bay stations to achieve desired throughput, as shown in section 7.5.2.

High-level control strategies may be layered on top of an existing control system by the central controller or at the track layout stage of design. For instance, faster travel may be achieved by using strategies such as selective station servicing, where every vehicle does not stop at every station. Each vehicle may, for instance, service two out of every three stations, with different vehicles serving different sets of stations. A passenger can still move from his origin to his chosen destination by selecting the correct vehicle or by changing vehicles. Since fewer stops are required, shorter travel time may be achieved. Also, demand-based station servicing may be used. For instance, vehicles may skip stations where there is no demand (which would require knowledge of the destination of each passenger).

7.5 Simulation

Preliminary simulations were performed to examine the basic limitations of the system. One such limitation is line throughput – the maximum capacity of the line. Using some basic assumptions, it was discovered that with a safe-follower headway the line has a capacity beyond the goal of 12,000 passengers per hour per direction (pphpd). The next most significant limitation investigated was throughput limitations of the stations, as they were assumed to reside on-line. The simulations described below focus on the limitations presented by the necessity to stop at on-line stations. With certain operational strategies, the simulations showed that the target throughput could be met with on-line stations.

7.5.1 Assumptions

The assumptions for the simulation are summarized in Table 7.1.

Table 7.1. Simulation Assumptions.

Item	*Value*	
Max Acceleration	1.6	m/s^2
Station Type	On-line	
Headway Type	"Safe"	
Dwell Time	15	s
Block Length at stations	18	m
Max Velocity	45	m/s
Vehicle Occupants	36	

The simulations were performed based upon a crude model of the control system previously described. The stopping place for each vehicle, a "bay", was assumed to be at the end of a block within the station. The assumed acceleration was based upon passengers standing in the vehicle to achieve maximum capacity. The throughput improves if the blocks are shortened due to the constraint of one vehicle per block constraint. Most of the benefit of shortening blocks is achieved with a block length of 18 meters but additional capacity is possible with shorter blocks within stations.

7.5.2 Results

Preliminary simulations were performed with a range of one to five bays per station. It was assumed that for multi-bay stations, a fleet of vehicles would move into the station, dwell, and leave the station. A cluster of vehicles stops at a sequence of bays in the station. To calculate throughput, the time was calculated from the start of the exit of one cluster of vehicles to the arrival of the next. This time was added to the station dwell time to realize the total time that the cluster occupied the station. From this value, the average throughput may be calculated. Figure 7.3 illustrates a scenario with 4 bays and 6 vehicles. As the first fleet of four exits, the next vehicles enter and stop at the first two bays in the station.

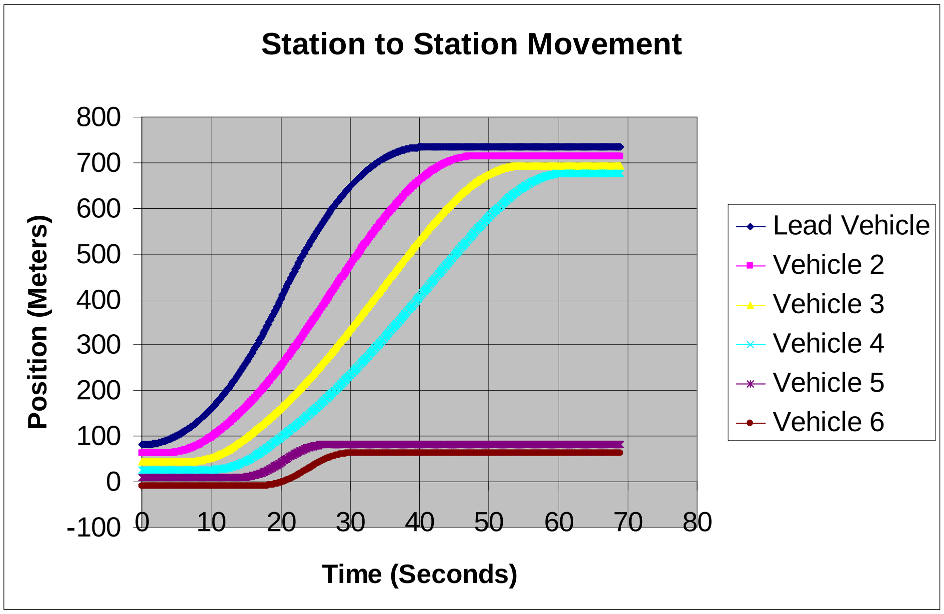


Fig. 7.3. Typical Simulation of a 4 bay station with 6 vehicles.

A summary of the results is shown in Fig. 7.4. A minimum of four bays is required to achieve the desired capacity of 12,000 pphpd. Selective stopping strategies may further enhance throughput, or reduce the required number of bays.

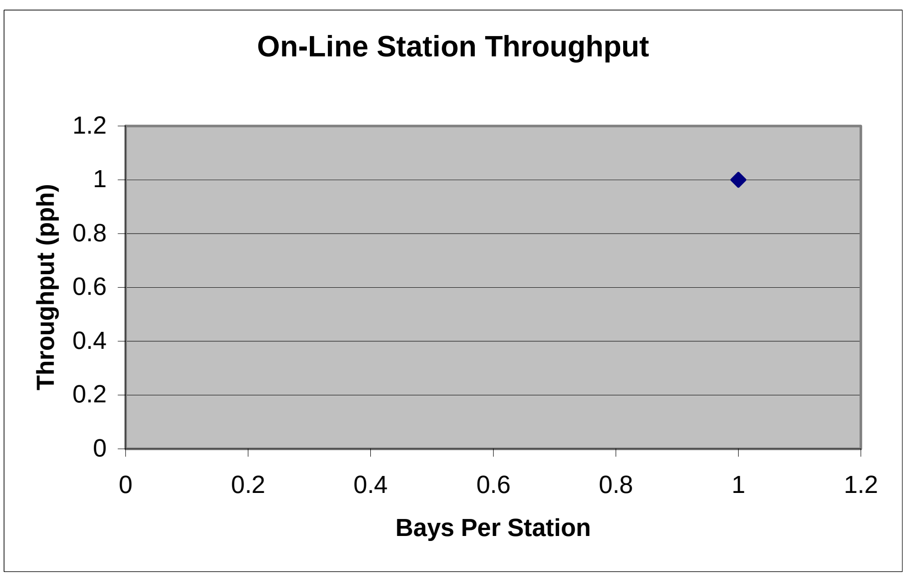


Fig. 7.4. Station throughput as a function of the number of bays.

7.5.3 Conclusions

The primary conclusion is that a throughput of 12,000 pphpd is achievable under certain assumptions using the control system concept described. The line is able to meet the capacity when using a "safe-follower" headway between vehicles of a cluster and 'brick-wall" headway between clusters. The stations are able to meet this throughput through the use of multiple sequential bays per station.

8 Demonstration prototype

The key suspension and propulsion design concepts have been tested by constructing the demonstration prototype shown in Fig. 8.1. This first generation prototype uses full-scale magnets but the vehicle is shorter and narrower than the vehicles described earlier. The prototype is fully functional, and has met its design objectives. The agreement between predicted and measured quantities ranged from fair to very good. The computer models used to design the demonstration system correctly predict the system behavior with good accuracy so it is reasonable to expect similar validity for the models of the full-size system.

Fig. 8.1. Photograph of prototype vehicle and guideway.

8.1 Static load test results

Table 8.1 gives data that is exemplary of the many measurements made and shows the relatively good agreement with predictions. Prototype testing involved a much wider range of load than is planned for a full-scale vehicle, so in normal operation the range of gap variation about the nominal value will be smaller (only about ±3 mm) than the range shown in Table 8.1. This data is for the initial prototype before the addition of lateral damping. The modified prototype is described later and has a higher lift force.

The operating gap of the nominally loaded vehicle is 0.6 mm larger than the predicted 20 mm gap. This deviance from the analytical model is due to the strength of the magnets. The incoming inspection data to measure the field strength of the raw magnets showed that the magnets generally had a higher field strength than the nominal value used in the FEA models. The effect of the stronger magnets can be seen in the test data, in this and subsequent sections, in the form of higher forces and a wider operating gap.

The power required to levitate the vehicle is very small, and the static, disturbance-free conditions in the laboratory accentuate this feature, resulting in the very small power requirements shown in Table 8.1. In a more realistic operating environment with disturbances due to wind gusts and guideway deflections, we estimate the levitation power requirement to be 100 W per Mg (i.e. tonne) of levitated mass.

Table 8.1. Static performance of initial demonstration prototype.

