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EVALUATION OF A FLYWHEEL-POWERED SHUTTLE CAR

FINAL TECHNICAL REPORT AS OF AUGUST 25, 1978

This report represents work on a program that was originated by the Interior Department's Bureau of Mines and was transferred to the Department of Energy on October 1, 1977.

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FOREWORD

This report was prepared by the General Electric Company, Engineering and Manufacturing Engineering, Schenectady, New York under U.S. Department of Energy (DOE) Contract No. ET-77-C-01 8890. The contract was initiated under the Energy Program of the U.S. Bureau of Mines and transferred to the U.S. Department of Energy effective October 1, 1977. It was administered under the technical direction of Tom Zemo as D.O.E. Project Manager. Mr. David Williams was the Contract Specialist.

The contract work was performed during the period December 10, 1976 to July 7, 1978, and the initial draft of the final report was submitted in February 1978.

The work was performed by Engineering and Manufacturing Engineering (E&ME), a part of General Electric Corporate Consulting Services. Key contributors were Mr. Pierre H. Boucheron and Mr. Dallas E. Cain. A number of other E&ME personnel contributed in specific areas. The Program Manager was Mr. Alan S. Rubenstein.

Expert consultation was provided by Mr. Charles B. Manula of Pennsylvania State University and by Mr. Leon J. Goldberg of Schenectady, New York. Jeffrey Manufacturing Division of Dresser Industries was the subcontractor on the vehicle portion of the work.

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ABSTRACT

The objectives of this contract were to study and evaluate the practical application of flywheel-stored energy devices to shuttle cars in underground coal mining and to study and evaluate the most practical methods of charging, recovery, and transmitting flywheel energy so as to provide power for the necessary functions of a shuttle car. The general conclusions of the study indicate that the mine mission requirements can be fulfilled with a flywheel energy storage system which can be designed within the present state-of-the-art, that a flywheel system can yield sufficient economic benefits to warrant a mine demonstration, and that there is promise of safety improvements due to elimination of the trailing cable presently used. In addition, the study indicates that specific operation problems associated with a flywheel-powered vehicle, like emergency movement of the vehicle and transmission of energy from the wayside to the vehicle, can be satisfactorily solved.

Specific studies leading to the above conclusions have been conducted, and a baseline flywheel energy storage system recommended.

Section 1

EXECUTIVE SUMMARY

Shuttle cars are transporting more than 90 percent of the coal in U.S. underground mines today from the face where it is mined to the first outby dumping point, and about 70 percent of these vehicles are powered through a trailing cable from the reel on the shuttle car to a stationary tie-off point. This cable is a source of delays due to cable breakage and replacement and represents a safety hazard to mine personnel. In addition, the requirement for a trailing cable restricts the path by which the car moves from the loading point to the dumping point, since the forward and return trip must use the same route. This tends to limit to two the number of shuttle cars which can be efficiently used.

To eliminate the trailing cable, battery and diesel-powered shuttle cars have been developed and used to a limited extent. The primary limitation of the battery-powered car is that the battery energy capacity is often insufficient to propel the car for a full shift when the bottom conditions are adverse or when the batteries are old. The diesel car has not found general acceptance in U.S. underground coal mines, and no change in this appears to be forthcoming at this time.

Development and testing of a low-emission steam engine for underground coal haulage has been conducted under funding by the USBM and DOE, and work to date is presently being evaluated under a separate DOE contract.

The recent extensive development activity in flywheel technology offers another possibility for eliminating the trailing cable. If a flywheel propulsion system proves practical for underground coal shuttle cars, the trailing cable could be eliminated, safety could be enhanced, increased flexibility in routing could be achieved including longer tram distances, and more shuttle cars could be used to increase productivity. The purpose of this contract was to evaluate the practicality of the flywheel as a power source for shuttle cars.

The Executive Summary, Section 1, contains a project summary, details the conclusions reached, and makes a recommendation for a baseline flywheel energy storage system and a program to incorporate in a vehicle for a mine demonstration a flywheel propulsion system.

The remaining sections of the report discuss the technical work leading to the conclusions. The appropriate sections address the energy required for the mission, flywheel technology, face haulage economics which impact the viability of an energy storage system, the baseline conceptual equipment and the wayside-to-vehicle interface, and appropriate trade-offs in the baseline equipment selection.

Also addressed are safety considerations for conventional shuttle cars and appropriate special mine practices required for flywheel energy storage systems.

1.1 PROJECT SUMMARY

1.1.1 Objective for Phase I

The objective of Phase I is to study and evaluate the Practical Application of Flywheel-Stored Energy devices to face haulage vehicles in underground coal mining, and this was pursued using the following work sequence.

Energy Storage Determination

The energy storage requirement for an on-board flywheel system was determined by utilizing the Pennsylvania State University/United States Bureau of Mines (PSU/USBM) Underground Mine Haulage Simulator to determine energy requirements for a broad range of bottom conditions and seam heights. These results were analyzed and an energy requirement selected, that in the judgment of the subcontractor, C.B. Manula of PSU and the TPO, represented 90 percent of the potential shuttle car applications.

Study of Flywheel Technology

Shapes, sizes, and materials commensurate with the available space in both conventional shuttle cars and tractor-trailer cars were investigated. The key factor is that conventional shuttle cars have only the cable reel space for the flywheel, resulting in a limitation on the maximum flywheel diameter that can be accommodated. With flywheel speeds limited by other considerations such as bearings and motor speed, the energy density utilization of the flywheel is low, limiting the energy storage capability. A tractor-trailer car, on the other hand, has space in the tractor for a larger diameter, higher energy flywheel. Steel wheels of several shapes, and composite materials which can be made available for construction, have been studied, for an evaluation of the state-of-the-art.

The conclusions and recommendations were formed considering the desirability of near-term application of a flywheel system for demonstration.

Face Haulage Economic Analysis and System Trade-offs

Annual operating costs and productivity in terms of cost/ton have been explored for shuttle cars of the following types:

- Conventional
- Diesel
- Steam
- Battery
- Flywheel

In addition, consideration has been given to the flywheel system for both two and three car utilization with several different flywheel charge times. There are a number of significant tradeoffs, including the size of the charging system and the charging motor if an electrical charge is used, two versus three car operation, and the load capacity of the flywheel car -- all versus cost/ton of coal produced.

The configuration of the flywheel itself does not enter into the economic analysis. A wayside inverter and on-board motor/alternator flywheel package have been used to evaluate charge time versus annual cost and productivity.

The PSU/USBM simulator was used to determine tons of coal produced for the various options.

Flywheel Package and Electrical Package Design Concept

The first phase of the study assumed an electrical charging system for the flywheel system. An electrical charging system requires a motor/alternator on-board and mechanically connected to the flywheel. During charging, this motor is connected to the wayside power to convert the electric power to flywheel energy. The motor also acts as an alternator to convert flywheel energy to propulsion power. The Phase II work evaluates a mechanical charging system to explore whether such a system makes a mechanical connection from the flywheel to the vehicle drive system more attractive.

A motor type was selected based on previous work and conceptual designs were made for various charging times and for several frequency ratings.

The motor/flywheel assembly was configured with alternate flywheels to arrive at an optimum concept, considering pertinent design and manufacturing limitations.

Alternate Flywheel Systems for Shuttle Car

A study was conducted to make a preliminary evaluation of alternate flywheel systems for shuttle cars, including:

- Flywheel/hydromechanical drive
- Flywheel/mechanical drive
- Flywheel/hydraulic
- Flywheel/torque converter

Safety Impact

An investigation of the safety considerations in tethered shuttle cars was conducted, primarily with an extensive literature search, and conclusions were made regarding improvement to be expected with flywheel powered cars.