	Gap (design)	*Gap (actual)*	*Mass (design)*	*Mass (actual)*	*Lev power*
Light	25 mm	24.6 mm	734 kg	777 kg	2.0 W
Nominal	20 mm	20.6 mm	958 kg	981 kg	2.0 W
Heavy	15 mm	15.8 mm	1284 kg	1229 kg	2.3 W

The guideway is 6 m long and allows a vehicle to move 3.9 m. With a nominal load of 981 kg the maximum test speed was 1.74 m/s (3.8 mph) with an acceleration of 2 m/s^2. Figure 8.2 shows some of the component detail of the second generation test vehicle.

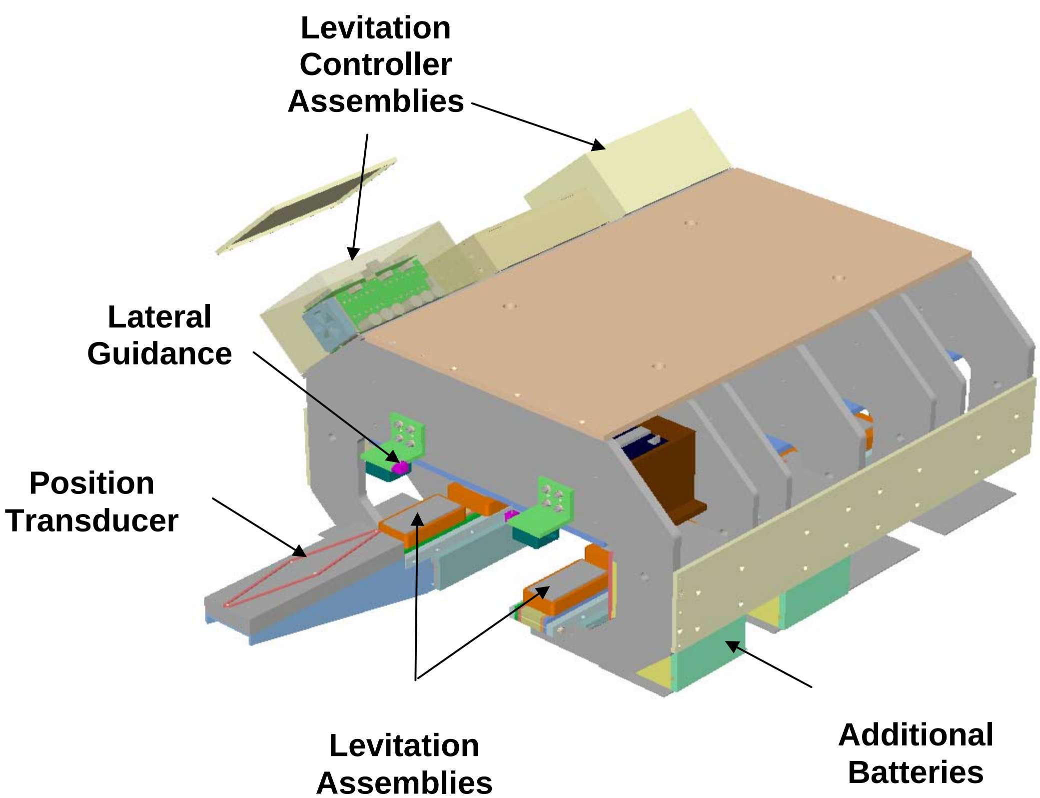


Fig. 8.2. Maglev test vehicle.

8.2 Lateral stiffness test results

The lateral stiffness of the suspension system was measured with the apparatus shown in Fig. 8.3. The levitated vehicle is displaced laterally via a leadscrew, cable, and pulley arrangement. The lateral force required to displace the vehicle is measured with a strain gauge. The results of this test series are shown in Figure 8.3. As expected, the lateral force increases roughly in proportion to the lateral displacement. The stiffness associated with this relationship is 101 N/mm. The measured data agrees in form with the predictions of the FEA model, but the measured data shows approximately 10% larger forces due to the magnet field strength being greater than assumed in the FEA model.

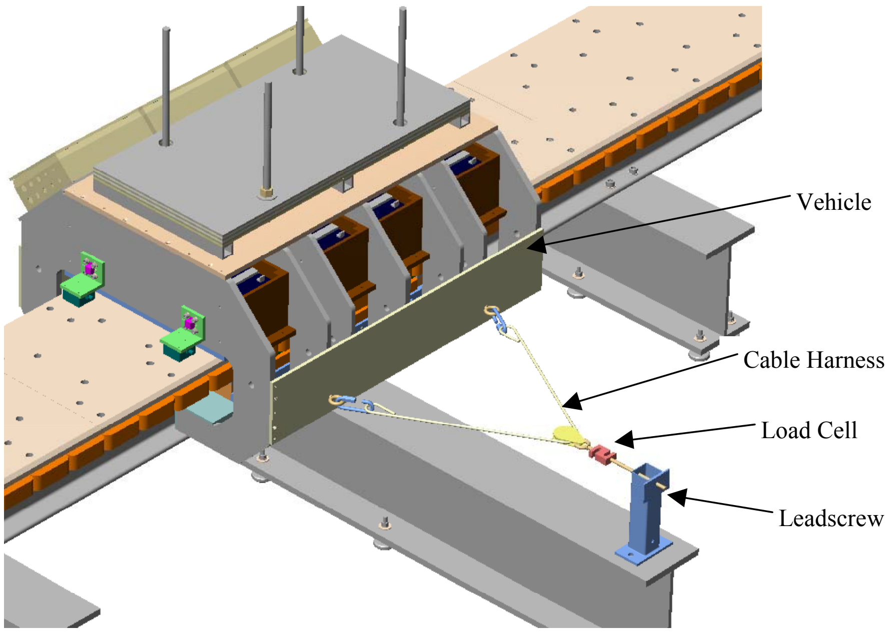


Fig. 8.3. Lateral displacement test setup.

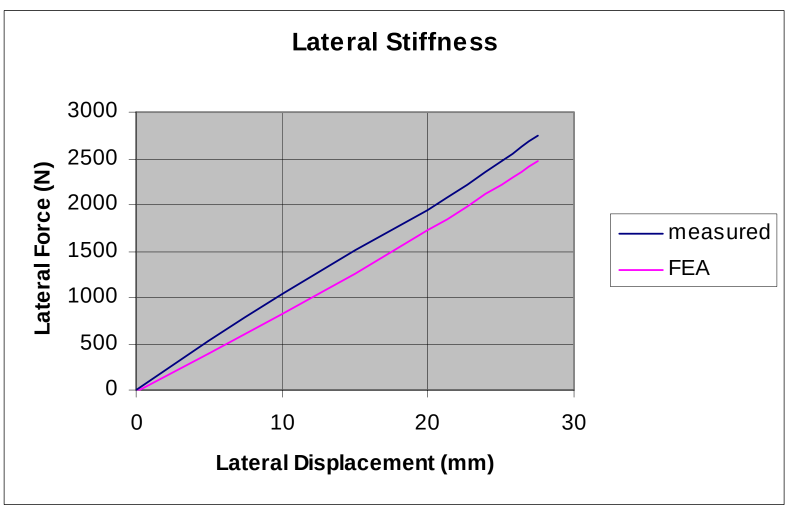


Fig. 8.4. Lateral stiffness measurement results.

8.3 Lateral damping test results

For our second generation test vehicle a key objective was to demonstrate lateral damping using active control coils. Four tasks were necessary to achieve this objective.

1. Analyze lateral damping dynamics and determine requirements.
2. Modify the integrated propulsion and levitation pods of the original demonstration system to include active magnetic damping of lateral motion.
3. Develop active controls for levitation and lateral damping.
4. Confirm the analysis and lateral damping performance.

The lateral damping mechanism facilitates damping of the yaw and sway axes, thus enabling the levitation controller to damp vehicle movements in yaw, pitch, sway, roll and heave.

8.3.1 Analysis

The implementation of an actively controlled electromagnetic damping mechanism required a comprehensive magnetic analysis effort. Several objectives must be met to successfully complete the magnetic design of the levitation assembly for the demonstration system. First, we determined the lateral force capability needed to provide adequate damping of the lateral vehicle dynamics. Next, we determined the levitation assembly configuration required to yield the desired lateral force. Lastly, we determined the optimal magnet dimensions to minimize force and pitch variations. Each of these objectives is discussed in the following sections.

8.3.2 Lateral force requirements

The analysis in this section determines the lateral force capability required to provide adequate lateral damping of the demonstration vehicle. The lateral guidance force for the demonstration system is $F_z = -65z - 0.0195z^3$ N and the vehicle mass is 1275 kg.