1.1.2 Objective for Phase II

The objective of Phase II is to study and evaluate the most practical methods of charging, recovery, and transmitting flywheel energy to provide for the necessary functions of the shuttle car. This objective was pursued using the following work sequence.

Flywheel Package and Electrical System Design Concept

The work performed in Phase I on the flywheel package was extended by developing a concept design for the mounting of the flywheel in a shuttle car and by further investigating gyroscopic forces, bearing design and lubrication, and by considering in some depth the windage losses of the flywheel-motor assembly and the relationship of these losses to air pressure in the flywheel enclosure.

Face Haulage Economic Analysis and Systems Trade-offs

The work performed in Phase I was extended to consider the feasibility of lengthening the tram distance and the relationship of tram distance to the required flywheel size. Critical vehicle components were evaluated based on spin up time requirements. A study was conducted to evaluate the desirability of utilizing regeneration on the vehicle.

Wayside-to-Vehicle Interface and Alternate Wayside Charging Systems

Alternate electrical and mechanical charging concepts were explored and evaluated, and a recommendation of an optimum concept was formulated, including specifications for the preferred system. A Consultant, Mr. Leon Goldberg, who is an expert on electrical contact design, contributed to this task.

Preliminary Design of a Flywheel Package on Shuttle Car

Based on the results of Phase I work, indicating the desirability of utilizing a tractor-trailer shuttle car, a subcontract was awarded to Jeffrey Mining Machinery Division of Dresser Industries to advise on the selection of an appropriate tractor-trailer shuttle car for incorporation of the flywheel package, to make preliminary layouts of the on-board flywheel equipment, and to assist in identifying problem areas in the vehicle and its subsystems which must be addressed in future design phases.

Emergency Procedures

A study was conducted to evaluate alternate means of recovering a stranded vehicle and to formulate a recommendation for the optimum method.

1.2 CONCLUSIONS

The general conclusions of the study indicate that the mine mission requirements can be fulfilled with a flywheel energy storage system that can be designed within the present state-of-the-art, that a flywheel system can yield sufficient economic benefits to warrant a mine demonstration, and that there is promise of safety improvements due to elimination of the trailing cable presently used. In addition, specific operation problems associated with a flywheel-powered vehicle, like emergency movement of a discharged vehicle and transmission of energy from the wayside, can be satisfactorily solved.

The specific conclusions from the study may be grouped as follows:

- Energy storage requirements
- Economic viability of a flywheel powered shuttle car system
- Flywheel package and electrical systems package
- Wayside to vehicle connector
- Emergency procedures
- Alternate shuttle car drive systems
- System safety
- Selection of vehicle for demonstration

1.2.1 Energy Storage Requirements

The Pennsylvania State University/U.S. Bureau of Mines (PSU/USBM) Underground Mine Simulator was employed to determine the energy required to operate a shuttle car over the longest tramming route in a typical cut plan. A number of computer runs were made using a six-entry cut plan, four, six and eight foot seam heights, and a range of bottom conditions. In addition, other sources of data were studied, including actual measurements of shuttle car tramming and auxiliary power requirements at several mines, and other studies as indicated in Section 2.

It was judged that 90 percent of the face haulage applications would be included within the following conditions:

• Seam height	6 ft
• Rolling resistance	300 lbs/ton
• Grade	5%
• Coefficient of traction	0.4

The simulation, plus consideration of the other data in Section 2, led to the conclusion that a 4.5 kW hr usable energy (6.0 kW hr total energy) flywheel is required. This energy would be adequate for vehicles with gross weights of about 20 tons, which would include the recommended Jeffrey Steam Ramcar, operating with the above grade and bottom conditions.

Further study also showed that a flywheel designed to store 4.5 kW hrs of usable energy could be accommodated in the engine or battery compartment of a tractor-trailer type of shuttle car.

Studies have been conducted by C.B. Manula of Penn State University to determine the range of severity of bottom conditions encountered in underground coal mines. Results from these studies indicate that actual bottom conditions reported are somewhat less severe than those used in the simulations above. Eighty percent of the cases reported have bottom conditions which are less than:

• Rolling resistance	200 lbs/ton
• Grade	3%
• Coefficient of traction	0.44

Applying these bottom conditions to the PSU/USBM Simulator shows that a 3.0 kW hr usable energy flywheel would provide the energy required for the longest tramming path in the selected cut plan. It may be possible to fit a 3.0 kW hr flywheel into a conventional shuttle car. If so, there is a potential cost benefit of about $.50 per ton to be realized by eliminating the turnaround times of the tractor-trailer type of car.

The same study showed that average bottom conditions are:

• Rolling resistance	165 lbs/ton
• Grade	2.07%
• Coefficient of traction	0.44

Investigation with average bottom conditions has shown that a 4.5 kW hr usable energy flywheel can be utilized for tram distances as much as 80 to 100 percent greater than the six-entry cut plan used in the economic evaluation. A 7.5 kW hr usable energy flywheel is required for worst-case bottom conditions at the extended tram distances. To maintain productivity for these tram distances, the number of cars must be increased to minimize the continuous miner wait time.

The study considered a 4.5 kW hr flywheel size with attention to 6.0 kW hrs. If it is desired to use larger flywheel sizes, design justification beyond that which was considered in this study must be obtained.

1.2.2 Economic Viability of a Flywheel System

Cost/Ton and Rate of Return

The PSU/USBM simulator with a six entry cut plan for the seam height and bottom conditions representing 90 percent of the applications was used to determine tons produced/shift for a flywheel-powered shuttle car for a number of variables:

- Number of cars used(2 or 3)
- Type of car (conventional or tractor-trailer)
- Spin-up delay (30, 60, and 90 seconds)
- Car capacity (200, 236, and 270 cubic feet or 5, 5.9 and 6.8 tons)

The cost/ton was determined by a detailed breakdown of mine cost factors, and the results were compared to a base case of two conventional tethered shuttle cars.

The results show that the cost/ton and rate of return for the flywheel system on a tractor-trailer car are superior to the base case when used as follows:

- With equal capacity cars in a three-shuttle car system with average charge time up to 90 seconds
- With 14 percent greater capacity in a two-shuttle car system with a 30 second or less average charge time

The results also show the cost/ton for a flywheel system on a conventional shuttle car (if it were possible to fit the required size flywheel) is superior to the base case when used as follows:

- With equal capacities in a two-car system with average charge times of 60 seconds or less
- With equal capacities in a three-car system with average charge times of 90 seconds or less

The improved economic results achieved with a flywheel on a conventional car is due to its ability to operate bi-directionally, thus avoiding turnaround times.

Annual Operating Costs

A study of annual operating costs of a battery-operated Ramcar, a diesel-powered Ramcar, the Jeffrey development steam car, a conventional tethered car, and the projected flywheel car was made.

The results show that annual operating costs, exclusive of production labor, of a flywheel system is equivalent or advantageous to all systems.

1.2.3 Flywheel Package and Electrical Systems Package

Flywheel Package and Mounting

Based on the 4.5 kW hrs of required usable energy storage, it is feasible to design a flywheel package consisting of a suitable flywheel directly connected to an inductor motor/alternator in a sealed package, and to provide a design for mounting the system in a tractor-trailer vehicle.