The lateral dynamics of the demonstration system, with a damping force of $F_z = -9.1\frac{\mathrm{dz}}{\mathrm{dt}}N$ are shown in Figure 8.5. The response is well behaved, with an overshoot of approximately 17%. The lateral dynamics of the demonstration system, with a damping force of $F_z = -13\frac{\mathrm{dz}}{\mathrm{dt}}N$, are shown in Figure 8.6. The response is well damped, with an overshoot of only 4%.

The maximum lateral displacement for the vehicle is 40 mm. The recovery transient from this initial displacement would require a peak lateral damping force of 1718 N for the model used for Fig. 8.5, and a peak lateral damping force of 2093 N for the model used for Fig. 8.6. Both of these peak force requirements are quite large, considering that the system will rarely operate with such an extreme lateral transient. If the system from Fig. 8.5. is operated with a maximum lateral damping force limit (i.e. a 'clipping' or 'saturation' type limit) of 910 N, the response shown in Fig. 8.7 is the result. The first overshoot is larger, due to the saturated nature of the damping force, but the response then quickly settles in a well-behaved manner. Thus the most extreme displacement is damped by a factor of two in the first overshoot, and then settles in an unsaturated manner. This behavior is quite acceptable, and substantially reduces the peak damping force requirement from the unsaturated case in Figure 8.5. Figure 8.8 shows the transient response of the system of Fig. 8.6, but with the same maximum lateral damping force limit of 910 N. Once again, the extreme initial displacement is damped by a factor of two on the first overshoot, and then settles in an unsaturated manner.

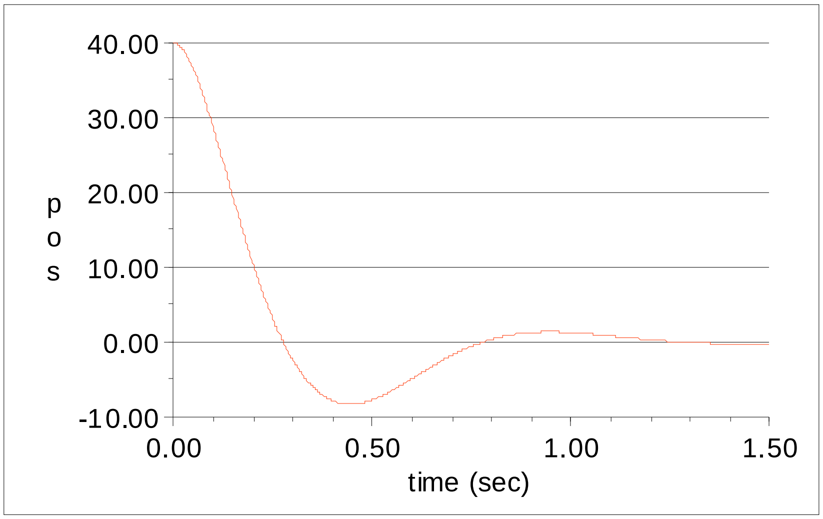


Fig. 8.5. Lateral dynamics with K_v = -9.1 N/(mm/sec) × dz/dt

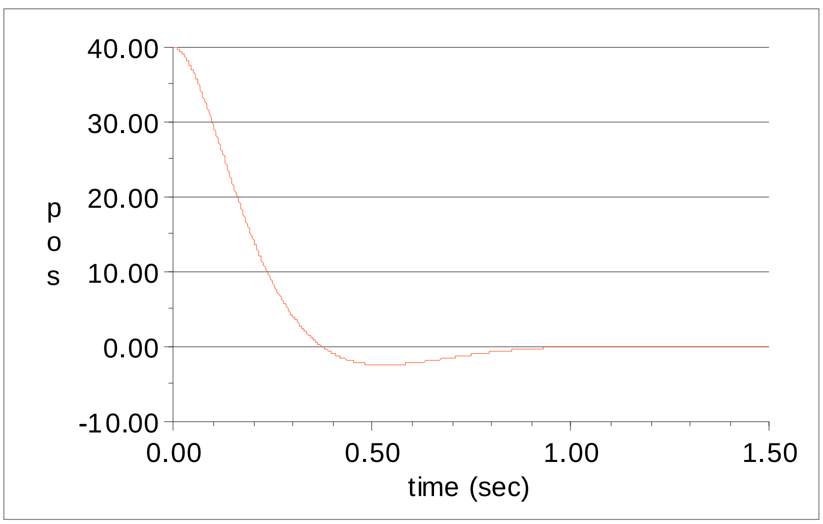


Fig. 8.6. Lateral dynamics with K_v = -13 N/(mm/sec) × dz/dt.

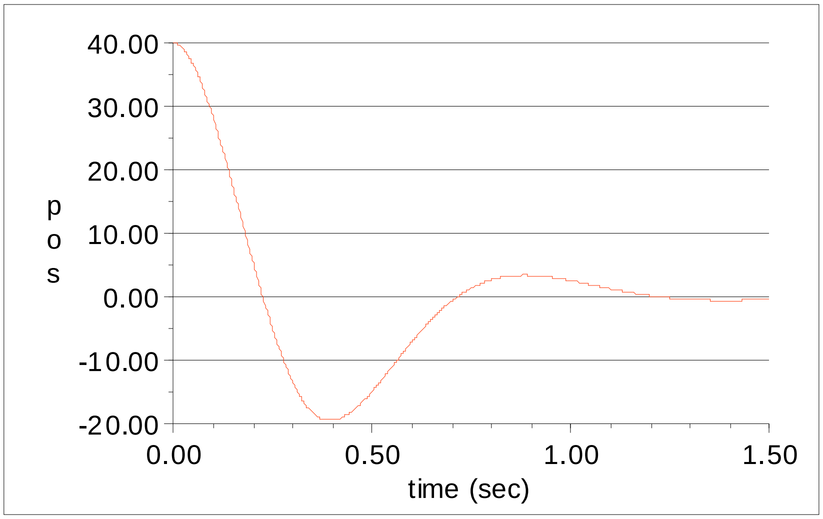


Fig. 8.7. Lateral dynamics with K_v *= -9.1 N/(mm/sec) × dz/dt, force limit of 910 N.*

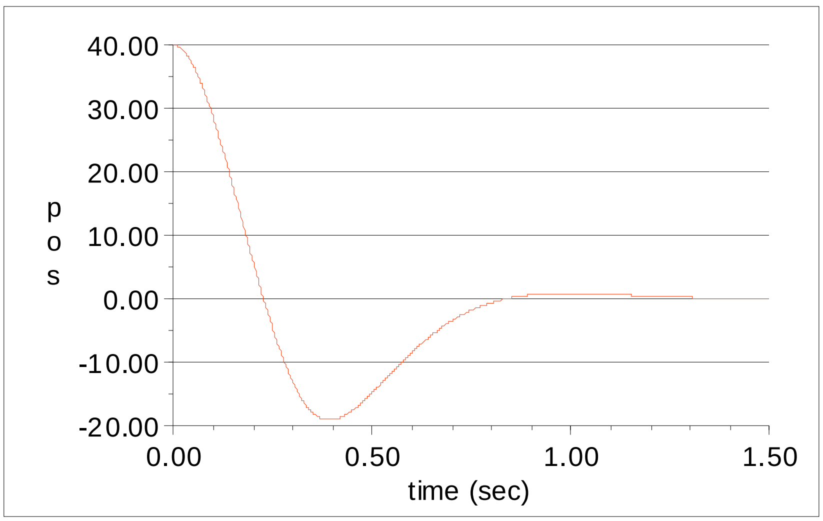


Fig. 8.8. Lateral dynamics with K_v *= -13 N/(mm/sec) × dz/dt, force limit of 910 N.*

In conclusion, a maximum lateral damping force capability of 910 N (or 455 N per pod) is required to adequately damp the lateral dynamics of the demonstration system. A damping force coefficient in the range of –9.1 to –13 N/ (mm/sec) yields a well-behaved transient response. A choice towards the top of this range will provide the benefit of a smaller overshoot at the expense of more control power for a given disturbance spectrum. Since the value of this coefficient is a software feature, it may be easily selected within this range based on the operating environment of the system.

8.3.3 Lateral Dynamics test results

The dynamic response associated with lateral movement was measured to verify the analytical results. An inductive position sensor was placed in proximity to the side of the vehicle, thus allowing the lateral position of the vehicle to be measured and recorded on an oscilloscope. The vehicle was pulled laterally to an offset position, and then released to examine the step response of the suspension system to lateral disturbances. The results of this test are shown in Figs. 8.9-11. Figure 8.9 shows the step response to a lateral disturbance with the controller coefficients set to provide no lateral damping. As expected, this plot shows a very lightly damped response with a natural frequency of approximately 1.17 Hz. Figure 8.10 shows the lateral step response with the controller coefficients set for a lateral damping force of approximately -10 N/(mm/sec). There is a marked improvement in the system damping, but the response is not as well damped as predicted. Observation of the vehicle motion reveals that the roll mode of the vehicle is also being excited in this response. Analytically, this makes sense; the lateral damping force is applied at the suspension gap, which is approximately 200 mm below the center of gravity of the vehicle. Thus, the lateral damping force causes a torque about the roll axis of the vehicle, which excites the roll mode.

Figure 8.11 shows the lateral step response of the vehicle with the controller coefficients set for a lateral damping force of -10 N/ (mm/sec), as well as a compensating roll torque to null out the effect of lateral force on the roll axis. The response is improved, showing a very well damped behavior. Observation of the vehicle motion during the transient reveals that there is still a roll axis excitation. The roll and sway (lateral) axes are coupled through the lateral stiffness force measured in the previous section. Since this force, like our lateral damping force, is applied at the levitation gap which is below the center of gravity of the vehicle the roll and sway axes are dynamically coupled to one another. This coupling is not a problem in practice since all of the vehicle modes are well damped. The coupling merely makes the isolation and laboratory measurements of the response of a particular mode difficult.

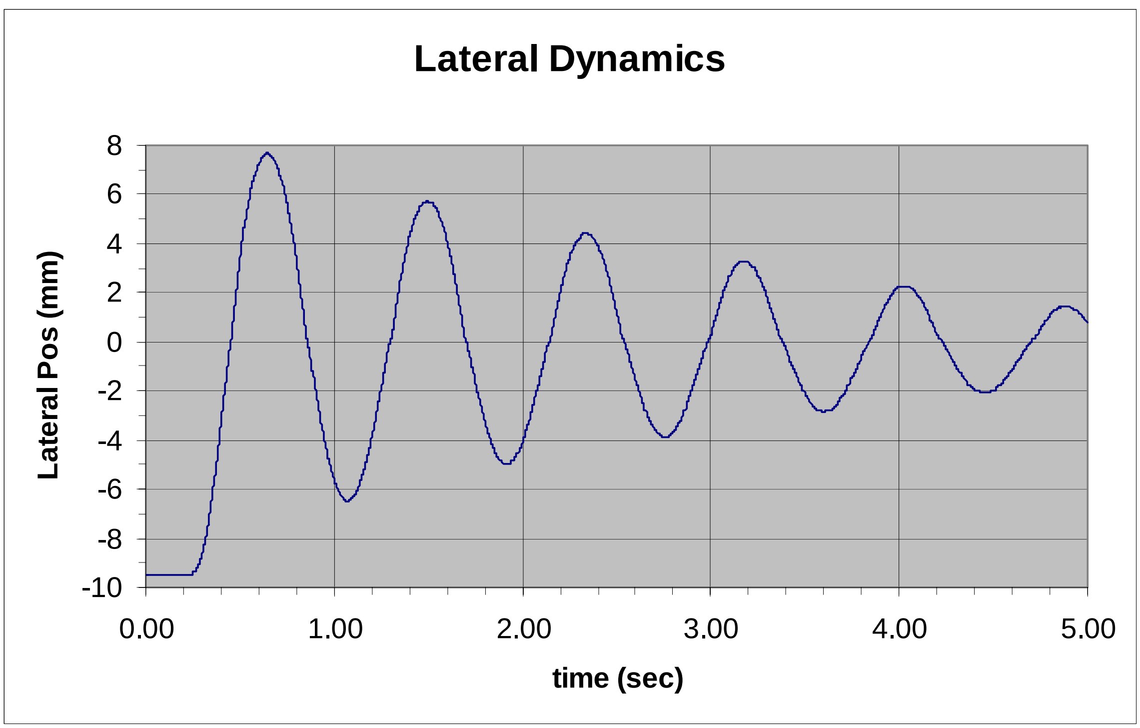


Fig. 8.9. Lateral step response with no damping.

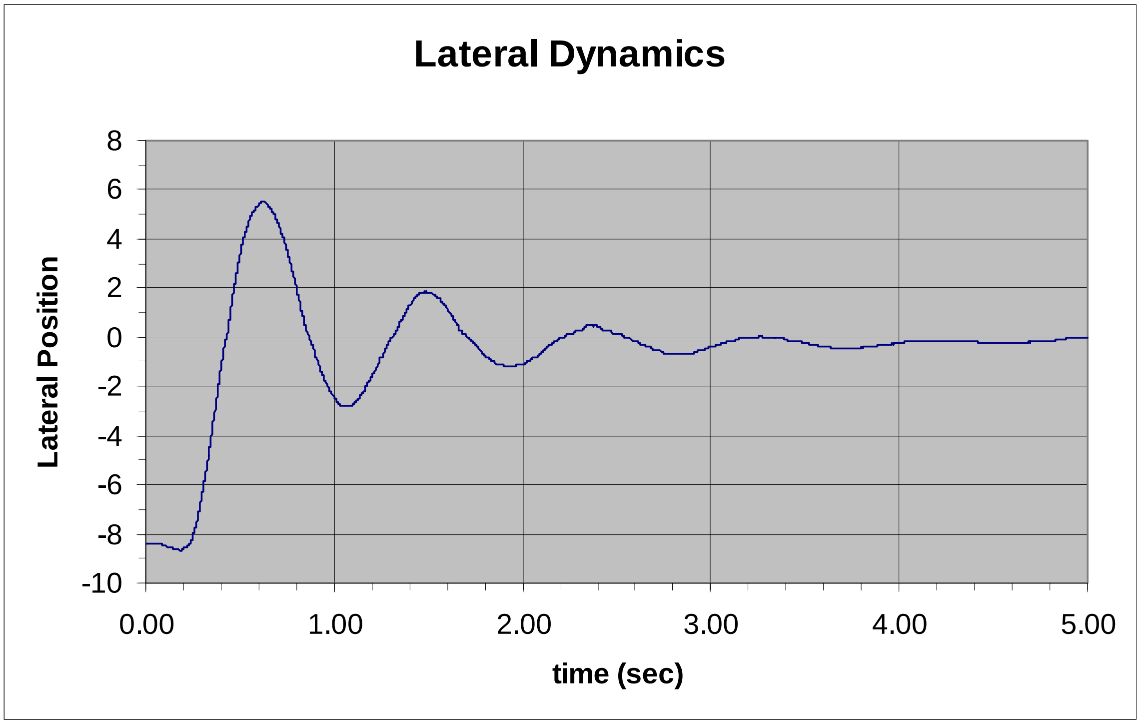


Fig. 8.10. Lateral step response with lateral damping.

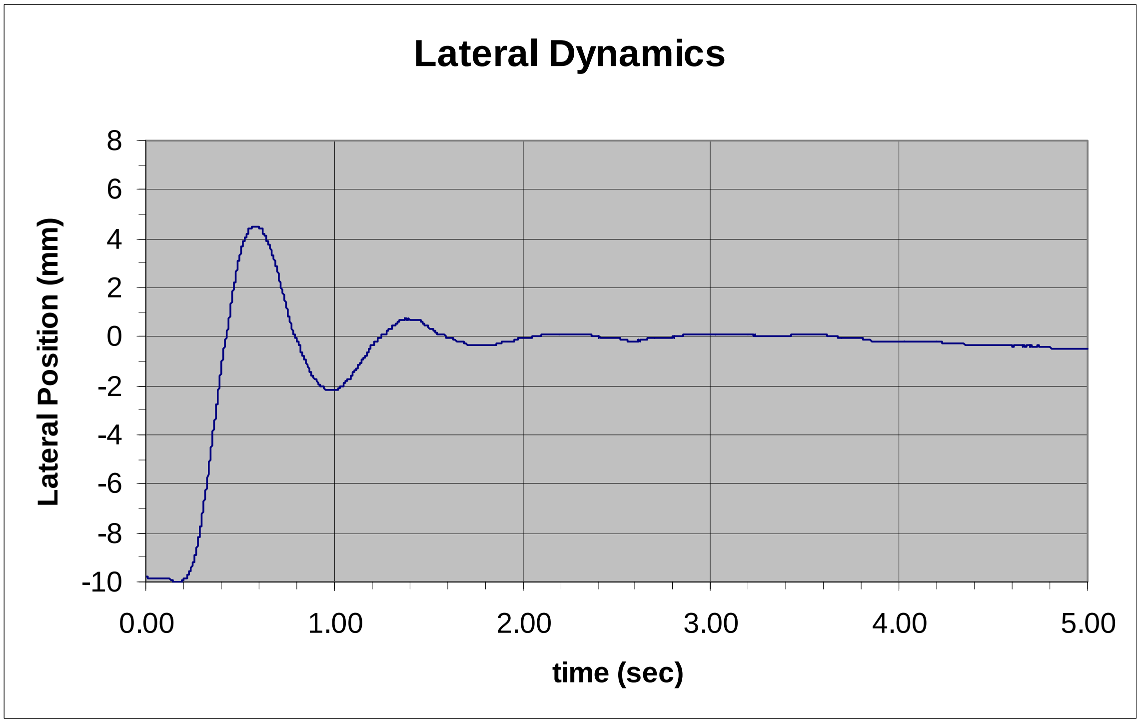


Fig. 8.11. Lateral step response with lateral damping and roll torque compensation.