Inductor Motor/Alternator

A charging motor/alternator rated at 203 kW will provide the maximum power required for vehicle operation and will require 80 seconds to put a 4.5 kW hr charge into the flywheel. However, the full 80 seconds charge time will seldom be used since with average bottom conditions a shuttle car trip will use considerably less than 4.5 kW hrs, and the charge time will be correspondingly less. For average bottom conditions, the average charge time will be less than 30 seconds. Therefore, there does not appear to be sufficient reason to utilize a larger and heavier motor and inverter that is sized to permit a 4.5 kW hr charge in 30 seconds and is oversized for the vehicle energy usage requirements.

Flywheel

Studies were performed on five different flywheel shapes using both composite materials and steel for large diameter flywheels suitable for tractor-trailer cars and smaller diameter flywheels suitable for application to conventional shuttle cars.

For the larger diameter flywheels, composite construction offers weight advantages which will ultimately be realized as the state-of-the-art progresses. However, it is concluded that the state of development of composites is not sufficiently advanced to provide a low-risk design for a near-term development for which the prime purpose is to demonstrate the economic and practical viability of a flywheel-powered shuttle car system. A steel flywheel should be used for larger diameters,

For a smaller diameter flywheel, it is concluded that there is no advantage to using composites, and that a steel cylindrical disk flywheel without a shaft hole is the optimum construction.

Electrical Systems

An on-board solid state diode rectifier control system is suitable for converting flywheel motor/alternator energy to power the vehicle during flywheel discharge. A solid state wayside inverter shared among vehicles is required for charging the flywheel.

Kinetic and potential energy available is insufficient to warrant recovery. Therefore, simplicity of design should not be compromised for this purpose.

Mine Power Center

A study of mine power centers has shown that a 750 kVA mine power center can accommodate a charging motor/wayside inverter load of 200 to 250 kW without a serious voltage drop which might adversely affect other mine equipment. This is consistent with the requirements of the 203 kW charging motor.

1.2.4 Wayside-To-Vehicle Connector

Electrical and mechanical wayside to vehicle connection systems were studied. To accomplish an early demonstration of a practical flywheel powered shuttle car, an electrical interface between a wayside electrical system and the vehicle for spinup power will result in the simplest and least risk development program. This is due to the relative simplicity and flexibility of a wayside charging station and vehicle to wayside connector.

1.2.5 Emergency Procedures

A number of alternate methods of rescuing a stranded shuttle car were studied. Towing is the simplest and most practical solution. It is concluded that even if other more sophisticated means are provided, towing would be the method used by mine personnel.

1.2.6 Alternate Shuttle Car Drive Systems

A study of the possibilities of using the flywheel with a hydromechanical drive, an electromechanical drive, a hydraulic drive, and a torque converter drive indicates that all of these systems are potentially realizable with possible savings in weight and size for all but the electromechanical system. However, an electrical system wherein the flywheel power is converted to electrical energy through a motor/alternator is superior to mechanical systems when the complexity of flywheel connection to the vehicle drive system is considered.

1.2.7 System Safety

A review of several studies and data available on the subject of mine accidents indicates that shuttle car trailing cables represent a major safety hazard which could be eliminated with an internally powered shuttle car. The flywheel-powered shuttle car offers the promise of eliminating the trailing cable hazard. To avoid the creation of other safety problems the flywheel containment, charging station, and charging interface must be conservatively designed with personnel safety a prime consideration.

The study indicates that of the 750 to 1000, or more, mining electrical accidents and injuries per year some 40 percent occur in the face area of underground mines. Additionally, nearly 31 percent of the electrical accidents and injuries are caused by cables. While shuttle car cables represent some 20 to 40 percent of the electrical cables in the face area, the very nature of their use - constantly flexing and rubbing against the ribs and bottom - makes them much more prone to failure. It is conservatively estimated that 50 percent of the 90 to 120 electrical accidents and injuries caused by electrical cables in the face area of underground coal mines may be attributed to shuttle car tether cables. Therefore, it may be concluded that the introduction of the internally powered flywheel shuttle car holds the promise of eliminating at least 45 to 60 electrical accidents and injuries per year in these mines. The fatality data reveal that 38 percent (3 out of 8) of the cable-related fatalities were due to shuttle car operations. At least two of these were the direct result of splicing activities.

1.2.8 Selection of Vehicle for Demonstration

The desired size of the flywheel indicated that a tractor-trailer shuttle car would be the most desirable vehicle for a demonstration of a flywheel energy storage system. A study was made of several vehicles with approximately 270 ft^3 capacity, designed for a 4 ft to 6 ft seam height with space suitable for the 43-inch diameter flywheel. While several vehicles were possibilities including both two-wheel and four-wheel drive designs, it was concluded that a four-wheel drive vehicle would be desirable for demonstration purposes to provide the best chance for success in adverse bottom conditions. It was also concluded that the selected vehicle should require minimum modifications so that a demonstration program could focus upon proving the flywheel energy storage system in a mine rather than proving new vehicle concepts.

Studies of operator visibility as a function of flywheel rating have shown that a smaller usable energy flywheel (4.5 kW hrs) projects above the vehicle less than a higher usable energy flywheel, so that using the smaller energy flywheel in a high-seam mine would result in better visibility (as would be expected). In seams of 60 inches or higher, the operator's seat can be raised over the standard location. For operation with greater than 4.5 kW hrs in seams of less than 60 inches, a design trade-off decision for visibility is required.

1.3 RECOMMENDATIONS

The following subsections summarize the recommended baseline flywheel energy storage system, including the flywheel package, the electrical systems, the wayside to vehicle interface, and the vehicle. Recommendations for follow-on work are also included.

1.3.1 Recommended Baseline System

The recommended baseline system is summarized below:

Flywheel Package

The flywheel package illustrated in Section 4 consists of a steel flywheel with a modified constant stress shape, directly coupled to an inductor motor/alternator. The flywheel has 6.0 kW hrs of stored energy at 10,000 rpm and produces 4.5 kW hrs when the speed drops to 5,000 rpm. A larger flywheel can be designed, made up of additional conical sections to accommodate the required energy storage for a longer tram distance where worst case bottom conditions exist.

The inductor motor/alternator, rated at 203 kW, is a solid rotor synchronous machine with both the ac and dc windings on the stator. The rating is selected to allow a 4.5 kW hr recharge of the flywheel in 80 seconds. This motor rating is also adequate if a longer tram path, with higher than a 4.5 kW hr requirement is desired; since more shuttle cars would be used, longer spin-up times would not significantly affect miner wait time.

The flywheel is guarded by a two-inch steel containment ring.

The inductor motor/alternator-flywheel is operated in a partial vacuum at a pressure of 0.01 to 0.05 psia to minimize windage losses.

The package is mounted to the vehicle, utilizing a soft suspension which allows the flywheel to move in response to flywheel precessional torques and thus to protect the spin axis bearings from excessive loads.

Electrical Systems

The on-board electrical system, Section 4, consists of a diode rectifier which converts the inductor alternator voltage to dc when the flywheel is supplying power to the vehicle drive system. During this time, the field of the inductor machine is regulated to provide approximately constant voltage.

The wayside system consists of a load commutated inverter and control which is powered by dc voltage from the mine power supply. The inverter provides variable frequency, variable voltage to the inductor motor during charging of the flywheel.

Wayside-to-Vehicle Connector

The recommended conceptual wayside connection scheme, shown in Section 8, consists of a contact device operating from the wayside which operates to engage a mating connector on the vehicle. The contacts do not make or break under load. A semi-automatic positioning system is utilized to help steer the vehicle into the appropriate position for recharging.

Vehicle and Equipment Layout

The recommended vehicle for demonstration purposes (Appendix D) is the four-wheel drive Jeffrey Steam RAMCAR vehicle with the engine removed and the resultant space used for the flywheel package, which includes the flywheel, on-board electronics and vehicle portion of the connector.