8.4 Magnet optimization

The configuration of the end magnets was optimized to minimize the variation of the pitching moment, while also retaining a small cogging force. In addition, the end magnet configuration must conduct half of the flux from a full magnet to ensure that the flux is distributed evenly throughout the vehicle back iron. To accomplish these objectives, the end magnet width (in the longitudinal dimension), thickness (in the vertical dimension), and spacing relative to the full magnets were varied. The optimization process was conducted on a 2D model, since it was determined that more than a year of compute time would be required for optimization on a 3D model. A set of scripts allowed adjustment of one of the variable parameters, while holding the other two parameters constant. The scripts were run iteratively to search for the optimal solution.

8.5 Conclusions

The first generation prototype demonstrated that our design methodology was sound and all critical parameters agreed with predictions within a few percent. The first test vehicle was modified with the principal objective of demonstrating active lateral damping and reduced pitching due to end effects. The resulting second generation test vehicle met all expectations and MagneMotion is confidant that when the design is scaled up to a full size vehicle the performance will meet our objectives.

9 Application examples

Different applications require different design strategies to achieve optimum performance. The following Sections give typical design parameters for four types of applications that span a wide range.

The performance for a typical trip of 3.2 km (2 miles) is shown in Fig. 4.3. This Figure shows the velocity vs. time based on a simulation for the LSM and vehicles described in this paper. The LSM is capable of acceleration and braking at a rate of 2 m/s^2 for a nominal load. For normal operation the acceleration and braking are limited to 1.6 m/s^2 in order to limit the forces on standing passengers. At higher speeds the motor inductance and aerodynamic drag limit acceleration. Fig. 4.2 shows that with regenerative braking the average LSM power requirement is 65 kW for a typical mission.

From Fig. 4.3 it can be concluded that the travel time is 30 seconds more than it would be if the acceleration and braking were instantaneous. For example, a trip of 3,200 m requires a travel time of 3,200/45 + 30 = 101 seconds. Assuming 20 seconds for dwell time to unload and load passengers the trip takes 121 seconds for an average of 26 m/s (59 mph). This is optimistic because it neglects possible increases in time due to hills and turns, but it shows the potential of the *M3* system. This time estimation method has been used to arrive at the average speeds given in the following discussions of applications. It is recognized that many other factors will affect performance so the speeds given here are probably somewhat higher than might be realized in practice.

9.1 Shuttle

There are many examples of Automated People Movers (APM) used for shuttling passengers a short distance, such as within an airport or from a parking lot to an arena. This example assumes a much higher speed than would normally be used for such an application, but the speed has very little impact on cost and may even lower it.

9.1.1 Design problem

Design a 1-mile long shuttle for moving 12,000 people per hour per direction (pphpd) at speeds up to 45 m/s (101 mph). Assume a maximum capacity of 40 passengers per vehicle and dwell time of 48 seconds for unloading and loading. The design should minimize life-cycle-cost of the vehicles and propulsion system.

9.1.2 Conventional solution

Use light-rail with a top speed of 45 mph and acceleration limit of 0.8 m/s^2. The propulsion is by vehicle based motors, either rotary or LIM, with power transferred to the vehicle via an overhead catenary. A typical design uses a train of 2 vehicles, each of which is articulated for sharper turns. The empty weight of each of the vehicles is 40 tons. For this heavy train it is economically prohibitive to use higher acceleration or speeds. The result is the need to use a larger number of trains and to increase passenger trip time as compared with what is possible with *M3*. The use of smaller vehicles would require headways that are too short for conventional control systems. LIM propulsion of wheeled vehicles has more reliable traction than rotary motors but it would increase cost and not allow as short a headway as is possible with an LSM.

9.1.3 M3 solution

The baseline vehicle carries no propulsion or control equipment so it can be light and relatively inexpensive. The light weight makes higher acceleration feasible and this makes higher top speeds useful. LSM propulsion can allow short headway between adjacent vehicles without risk of accident because the propulsion does not depend upon adhesion and the control has precise position information at all times and, most important, it does not depend upon communication with a moving vehicle. In order to allow stopping without offline loading the vehicles are operated in clusters with sufficient headway between clusters to allow unloading and loading one cluster before the next cluster arrives. Each vehicle in a cluster has its own stopping place so the cluster acts as a virtual train, but without the guideway and propulsion requirements to support and power a physical train.

Because the trip is short the vehicles are configured with fewer seats so as to allow 40 passengers without exceeding the weight limit. Five vehicles are organized as a cluster with one cluster departing from each end every 60 seconds so as to achieve the desired capacity. The individual vehicles operate with headways as short as 4 seconds. At each end of the trip the vehicles negotiate an 18.3 m (60’) radius turn. In order to maximize capacity it is important to minimize dwell time. The best way to do this is to use 2 doors and to unload and load at different locations. The unloading is done in 24 seconds and the loading is done after the vehicle negotiates the turn in another 24 seconds. Each vehicle takes 240 seconds for a complete round trip so 4 clusters, or 20 vehicles, are required to provide the desired capacity. The travel time is 65 seconds and with a cluster leaving from each end every 60 seconds the average wait time is 30 seconds. The average trip time is thus 95 seconds giving an average trip speed of 17 m/s (38 mph, 61 km/h). There are almost always some vehicles accelerating or cruising while others are decelerating so it is possible to use virtually all of the regenerated energy.

9.1.4 Variations

The length could be changed and intermediate stops could be added. If capacity requirements are reduced there do not need to be loops at each end and single clusters of vehicles could shuttle back and forth on each side, similar to conventional shuttle designs but with faster travel time and increased capacity.

There is presently a plan to install a 3-mile shuttle from Oakland Airport to the BART Coliseum Station with up to 2 intermediate stops. An *M3* system could provide the required throughput with almost a factor of two reduction in travel time and vehicle capacity and with a saving in guideway and energy cost. With conventional technology the projected cost of the shuttle is almost $300 million while an *M3* system would cost less than half as much.

9.2 One-way loop

With a high top speed it is feasible to use a 1-way loop for distances up to several miles. This reduces cost and minimizes the visual impact of the guideway.

9.2.1 Design problem

Design a 4 km long 1-way loop that can carry at least 3,600 pph at speeds up to 30 m/s (67 mph). There should be 4 stops, a main stop and 3 satellite stops. Most people will be traveling between the main stop and a satellite but a significant number will be traveling between 2 satellite stations.

9.2.2 Conventional solution

A monorail propelled by rotary motors is a common solution. An alternative is a cable propelled system such as ones being built by Otis and Doppelmayr. These can be less expensive than light rail but average speeds are typically less than 10 m/s (22 mph, 36 km/h) and capacity is usually less than 3,000 pphpd.

9.2.3 M3 solution

The short distance and necessity of turning in a moderately short radius makes a top speed of 30 m/s (67 mph) about all that can be useful. This reduces cost of the propulsion system at the expense of using more vehicles, but for the required capacity the number of vehicles is fairly modest. Cost and travel time are both reduced by using a station skipping strategy. Vehicles are operated in clusters of 3 but each vehicle skips one satellite station on each passage around the loop.

The time allowance for traveling 1 km is 55 seconds and for a 2 km trip it is 90 seconds. Dwell time allowance is 50 seconds at the main station and 40 seconds at the satellite stations. By using 3 clusters of 3 vehicles each there are 120 vehicle round-trips per hour. With 36 passengers per vehicle this provides a system transport capacity of 3,888 pph. The average time between clusters is 100 seconds and the minimum vehicle headway is 4 seconds. There are always some vehicles accelerating or cruising while others are decelerating so it will be possible to use virtually all regenerated energy.

9.2.4 Variations

If required at a later time, the capacity can be increased by adding an additional vehicle to each cluster without any other change except to increase the capacity of the rectifier station. By giving up the station skipping strategy it is possible to add an additional cluster for still higher capacity.

It is instructive to compare an *M3* system with the new $650 million Las Vegas Monorail. This state-of-the-art monorail has a dual guideway 4 miles long with 6 intermediate stops. Nine 4-car trains, each with a capacity of 72 seated and 152 standing and operating with 2-minute headway provide a capacity of 6,720 pphpd. The maximum speed is 45 mph but travel time from one end to the other is 14 minutes so average travel speed is only 17 mph. An *M3* system with greater capacity would cost on the order of $120 million plus the cost of stations, and reduce end-to-end travel time to 6.5 minutes. There would be 42

40-passenger vehicles operating in 3-vehicle clusters with 60 second headway between clusters and a capacity of 7,200 pphpd.

9.3 Airport connector

9.3.1 Design problem

Design a high speed transit system to carry people between an airport and city center. The travel distance is 30 km and the objective is to minimize travel time without exorbitant cost.

9.3.2 Transrapid maglev solution

This example is based on the Transrapid installation in China that is now carrying passengers from Pudong Airport to Shanghai, a distance of 30 km. The top speed is 119 m/s (267 mph, 430 km/h), the travel time is 8 minutes, train headway is 10 minutes and average trip time is 13 minutes. There are three 5-car trains capable of carrying 574 passengers each for a capacity of 3,444 pphpd. The reported cost of the Shanghai installation is $1.2 billion or about $60 million per dual guideway mile. The high cost is due to the heavy trains (260 Mg each), very heavy guideway (7.7 Mg/m), high power inverters (estimated to be 7.2 MVA per inverter), and the mass of the suspension rails (twice the width of the rails used by *M3*). The suspension uses electromagnets which require more than 1 kW/Mg and there are separate guideway-mounted guidance rails and onboard magnet structures on each side of the trains. The guidance system causes substantial power loss at high speeds, in some cases comparable to the suspension loss. Sliding contact power transfer is used to provide onboard power for speeds up to 22 m/s (50 mph, 80 km/h) so some of the contact-free advantages of maglev are not achieved.

9.3.3 M3 solution

The *M3* design delivers comparable performance at much lower cost. The maximum speed is assumed to be 60 m/s (134 mph). Although higher speeds are possible, this example shows that the lower top speed but higher acceleration and shorter vehicle headway gives comparable performance to Transrapid for trips of this length. Since the top speed is only one half that of Transrapid, the guideway layout can tolerate a factor of 4 reduction in turn radii and the guideway beams need not be as stiff or precisely aligned, so guideway cost can be reduced. The cost of the power system and the energy consumed are both reduced by a large factor. Inductive power transfer is used at all speeds so the cost and maintenance issues associated with sliding contacts are avoided.

The proposed design uses a 3-vehicle cluster with minimum headway of 4 seconds near the stations and 10 seconds in the high speed regions. There are 3 stopping places at each end so each vehicle in the cluster has a unique stopping place. Since the trip is long it is preferable to not have standees so the vehicle is configured to have 36 seated passengers and no standees. A cluster leaves every minute for a capacity of 5,760 pphpd with an average travel time of 9 minutes and an average trip time of 9.5 minutes. At a later time a 4th vehicle could be added to each cluster to increase the capacity to 7,260 pphpd with no change in minimum headway. Each vehicle makes a round trip in 20 minutes, so there are 20 clusters for a total of 60 vehicles. The vehicles seat a total of 1,920 passengers, giving a passenger capacity per seat that is 50% higher than Transrapid, and the vehicles are much less expensive on a per-seat basis.

The guideway has a loop at each end. This allows the vehicles to be designed for unidirectional operation with reduced aerodynamic drag. The option of using a switch and bidirectional vehicles is possible, but this would entail the use of more vehicles for the same capacity and does not allow some of the advantages of unidirectional vehicles. For Transrapid the minimum turn radius is so large that they need to use a switch and bidirectional vehicles.

There are always enough vehicles accelerating or cruising to absorb all regenerated braking energy, but the energy savings are a much smaller percentage of energy usage because the trips are long and fast

9.4 Intercity

The focus of the *M3* development has been on urban maglev with maximum speeds in the range 30-60 m/s (67-134 mph), but the suspension and propulsion systems are capable of much higher speed with no change in the basic design. We can envision a higher speed intercity maglev system that uses the same guideway as urban maglev for distribution of passengers in urban environments, just as high speed rail uses low speed tracks for operation in cities. The key is to make the high and lower speed designs interoperable so that all vehicles can traverse the same guideway, albeit with the speed and thrust limits determined by a combination of vehicle and guideway attributes. Although the *M3* technology is almost certainly capable of speeds as high as Transrapid, we feel that for U.S. applications the most cost-effective design is for a top speed in the range 75-90 m/s (168-202 mph, 270-324 km/h), the same range as for existing high speed rail.

Following are the major changes required to make the *M3* design competitive with high speed rail:

- *Increase vehicle length*
  At high speeds the aerodynamic drag becomes a dominant parameter and the per-seat drag is reduced by using longer vehicles with more seats and better nose and tail streamlining. Longer vehicles also allow more comfortable seating and room for toilets and vending machines. With longer and faster vehicles it is economically feasible to have cabin attendants to deal with a variety of issues that arise on longer trips. The longer vehicle is constructed by using 6 or 8 magnet pods and a 2 or 3 section articulated vehicle for 60-100 seated passengers.

- *Increase guideway stiffness:*
  This can be done by decreasing pier spacing (e.g. use 24 m pier spacing), increasing beam stiffness, or some combination.

- *Increase vehicle and cluster headway times:*
  The longer and faster vehicles will require greater headway between vehicles, on the order of 6 seconds for a speed of 75 m/s (168 mph). Since vehicle travel is likely to be scheduled it is not necessary to have as short a time between clusters so cluster spacing should be about 4 minutes. This long spacing between clusters will allow offline loading and unloading and thereby allow a mixture of express

and local service to use the same guideway. It will also allow long enough clusters to achieve a capacity of at least 8,000 pphpd.

- *Increase per-vehicle propulsive power*
The heavier and faster vehicles will require higher propulsive power per vehicle. For long distance travel there are good reasons to reduce peak thrust to about 1.6 *g* but the peak power per-vehicle will need to be increased. This might be done by increasing the DC bus voltages to ±1,500 VDC and using readily available, higher voltage IGBTs. The number of turns in the winding do not need to be changed. Shorter low speed vehicles will be able to operate on a high speed guideway with thrust up to 1.6 *g* and longer high speed vehicles will be able to operate on a low speed guideway at reduced speed.

- *Increase guideway and vehicle tilt*
Higher speed capability is better utilized if the vehicle can tilt while negotiating turns so as to allow more versatility in following available rights-of-way. A guideway tilt of 10º and a vehicle tilt of 7º will allow a vehicle to make a 1.5 km (0.9 mile) radius turn at 75 m/s (168 mph).

The sum total of all of these changes would increase the system cost by 10-25% depending upon the maximum speed and other parameters, but it would still be substantially less expensive than a new installation of high speed rail or Transrapid maglev. It could be constructed over existing rail lines thereby allowing the existing rails to be used exclusively for freight. With relatively high acceleration and tilt capability it is possible to negotiate the turns required to follow a rail line at high speed without losing too much time slowing for turns.

As a specific example, the *M3* design is a suitable replacement for CHSST for the Colorado Maglev Project as reported in documents accessible at http://www.dot.state.co.us/publications/maglev/maglev.htm. We would propose use of a 75 m/s (168 mph) top speed, more use of point-to-point express scheduling, and vehicle tilting so as to increase the average speed from Chubu's estimate of 31.7 m/s (71 mph) to at least 45 m/s (101 mph). We have not worked out details of vehicle and guideway design so as to be able to give reliable data on energy consumption but are confident that we can provide faster travel at no higher energy or dollar cost than for a CHSST system.

MagneMotion does not advocate a high speed intercity application for an initial installation but would like to avoid making design decisions that preclude the evolution of a maglev system that has the versatility of conventional rail and allows a wide range of speeds. We believe that *M3* has greater versatility than Transrapid and will turn out to be a more cost effective design for all speed ranges of interest in the U.S.

9.5 Summary

The four examples represent very different types of applications, all served by common suspension, propulsion and guideway designs. The vehicles for any one system could operate on the other systems with modified performance.