1.3.2 Recommendations for Follow-on Work

The study has identified a number of design areas which make it desirable to proceed toward a demonstration with an orderly phased program which includes design and testing phases for the flywheel, motor alternator and on-board and wayside electrical systems.

The areas which are expected to require significant design effort are identified in the report. Some of these are listed below to illustrate the effort required to produce a satisfactory prototype. These areas should be satisfactorily resolved before committing to a mine demonstration.

Design Area	Text Reference
Flywheel Stress Analysis	4.4.8
Flywheel Construction Including Heat transfer	4.4.8
Critical Speed Design	4.4.8
Bearing and Lubrication Design	4.5.7
Vacuum System Design	4.5.7
Energy Storage Cooling Under Transient Conditons	4.5.7
Gyroscopic Action of Flywheel	4.6.7
Mounting of Flywheel Enclosure to Vehicle	4.6.7
Utilizing Capsule Enclosure to Augment Safety	4.6.7
Flywheel Energy Storage-Vehicle Design	9.5
Solid State Wayside Charging Design	4.7
Vehicle-Wayside Connector Design	8.4
Vehicle Modification Tasks	D.7

The study has shown that the recommended vehicle would require relatively little development and design modification to operate with a flywheel energy storage system. Therefore, it is appropriate to phase the program so that the flywheel energy storage system is designed first and that commitments of remaining phases are contingent upon satisfactory laboratory demonstration of the energy storage system.

A brief summary of the recommended phases follows.

Phase A - Design and Laboratory Testing of the Flywheel Energy Storage System

Using the baseline system recommendation as a starting point, design the flywheel, inductor motor, flywheel package, wayside electronic equipment, and on-board electronic equipment. After suitable design reviews, which are important from a safety standpoint, construct a prototype system to be laboratory tested. The test program will require a suitable laboratory test bed and a test plan aimed at addressing identified design problem areas.

The design and laboratory testing plan should allow time for modification and retesting of the prototype for "debugging" and design optimization.

Phase B - Vehicle Design and Testing with Flywheel Energy Storage System

Based on successful completion of Phase A, the prototype flywheel system should be rebuilt, if required. The vehicle would be procured and necessary modifications made to the structure, drive system and control system. Integration of the flywheel system with the vehicle is required in areas such as flywheel system mounting, cooling, and integration of the vehicle operating systems with the flywheel system. An aboveground demonstration should be planned and executed and the total system modified as necessary to achieve satisfactory performance.

Implementation of this phase requires selection of the mine test site since a number of factors must be decided including seam height, haulage clearance, and mine operating policies.

Phase C - Construction of Required Number of Shuttle Cars and Flywheel Systems

Based on successful completion of Phase B, construct one or several more flywheels and procure one or several more shuttle cars if it is decided to demonstrate a multi-vehicle system. This will require additional on-board flywheel systems but only a single wayside system.

Phase D - Mine Demonstration

Demonstrate productivity, reliability, performance, and operator acceptability in a mine section utilizing only flywheel-powered shuttle cars for a three month period. This will require close integration with the mine prior to the start of the demonstration program and possible installation of additional equipment or electric service.

Other Work

It has been shown that composite flywheels offer ultimately a lower weight, lower cost flywheel when the economic and performance problems are solved. It is, therefore, recommended that this area of technology be monitored and that significant advances be incorporated in the above program if it appears desirable within the time and funding constraints.

Section 2

1 MISSION ANALYSIS AND ENERGY STORAGE REQUIREMENTS

To make an evaluation of a flywheel system, it was deemed necessary to determine the energy storage requirement, since this is the main factor in sizing the flywheel, all associated components, and the suitability of a particular vehicle. This section documents the mine data gathered, the mine simulation results, and the resultant flywheel sizing to provide a basis for the remainder of the evaluation.

2.1 SUMMARY AND CONCLUSIONS

The energy drain on a shuttle car is caused by the tire losses due to rolling resistance and the work required to move the shuttle car up a grade. The amounts of energy involved in aerodynamic drag and acceleration are so small that they are not considered in this study.

With a knowledge of the parameters which needed to be measured, trips to mines yielded data on energy consumed during a mission. A mission is composed of four segments: unload, tram to miner, load, and tram to unload. There are wait times and auxiliary loads that occur throughout the mission.

The PSU/USBM simulator was then used to determine the effects of various bottom conditions, slopes, and seam thicknesses on the energy requirements for the flywheel. The data from the mine visits were used to translate the output to the simulator into kW hrs for sizing the flywheel.

After this work had been completed, a meeting was held with the Technical Project Officer, C.B. Manula of Pennsylvania State University, and Mr. A.S. Rubenstein of the General Electric Company (GE). Consideration was given to the energy values resulting from the simulations as well as those calculated from the mine data in Section 2.2, including information obtained from Lee Engineering Division-Consolidation Coal Co. The results of the simulations and calculations from mine data were reviewed and it was agreed that flywheel design should be based upon a usable energy of 4.5 kW hrs (6 kW hrs total). This figure represented a balance between the high value obtained from the simulations with bad bottom conditions and thick seam coal and the relatively low requirements indicated by the mine data. It is probable that a 6 kW hr total energy flywheel may be too large to fit in the cable reel compartment, which is the only space available in a conventional shuttle car.

From Figure 1 it can be seen that the 40 hp motor used in the Mathies Mine shuttle cars has a peak load of 200 amps at 600 Vdc. Since 75 hp motors are also available and used, a flywheel with 4.5 kW hrs useful energy will have to be designed for a 375 ampere peak load with 600 volt dc input to the inverter or output from the rectifier. This peak rating will determine the sizing of the flywheel spin-up motor when the charge time is determined by the duty cycle, but for short (30 second spin-up under all conditions) charge times, the peak charging current is the dominant design parameter of the motor.

2.2 MINE DATA

Preliminary Assessment of Required Data

To determine what data had to be obtained from the mines, a preliminary assessment was made of the factors which would consume flywheel energy during a shuttle car mission profile and hence the energy required for spin-up, or recharging.

These factors are:

- Aerodynamic drag
- Kinetic energy of motion
- Potential energy due to grades
- Efficiency of drive system
- Tire rolling resistance losses

Factors pertinent to regenerative braking:

- Kinetic energy available
- Potential energy from grades

Based on the energy requirements for a 40,000 lb vehicle to travel 500 ft with a 5% grade with a top speed of 3 mph, the following are the calculated energy requirements:

Factor	kW hrs
Aeronautic drag	9.0×10^{-5}
Kinetic energy	4.5×10^{-3}
Potential energy	0.4
Tire loss $\left(\text{Rolling resistance} = \frac{100\#}{1{,}000\#}\right)$	0.75

It can be seen that aerodynamic drag and kinetic energy of motion can be ignored and that prime attention must be devoted to operation with grades and to the effect of tire rolling resistance. These two factors will primarily determine energy usage. The drive train efficiency will be important in sizing the flywheel since the energy needed at the wheels must be divided by the efficiency to derive the energy needed in the flywheel. Regenerative braking will not be an effective means of energy recovery since essentially all of the potential energy available will be absorbed in overcoming rolling resistance, even at a 10% grade. On top of that, the efficiencies of the drive train, inverter, and spin-up motor would further detract from the potential energy available for spin-up.