Table 9.1 gives a comparison of some of the performance predictions. The average power is for the time the vehicle is moving and the energy usage assumes full use of regenerated energy and a 50% passenger load factor. The Intercity example has not been worked out in enough detail to give energy consumption details.

Table 9.1. Comparison of four examples of *M3* applications.

	Shuttle	*Loop*	*Airport*	*Intercity*
Capacity, pphpd	12,000	3,600	4,300	6,000
Maximum speed, m/s	45	30	60	75
mph	101	67	134	168
Distance, km	1.6	4	30	250
Stations	2	4	2	17
Average speed, m/s	25	20	35	45
mph	55	44	78	101
Average power , kW/veh	60	80	230	
Energy, J/pas-meter	524	400	814	
BTU/pas-mi	843	644	1,310	
Regeneration energy savings, %	50	45	6	

For comparison, the Transportation Energy Data Book: Edition 23 (Oak Ridge National Laboratory, October 2003, available at: http://cta.ornl.gov/data/) gives the following energy intensities in BTU/pas-mile for conventional transit in 2001: commuter rail 2,717; rail transit 3,114; transit bus and trolley bus 4,125; Amtrak 4,127. This reference assumed an electricity generation and distribution efficiency of 29% so 1 kJ is equivalent to 3.268 BTU and this factor was used in computing the BTU/pas-mi in Table 9.1.

Even allowing for substantial energy increases due to HVAC and other loads, the LSM approach uses much less energy. This is due to kinetic energy recovery, lighter vehicles and more efficient propulsion. Another major source of energy conservation is the ability to fine tune vehicle capacity to need and avoid using long and nearly empty trains for off-peak travel.

10 Cost estimates

This section develops estimates of the cost per mile for the *M3* Urban Maglev System. Costs are compiled from information supplied by component designers and manufacturers and have been confirmed by a second source where possible. In a few instances there is not enough information to make an accurate estimate, but all of the primary costs have been determined after consultation with appropriate manufacturers and vendors. MagneMotion will continue to refine the cost estimates as the design evolves.

The cost estimates for all guideway related items are computed on a per-unit-beam-length basis. The baseline design calls for double-span beams that are 72 meters long. Each 144-meter length of guideway contains the following:

- 4 72-meter long beams.
- 4 piers.
- 16 36-meter long LSM stators.
- 1 inverter station containing 8 inverters and associated switches and controllers.
- 1 hub controller and communication module.
- Power cables for distributing DC and AC power

The DC power is provided by a rectifier station located every 8 km (4.97 miles), and each station contains a power transformer and rectifiers that provide separate +900 and –900 VDC power with a total power rating of 1.5 MW. The rectifier station may include a source of emergency power, but this cost has not been included; a rough estimate is $100,000 added cost for every rectifier station for an automated 100 kW DC generator.

Power station rating and spacing is consistent with operating 4 vehicles per mile of dual guideway, so this is used as the nominal vehicle requirement. If the average speed is 50 mph then the capacity is 3,600 pphpd. For different applications the number of vehicles per mile and power station spacing and power could vary substantially. If smaller and/or lower speed vehicles are used the cost of the vehicles and power system will be somewhat lower.

The order of the costing section follows highest to lowest cost components.

1. Guideway
2. LSM stator
3. Power system
4. Vehicles
5. Controllers and software

The following sections discuss the basis of the cost estimates and a summary is given in Section 10.6.

10.1 Guideway

Guideway cost is based on our baseline design: a dual guideway with double-span 72-meter long guideway beams and pier construction. The cost estimate includes provisions for mounting the LSM and labor hours for anchoring and aligning the LSM stators to the guideway beams. Pricing assumes concrete guideway beams and piers.

A 25% contingency factor has been added in the expectation that additional expenses will be necessary to meet installation and operational requirements. Structural requirements have been met in the preliminary investigations of guideway designs but operational dynamics may dictate refinements to these designs. When a site is chosen guideway optimization simulations will dictate full beam requirements.

Cost estimates are based on information supplied by Earth Tech of Long Beach CA, and are based on their recent experience with installation of the Vancouver Skytrain, and Bankok Transit system.

10.2 LSM stator

The LSM stator is made up of two major components: laminations and windings. Included in the LSM stator are the costs for mounting and aligning the stator laminations and installing the windings. Lamination estimates are by Tempel Steel assuming that the lamination stacks are fabricated on site from stamped and spooled M19 24-gauge electrical steel. Winding estimates are based on corporate experience with producing and installing LSM windings. Although manufacturing methods for the 3-phase windings have not been determined, wound on or off site cost estimates are expected to be similar. A 15% contingency has been included.

10.3 Power system

Key components of the propulsion system are the inverters that transfer power between the DC bus and the LSM windings. For the baseline design it is assumed that there is an inverter station at every 4th pier and it contains inverters for driving the port and starboard motors for each lane of a dual guideway for 72 meters in each direction. Near stations where speed is low it is desirable to have a separate inverter for every block, and possibly use shorter blocks, but in regions where the speed is high each inverter can drive several blocks. We estimate that, on average, there will be an inverter for every two 36-meter stator blocks. The cost estimate includes the requirement for 3-phase power cables to distribute the inverter output power.

Cost estimates are based on the following assumptions:

- There is an inverter station every 144 meters and it contains 8 inverters. Each inverter has a rating of 500 kVA and operates off of a 900 VDC supply. The maximum power output per inverter is 250 kW.
- Each inverter contains a pair of 3-phase switches so it can provide power to either of two 36-meter long LSM sections.
- Power is distributed by a 3-wire DC bus with voltages of +900, 0 and -900 VDC. The power cable is installed on one side of a dual guideway.
- Rectifier stations have a rating of 1.5 MW and are located at 8 km (4.97 miles) intervals in each direction. Up to 2.25 MW is available for short periods.
- Inverter pricing includes power modules and filtering but excludes control boards which are included as part of the control system cost.

- Inverters have regeneration capability but no braking resistors.
- Inverter cooling and heating is suitable for any U.S. urban environment.

Costs for the rectifier station and power cables are based on discussions with Massachusetts Electric Construction Co. Inverter cost estimates are based on discussion with Powerex and other semiconductor manufacturers. Other costs are based on MagneMotion's experience in designing and building LSM propulsion systems. A 15% contingency has been added.

10.4 Vehicle

The baseline vehicle can carry 24 passengers seated and 12 standees or can be reconfigured to have 16 seats and 24 standees or 32 seats and no standees, if these alternatives are better for specific installations. Smaller and larger vehicles are possible but a detailed costing has not been done for other vehicle sizes.

A secondary eddy current brake is planned for emergency use and a tertiary mechanical brake is planned as an added safeguard but with the expectation that it will never be used. No detailed designs have been completed so only rough estimates are used for budgeting purposes. The vehicle body estimates are based on discussion with Hall Industries, TPI Composites Inc., CWA and others. The suspension component costs are based on discussions with MagneMotion magnet and structural component vendors.

The baseline vehicle has 4 magnet pods that include a secondary suspension suitable for speeds to 45 m/s (101 mph) but with few changes it could accommodate speeds up to 60 mps (134 mph). For speeds below 20 m/s (45 mph) the secondary suspension can be omitted.

Onboard power is provided by Inductive Power Transfer (IPT) from a guideway power distribution system. The average onboard usage is estimated at 12 kW average with up to substantially higher peak power. This power rating is primarily based on HVAC needs and can be increased if unusually hot or cold operating conditions are expected.

Preliminary estimates indicate that a 36-passenger vehicle will weigh 5.5 Mg empty and cost about $800 thousand. For comparison, a typical articulated light rail vehicle weights 40 Mg empty and costs about $2,500 thousand. The light rail vehicle has a crush load capacity of about 200 passengers, but in typical operation it only takes three 36-passenger maglev vehicles to provide the same capacity as one light rail vehicle because of the higher average speed. Thus total maglev vehicle cost is comparable to the cost of light rail vehicles with the same operational capacity, but maintenance and energy costs should be much less. The improved speed and comfort for passengers should be important selling points.

10.5 Control electronics and software

The control system includes the central control and inverter controllers located along the guideway. MagneMotion has developed very similar controllers for other applications so we are confidant that the estimates are valid.

10.6 Cost summary

Table 10.1 summarizes costs in 2004 dollars of each major subsystem. It is assumed that the installation is at least 10 km (6.2 miles) long with the expectation that costs will be somewhat higher for shorter installations. The estimated cost includes the contingency factors for component parts, but does not include civil works, shipping, stations or land acquisition costs.