Assumed Mission

The following is an assumed mission to obtain a preliminary assessment of flywheel energy requirement with a 30,000# empty vehicle and 45,000# loaded vehicle:

- Empty vehicle - down 20% grade for 500 ft in 90 seconds
- 30 second load time
- Loaded vehicle up 20% grade for 500 ft in 105 seconds
- 30 second unload time

For this example, the breakdown of energy usage is as follows:

	All Numbers in kW hrs			
	Empty Vehicle Travel	Load	Full Vehicle Travel	Unload
Tire Loss $\left(\text{assume } \frac{100\# - 300\#}{2{,}000\#}\right)$	0.57 - 1.7	-	0.85 - 2.5	-
Grade (assume 20%)	1.1 (braking - no regen.)	-	1.66	-
Maneuvering during Loading/Unloading		0.5*		0.5*
	0.57 - 1.7	0.5	2.51 - 4.16	0.5

Total	4.07 - 6.86 kW hrs Required at Wheels
Assume Drive Efficiency 0.69	5.90 - 9.94 kW hrs Required in Flywheel

*Based on unconfirmed current measurements

National Mine Service Data

National Mine Service has supplied an oscillogram (Figure 1) of the dc current into a shuttle car for a mission at the Orient #4 mine of the Freeman Coal Company, W. Frankfort, Ill. It is included to provide documentation of the type of operation and electrical loading that is required by the shuttle car.

Table I shows the reduction of the data from Figure 1 to provide energy for each operation and total mission energy. For a flywheel vehicle, the mission energy (kW hrs) represents the energy that must be delivered to the car by the flywheel-generator-power conditioning package. For this mission, the efficiency of

Table I

National Mine Service Shuttle Car Mission for Orient #4 Mine, Freeman Coal Company

	Time (sec)	Energy (kW hrs)	Mission Energy (kW hrs)
Unload	43	0.34	-
Tram to miner	62	0.66	-
Load	48	0.22	-
Tram to feeder	86	0.75	1.97
Unload	51	0.44	2.07
Tram to miner	75	0.64	2.05
Load	44	0.17	2.00
Tram to feeder	61	0.66	1.91
Unload	53	0.44	1.91

Note: The dc shuttle car when idle (motor off) uses 30 watts with the lights off and 350 watts with the lights on.

The ac shuttle car when idle (motor is at 1/2 speed) uses 4,500 watts with the 45 hp motor or 10,000 watts with the 75 hp motor. The auxiliary loads include lights, hydraulic pump, motor windage losses, and transmission losses.

this package would be considered so that the required kW hrs from the flywheel = $\frac{2.07 \text{ kW hrs}}{\text{Efficiency}}$. Of course, this makes no allowances for auxiliary load losses during idle time.

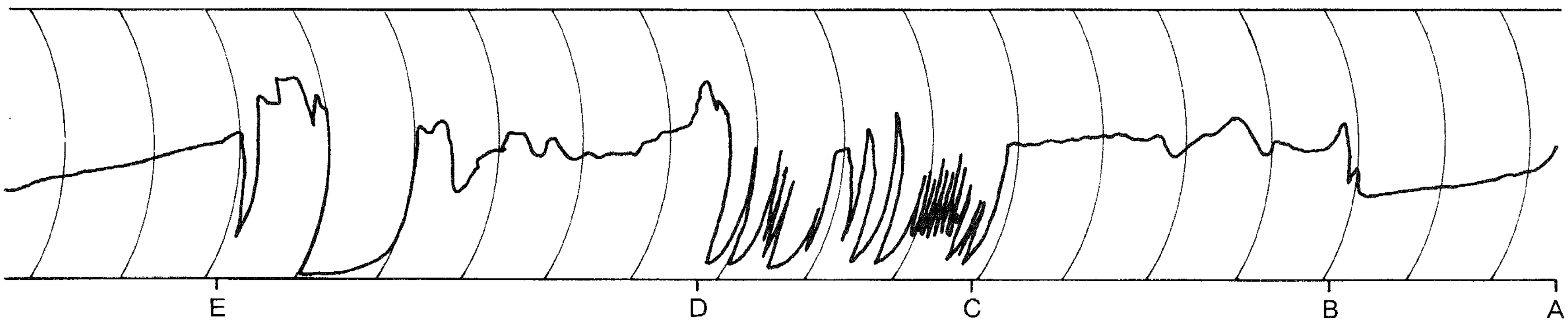
Lee Engineering Data

Lee Engineering Division has supplied oscillograms taken at the Thomas Portal of the Mathies Mine of Consolidation Coal. Figure 2 shows a sample.

Table II represents the reduction of the data from the oscillograms, showing an energy usage of a little over 2 kW hrs for a mission.

A to B Shuttle car is unloading. There is an initial surge followed by a gradual tapering off of the curve as the amount of coal remaining on the belt decreases.

250 VDC
300 AMPS FULL SCALE
2.5 SECONDS PER DIVISION



B to C Shuttle car is tramming from the unload point to the miner. Hump may be due to rounding a corner.

C to D Shuttle car is being loaded at the mine face. The car jockeys about to get a full load.

D to E Shuttle car is tramming from the mine face to the unload point. The pause is due to the car waiting for another car to go by (change out point).

Figure 1. National Mine Service Shuttle Car Mission for Orient #4 Mine, Freeman Coal Company (data taken on 1-18 -77)

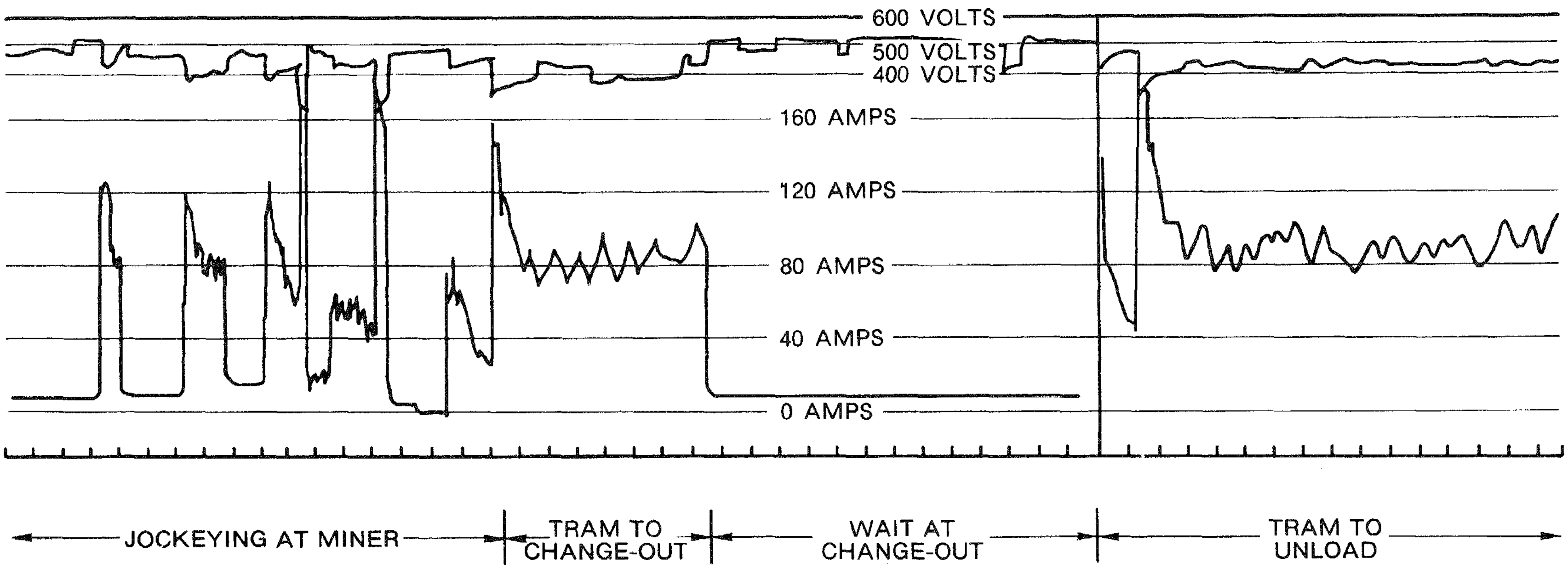


Figure 2. Shuttle Car Mission Current and Voltage Profile Consolidation Coal, Mathies Mine, Thomas Portal. 40 hp Motor, 600 Vdc, 240 A.

Table II

Consolidation's Mathies Mine - Thomas Portal
(data taken on 1/20/77)

	TIME (sec)	ENERGY (kW hrs)	MISSION ENERGY (kW hrs)
Shuttle Car #1 (small)			
Test #1			
Load	77	0.18	--
Tram to train	92	1.13	--
Unload	91	0.44	
Test #2			
Tram to train	82	0.94	--
Unload	107	0.51	--
Tram to miner	48	0.32	--
Load	106	0.32	2.09
Tram to train	91	1.03	2.18

NOTE: The shuttle car had to tram 400 - 500 ft up an S curve with a slight grade to the unload station. The operators were getting ready to move the unload point closer to the face.

The shuttle car uses 200 watts for lights and 4,800 watts for the hydraulic pump (auxiliary load).

Table III represents data provided by Lee Engineering from its analysis of required missions for flywheel vehicles. These data also show approximately 2 kW hrs requirement for the vehicle.

Table III

Lee Engineering Division, Consolidation Coal - Analysis of Mission Energy Requirements for Flywheel-Powered Shuttle Car (for 57,500 lb GVW Car on Level Bottom)

	Energy Required
Acceleration to 5 mph (empty)	35 watt/hrs
Tram to Changeout (250 ft)	276
Deceleration to Stop	7
Stand at Changeout (60 sec)	158
Acceleration to 5 mph	35
Tram to face (250 ft)	276
Deceleration to Stop	7
Load to Face (60 sec)	255
Acceleration to 4.5 mph (loaded)	52
Tram to Dump Point (500 ft)	861
Deceleration to Stop	12
Connect Charging Coupler (10 sec)	26
Unload (60 sec)	-0-
Total Flywheel Energy Required	2000 watt/hrs

Round-Trip Time = 343 seconds

Summary of Mine Data

Energy, expressed in kilowatt hours, is the product of power, in kilowatts times the period of time, in hours, that the car is using power at that rate. Table IV shows the average power utilization for the various segments of the shuttle car mission in the Orient #4 and Mathies Mines. These values will be used to convert the PSU/USBM simulator results, which are in units of time, into units of energy (kW hrs) needed for flywheel energy storage requirements.

Table IV

Energy Storage Requirements
Shuttle Car Power Levels*

	AVERAGE	HIGHEST
Unload	25 kW	31 kW
Tram to Miner	31 kW	38 kW
Load	12 kW	17 kW
Tram to Unload	39 kW	44 kW
Idle - Shut Down	30 W	30 W
Idle - Lights	275 W	350 W
Change-Out (Lights and Pump)	4.7 kW	4.8 kW

*Data obtained by averaging previous results of Orient #4 and Mathies Mines.

2.3 PSU/USBM SIMULATOR

The Pennsylvania State University/United States Bureau of Mines (PSU/USBM) Underground Generalized Materials Handling Simulator is a computer program that models a mine's underground activities. It was used to evaluate the effects of seam height, floor quality and grades, and haulage distances on such factors as loading, discharging, acceleration and deceleration, adequacy of tramming horsepower, auxiliary power requirements, and waiting times. These factors were then used to evaluate the shuttle car's performance and to determine its energy requirements.

Six-Entry Cut Plan

The six-entry cut plan shown in Figure 3 was used to obtain time and performance data as a function of the size of the working area. In the plan, the maximum and minimum haulroad distances for the 70-foot entry and 90-foot crosscut centers are established at 540 (Cut 84) and 170 (Cut 12) feet, respectively. Two 250 Vdc shuttle cars will be used behind a milling-type continuous miner cutting two side by side cuts of 10 and 8 feet,

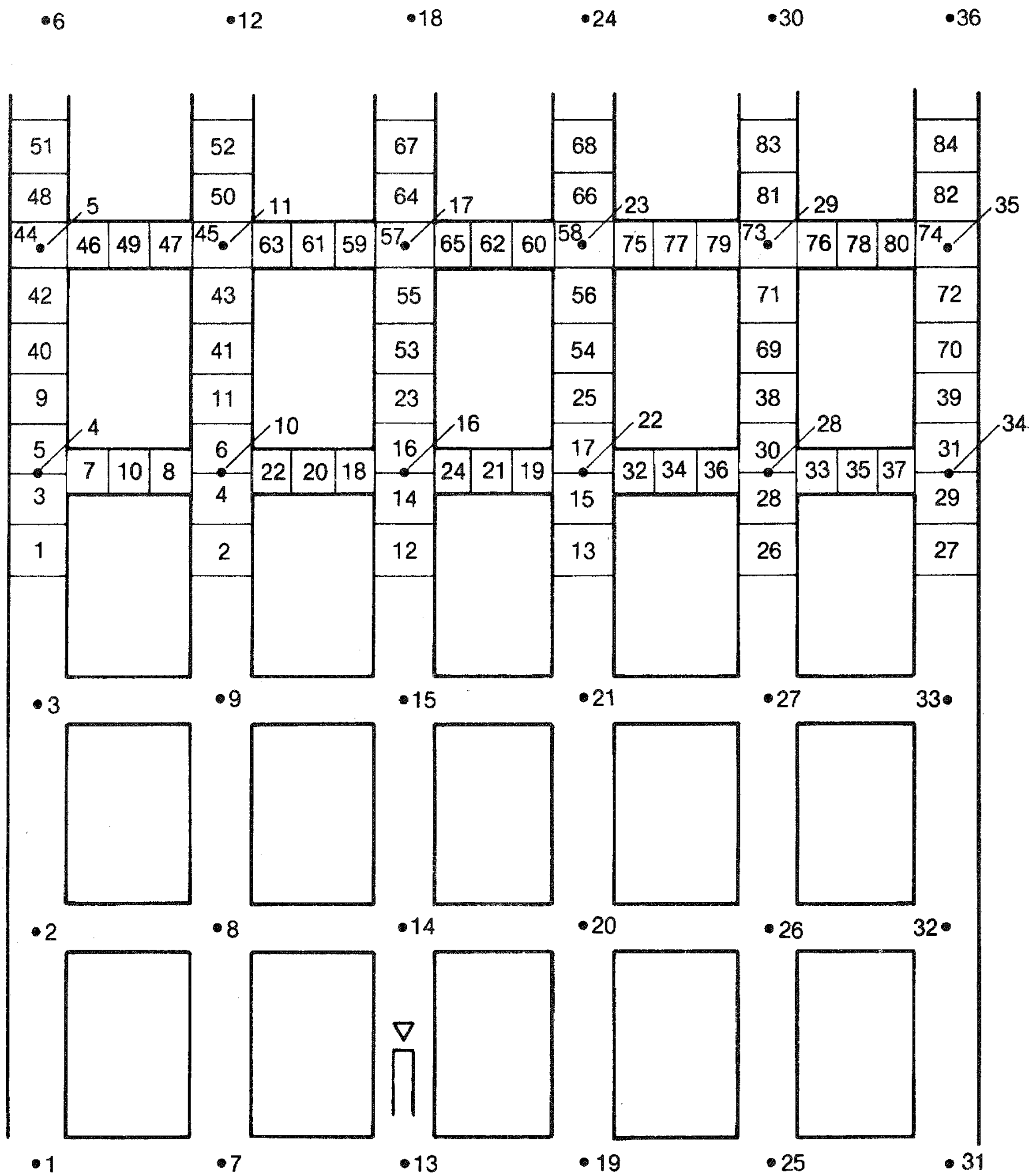


Figure 3. Six-Entry Cut Plan

respectively. Because the simulations will be applied to industry-wide conditions, the general operating data and miner performance characteristics listed in Tables V and VI are used as representative of industry averages.