Component contingencies account for uncertainties in our cost estimates and are based on discussions we have had with various vendors regarding the relative risk associated with the estimates. Even with the contingencies, *M3* costs are well below those of competing transit systems

Table 10.1. Total M3 system cost in k$/mile assuming 4 vehicles per mile.

	k$/mile	*%*
Guideway	10,131	37%
LSM stator	7,539	27%
Power system	5,102	18%
4 Vehicles	3,172	11%
Control electronics and software	1,643	6%
Total cost per mile of dual guideway in k$/mile	**$27,587**	100%

11 Continuing development

Our system development plan consists of 4 phases. The first phase calls for the construction of a 30 meter long indoor demonstration system in MagneMotion's Acton MA facility. The vehicle for the indoor demonstration system will be a test sled only. The outdoor demonstration system will consist of a 200 meter track and a full-size vehicle. Subsequent phases will incorporate a curve, switch, and finally, a longer stretch of straight track to demonstrate vehicle control and ride comfort at higher speeds. We have created a development plan and prepared cost estimates for our expanded indoor test facility. In parallel we are pursuing an outdoor location for the 200 meter track. Our goal is to find a site where the 200 meter track will become part of our first commercial installation.

11.1 Expanded indoor test facility

The existing test track has allowed us to demonstrate the viability of our design but it is too narrow and short to demonstrate key parameters such as operation at higher speed, operation on a turn with a radius of 18.3 m, and block-to-block hand-off of the propulsion control. It is desirable to develop final control systems in a controlled indoor environment and this task is a first step towards developing the indoor facility. The proposed indoor demonstration facility will include approximately 24 meters of straight guideway and a 30º arc segment (9.6 meter arc length) of an 18.3-meter radius curve. The guideway will be built with a full gauge rail spacing of 1.7 meters. The port and starboard LSMs will have independent position sensing and inverter drives. The guideway will be split into two blocks to allow for block-to-block handoff. The vehicle will consist of full-scale levitation assemblies, without secondary suspension. Although this vehicle will be a 'sled' with no shell, the size of the vehicle and levitation assemblies will be the same size as those planned for a 12-passenger vehicle. Maximum travel stroke will be approximately 25 meters, including travel into the curved section.

11.2 Site development

MagneMotion has initiated preliminary discussions concerning potential sites for a test facility and an initial commercial installation. We continue to pursue these leads with a focus on finding funding sources and locations for high-speed testing. There have been ongoing discussions with state agencies and government officials. We have an outline of a preliminary proposal prepared for the Massachusetts Bay Transit Authority for the use of property along side a railroad maintenance facility in Cambridge, MA. We have had extended discussions with Massachusetts Development, a private/public agency, that has been tasked with the redevelopment of a former Air Force base. We also continue discussions with other private institutions regarding the replacement and extension of monorail systems, but are not free to discuss details due to non-disclosure clauses.

Perhaps the most compelling site discussions we have had are those involved with a plan to redevelop East Providence, RI. We believe this application comes closest to meeting the FTA goal of building a demonstration track that could become part of a commercial system.

The East Providence Development Project will require a public transit system that will extend from Providence proper, through an existing tunnel to the Seekonk River, over the river via an existing bridge, and then extend 2 miles both east and west along the banks of the Seekonk River. The initial phase of the transit system will be approximately 6 miles, with other extensions a possibility.

11.3 Converging interests for East Providence Development Project

The FTA has indicated that their objectives for a funded project are threefold: a maglev vehicle that can carry passengers, a travel distance long enough for passengers to experience the ride quality of a maglev system, and a demonstration site that can be tied to a future commercial system. We believe this opportunity may meet all the FTA goals.

The following parties have an interest in establishing MagneMotion's Urban Maglev System as the transit system of choice for East Providence, Rhode Island.

MagneMotion

MagneMotion is seeking a location to build its outdoor demonstration system. The initial requirements call for a right of way at least 200 meters long and wide enough to construct an elevated guideway; a facility that could house the vehicles, engineering lab, and perhaps the first 20 meters of track; and a wide enough section of land at one end of the right of way to construct an 18.3 meter radius curve. Additional land would be needed to construct a switch, and to extend the line for up to an additional 1,500 meters. The total development effort for the indoor demonstration system and the 200 meter outdoor demonstration system, including a full sized vehicle, curve and switch, is approximately $15 Million. The extended line would require additional funding. MagneMotion is currently speaking with different investment groups to raise a minimum of $10 Million in order to complete the $15 Million funding requirement. There is no assurance that additional financing can be obtained. The prospects of receiving both an FTA grant and private investment are significantly enhanced if a test system leads directly to a commercial maglev system.

East Providence Development Team (EPDT)

EPDT is planning a major development project in East Providence that would include residential, commercial, and industrial buildings. It is believed that a public transportation system is essential for travel to and from Providence proper, and up and down the banks of the Seekonk River. In addition, EPDT would like to make this a high profile project to create interest from investors, politicians, and potential tenants. EPDT would also like to secure long term relationships with growth companies that would establish manufacturing operations in the new development.

RI Economic Development Commission (RIEDC)

RIEDC would like to attract industry and stimulate job growth in the East Providence area.

RI DOT

RI DOT would like to enhance public transportation, reduce traffic congestion, and provide a solution to the transportation requirements that would arise from a massive development project in East Providence. RI DOT may have access to other sources of government financing

RI Congressional delegation (RICD)

RICD would like to enhance economic development in the State. RICD has the ability to secure earmarked funds in the next budget for both development and deployment. RICD also has the ability to strongly endorse any FTA grant contemplated now or in the future.

Private investors

Private Investors would like their investment in Maglev augmented by government grants, and assurances that a first installation that could generate revenue and plant the seeds for a positive return on their investment is near term. Equally as important, investors in EPDT would like to ensure easy access to their development to ensure a high level of occupancy and traffic to retail outlets.

TPI Composites Inc. (TPI)

TPI, a RI based company and a manufacturer of composite vehicles, has been working with MagneMotion to design and build their Maglev vehicles. Any efforts to bring MagneMotion's Maglev system to a commercial state would greatly enhance TPI's business prospects. TPI is open to relocating to East Providence and expanding its manufacturing capabilities with the advent of a sizeable Maglev vehicle business.

11.4 Action plan for East Providence

Public transit is an engine of economic growth in communities throughout America. Creating development-orientated transit can boost and urban maglev technology and encourage economic growth. Our plan represents a public/private partnership that meets the goals of stake holders in the funding of public transportation, and the re-development of East Providence. MagneMotion will submit a proposal to the FTA. The proposal will contain the first two phases of our development plan mentioned above. The corner stone of this plan will be the development of a demonstration system in East Providence

12 Conclusion

The *M3* project has met the objectives set out in the Introduction. The resulting maglev system has applicability over a wide speed and distance range. The key results of the design and prototype demonstration are as follows:

- Simplified suspension using permanent magnets with a 20 mm magnetic gap and only one set of magnets for suspension, guidance and propulsion.
- Lightweight vehicles operated with high speed and acceleration take advantage of the maglev potential and lead to lower guideway and propulsion system cost.
- Control using vehicle clusters with vehicle headway as short as 4 seconds using the ‘safe-follower” strategy and cluster spacing as required to achieve a “brick-wall” control between clusters for safe operation at all speeds. Clustering allows capacities of 12,000 pphpd with 36 passenger vehicles.
- Energy consumption is expected to be less than half that of conventional transit designs with comparable performance.
- Reduced guideway size and minimal noise reduce environmental impact.
- Cost for a complete system is approximately $25 million per mile excluding vehicles, stations and land acquisition.
- The design can be modified to use larger and faster vehicles so as to be competitive with high-speed rail and Transrapid maglev.
- The performance of the design is accurately characterized by computer models and simulations so actual performance will be close to design expectations.
- MagneMotion has high confidence that a commercial system will be competitive with all conventional guideway-based transportation systems.

13 Acknowledgement

MagneMotion is pleased to acknowledge the many individuals, companies and institutions that have contributed to the work described in this report. We are especially appreciative of the FTA staff involved in the Urban Maglev Project, including particularly Venkat Pindiprolu, Jim LaRusch and Mary Anderson and their consultant Gopal Samavedam (Foster-Miller). We thank Walter Eggers (EarthTech) and Scott Phelan (Texas Tech) for their contributions to the guideway development. Pip Shepley from Massachusetts Electric Contractors has contributed to resolving guideway electrification issues and Bob Wind and others at TPI Composites Inc. have contributed ideas concerning vehicle design.