Table V

General Operating Data
(re: time in minutes)

Total shift time available			480 minutes
Travel In	30		
Prepare To Mine	30		
Service Equipment	20		
Lunch	30		
Miscellaneous	10		
Prepare To Leave	20		
Travel Out (and Early Out)	30		
Subtotal - Fixed Times		170	
Mechanical Delay (Miner) Miner Breakdowns	35		
Mechanical Delay Support Equipment Breakdowns	25		
Subtotal - Mechanical Delays		60	
Available Face Time	250		
Subtotal - Mining		250	
Total Time			480 minutes

Table VI

Equipment Performance Characteristics
and Related Jobtimes

Mining Rate (sump, shear, and tram)	7.0 TPM
Loading Rate	12.5 TPM
Tram Rate, Miner	20.0 FPM(15 FPMrev)
Cleanup Time, Per Cut	5.0 min
Lift Change Time, Per Lift	4.0 min
Gas Test, Every 20 Minutes	1.5 min
Set Up to Mine, Per Cut	6.0 min
Shuttle Car Speed, Maximum	300.0 FPM
Shuttle Car Speed, Corner	250.0 FPM

The shuttle cars travel between the miner at a given cut and the discharge station located between coordinates 13 and 14. After the miner finishes a cut, it moves to the next higher numbered cut. The cuts and coordinates shown in Figure 1 along with change out points were entered into the PSU simulator by means of parameter cards. The simulator starts the miner at Cut 1 and ends when Cut 84 is finished. The printed output of the simulator is is broken down by shifts. Each shift contains many details including: the number of trips made and the total time for each activity (tramming, waiting at change points, waiting for miner, loading, and discharging).

Seam Height

To study the relationship of shuttle car capacity to operational service, four, six and eight-foot seam heights were chosen. A lower limit of 4 feet was selected, since below this value the type of mining equipment will radically change. Similarly, at heights much above eight feet, new equipment must be introduced and/or multiple pass mining must be practiced. Tables VII, VIII, and IX list the performance characteristics for the selected shuttle cars.

Floor Conditions

Bottom conditions and grades are related in that a bottom which may be satisfactory in a flat-lying seam may be intolerable with moderate grades. Three grades were evaluated: less than 3% (flat), 5% or moderate, and 10% or steep. The resistance of the bottom condition is expressed as pounds of force per ton of vehicle weight, and the coefficient of traction is expressed as percent of vehicle weight which can be applied to drive the wheels before wheel spin occurs (assuming the motor and drive train are capable of producing this force). Three coefficients of traction and rolling resistances were considered, as follows:

Category	Coefficient of Traction	Rolling Resistance	Grade
Good Bottom	0.55	65	3%
Fair Bottom	0.44	300	5%
Poor Bottom	0.40	300	10%

A coefficient of traction of 0.55 corresponds to dry clay while that of 0.40 is for wet clay. The rolling resistance of 65 lbs/ton is the figure used for firm dirt roads with some tire penetration. At 300 lbs/ton the road is assumed to be soft or wet dirt with rutting.

Table VII

Selected Shuttle Car (4422021) for the Four Foot Seam Application

CODE NUMBER	MANUFACTURER	MODEL	CHARACTERISTICS		
4422021	Joy	16SC5 DC	Length (ft)	=	25.00
			Load (lbs)	=	7,900.00
			Deviation (lbs)	=	0.00
			Discharge Time (min)	=	0.50
			Braking Acceleration (ft/min^2)	=	10,000.00
			Max. Acceleration (ft/min^2)	=	20,000.00
			Max. Speed (ft/min)	=	380.00
			Max. Amperes	=	200.00
			Drive Efficiency Ratio	=	0.70
			Gear Reduction Ratio	=	43.50
			Wheel Radius (ft)	=	1.39
			Empty Weight (lbs)	=	21,900.00
			Rolling Resistance (lbs/tons)	=	100.00
			Corner Speed (ft/min)	=	250.00
			Multiplier for Reversing	=	1.00
			Two Traction Motors (hp)	=	15.00
			Bottom Condition Factor	=	1.00
			Ratio Feeder Capacity (lbs)	=	0.00
			Discharge Rate Ratio Feeder (lbs/min)	=	0.00
4422022	Joy	16SC6 DC	Same as above except: Load (lbs) = 9,950		

Table VIII

Selected Shuttle Car (4423082) for the Six Foot Application

CODE NUMBER	MANUFACTURER	MODEL	CHARACTERISTICS		
4423081	Joy	18SC13 DC	Length (ft)	=	28.00
			Load (lbs)	=	9,500.00
			Deviation (lbs)	=	0.00
			Discharge Time (min)	=	0.33
			Braking Acceleration (ft/min^2)	=	10,000.00
			Max. Acceleration (ft/min^2)	=	20,000.00
			Max. Speed (ft/min)	=	420.00
			Max. Amperes	=	200.00
			Drive Efficiency Ratio	=	0.70
			Gear Reduction Ratio	=	33.30
			Wheel Radius (ft)	=	1.35
			Empty Weight (lbs)	=	26,700.00
			Rolling Resistance (lbs/tons)	=	100.00
			Corner Speed (ft/min)	=	250.00
			Multiplier for Reversing	=	1.00
			Two Traction Motors (hp)	=	15.00
			Bottom Condition Factor	=	1.00
			Ratio Feeder Capacity (lbs)	=	0.00
			Discharge Rate Ratio Feeder (lbs/min)	=	0.00
4423082	Joy	18SC13 DC	Same as above except: Load (lbs) = 11,800		

$$\text{Discharge Time (min)} = \left(\frac{11,800}{9,500}\right) 0.33$$

Table IX

Selected Shuttle Car (4423042) for the Eight Foot Seam Application

CODE NUMBER	MANUFACTURER	MODEL	CHARACTERISTICS		
4423041	Joy	18SC8 DC	Length (ft)	=	30.00
			Load (lbs)	=	12,750.00
			Deviation (lbs)	=	0.00
			Discharge Time (min)	=	0.66
			Braking Acceleration (ft/min^2)	=	10,000.00
			Max. Acceleration (ft/min^2)	=	20,000.00
			Max. Speed (ft/min)	=	385.00
			Max. Amperes	=	300.00
			Drive Efficiency Ratio	=	0.70
			Gear Reduction Ratio	=	43.30
			Wheel Radius (ft)	=	1.75
			Empty Weight (lbs)	=	37,500.00
			Rolling Resistance (lbs/tons)	=	100.00
			Corner Speed (ft/min)	=	250.00
			Multiplier for Reversing	=	1.00
			Two Traction Motors (hp)	=	35.00
			Bottom Condition Factor	=	1.00
			Ratio Feeder Capacity (lbs)	=	0.00
			Discharge Rate Ratio Feeder (lbs/min)	=	0.00
4423042	Joy	18SC8 DC	Same as above except:		
			Load (lbs) = 15,500		

$$\text{Discharge time (min)} = \frac{15,500}{12,750}\ 0.66$$

2.4 FLYWHEEL SIZING

Five runs of the PSU/USBM Underground Materials Handling Simulator were made. The simulator yielded ampere hours for each shift of the simulation of the six entry cut plan shown in Fig-Figure 3. Conditions were simulated as follows:

Run	Seam Height	Grade %	Rolling Resistance lbs/ton	Coefficient of Traction
1	4	3	65	.55
2	4	5	300	.44
3	6	5	300	.44
4	8	5	300	.44
5	8	10	300	.40

Runs 2, 3, and 4 were subjectively judged by C.B. Manula to encompass 90% of the underground coal mine applications.

Tables X through XIV together with Figure 4 show the build-up to Total Flywheel Energy per trip from the tramming ampere hours derived from the simulation. The tables derive total Flywheel Energy per trip for each shift, but the worst case in each column is used in determining the flywheel size. The data on energy used by the shuttle car at the change point and during loading and unloading are shown in Table IV. These data were obtained at the Mathies and Orient Mines.

Figure 5 displays the data from the tables plotted as a function of seam height to graphically illustrate the effect of seam height on energy requirements.

Figure 5 is a plot of required kW hrs as a function of seam height and clearly indicates a maximum total usable flywheel energy of 7.5 kW hrs required for an 8 ft seam and the bottom conditions indicated. This number includes 0.5 kW hrs of energy required for shuttle car discharge. It is planned that this energy would be supplied from the wayside since the flywheel would be spinning-up while the shuttle car is discharging. Therefore, 7 kW hrs is a reasonable maximum size to work with.

Table X

Energy Usage for Mine
with Best Conditions for Minimum Energy

Seam Height - 4 ft
Grade - 3%
Rolling Resistance - 65#/ton
Coefficient of Traction - .55

A	B	C	D	E	F	G	H	I
SHIFT	TRAMMING	WAITING AT CHANGE POINTS	WAITING FOR MINER	LOADING	DISCHARGE	TOTAL CAR ENERGY	FLYWHEEL LOSSES	TOTAL FLYWHEEL ENERGY
1	1.12	.31	.36	.14	.26	2.19	.07	2.50
2	1.25	.38	.28	.15	.26	2.32	.08	2.66
3	0.85	.31	.26	.14	.26	1.82	.07	2.09
4	0.98	.33	.31	.14	.26	2.02	.08	2.32
5	1.17	.28	.27	.14	.26	2.12	.07	2.42
6	1.37	.35	.30	.14	.26	2.42	.08	2.77
7	1.32	.37	.34	.13	.26	2.42	.08	2.77
8	1.47	.30	.36	.14	.26	2.53	.08	2.89
9	1.48	.39	.41	.14	.26	2.68	.08	3.05
10	1.42	.33	.31	.15	.26	2.47	.08	2.82
11	1.18	.26	.31	.14	.26	2.15	.07	2.46
12	1.28	.28	.33	.14	.26	2.29	.08	2.62
13	1.38	.23	.26	.14	.26	2.27	.07	2.59
14	1.68	.22	.30	.15	.26	2.61	.08	2.98
15	1.57	.36	.26	.14	.26	2.59	.08	2.95
Worst possible combination:								
	1.68	.39	.41	.15	.26	2.89	.08	3.29

Table XI

Energy Usage for Typical 4' Seam Mine

Grade - 5%
Rolling Resistance - 300#/Ton
Coefficient of Traction - .44

A	B	C	D	E	F	G	H	I
SHIFT	TRAMMING	WAITING AT CHANGE POINTS	WAITING FOR MINER	LOADING	DISCHARGE	TOTAL CAR ENERGY	FLYWHEEL LOSSES	TOTAL FLYWHEEL ENERGY
1	1.33	.29	.37	.14	.26	2.39	.07	2.72
2	1.43	.26	.30	.14	.26	2.39	.08	2.73
3	0.98	.31	.28	.14	.26	1.97	.07	2.26
4	1.22	.37	.27	.14	.26	2.26	.08	2.59
5	1.30	.39	.28	.14	.26	2.37	.08	2.71
6	1.83	.31	.28	.14	.26	2.82	.08	3.21
7	1.55	.29	.26	.14	.26	2.50	.08	2.86
8	1.83	.29	.28	.14	.26	2.80	.09	3.20
9	1.73	.33	.34	.14	.26	2.80	.08	3.19
10	1.83	.28	.30	.14	.26	2.81	.09	3.21
11	1.63	.24	.27	.15	.26	2.55	.08	2.91
12	1.52	.31	.29	.15	.26	2.53	.08	2.89
13	1.58	.29	.28	.14	.26	2.55	.09	2.92
14	1.85	.24	.29	.14	.26	2.78	.08	3.17
15	2.12	.27	.38	.15	.26	3.18	.09	3.62
16	2.02	.45	.36	.14	.26	3.23	.09	3.67
17	2.35	.18	.29	.15	.26	3.23	.08	3.66
Worst Possible Combination:								
	2.35	.45	.38	.15	.26	3.59	.09	4.07

Table XII

Energy Usage for Typical 6 Foot Seam Mine

Grade - 5%
Rolling Resistance - 300#/Ton
Coefficient of Traction - .44

A SHIFT	B TRAMMING	C WAITING AT CHANGE POINTS	D WAITING FOR MINER	E LOADING	F DISCHARGE	G TOTAL CAR ENERGY	H FLYWHEEL LOSSES	I TOTAL FLYWHEEL ENERGY
1	1.63	.24	.25	.20	.22	2.54	.07	2.89
2	1.78	.32	.28	.20	.22	2.80	.08	3.19
3	1.23	.31	.30	.20	.22	2.26	.07	2.58
4	1.52	.32	.27	.20	.22	2.53	.08	2.89
5	1.67	.29	.30	.19	.22	2.67	.08	3.04
6	2.52	.29	.28	.20	.22	3.51	.08	3.97
7	2.30	.30	.26	.20	.22	3.28	.08	3.72
8	2.47	.39	.35	.20	.22	3.63	.09	4.12
9	2.00	.24	.27	.20	.22	2.93	.07	3.32
10	2.25	.35	.32	.19	.22	3.33	.08	3.77
11	2.03	.29	.29	.20	.22	3.03	.08	3.44
12	1.90	.36	.32	.20	.22	3.00	.08	3.41
13	2.07	.26	.29	.20	.22	3.04	.08	3.45
14	2.30	.31	.30	.20	.22	3.33	.08	3.77
15	2.98	.27	.28	.20	.22	3.95	.09	4.47
16	2.72	.32	.26	.20	.22	3.72	.08	4.21
17	2.98	.23	.26	.20	.22	3.89	.09	4.41
Worst Possible Combinations:								
	2.98	.39	.35	.20	.22	4.14	.09	4.68

Table XIII

Energy Usage for Typical 8 Foot Seam Mine

Grade - 5%
Rolling Resistance - 300#/Ton
Coefficient of Traction - .44

A SHIFT	B TRAMMING	C WAITING FOR CHANGE POINTS	D WAITING FOR MINER	E LOADING	F DISCHARGE	G TOTAL CAR ENERGY	H FLYWHEEL LOSSES	I TOTAL FLYWHEEL ENERGY
1	3.33	.20	.25	.27	.42	4.47	.07	5.03
2	3.65	.31	.26	.27	.42	4.91	.08	5.53
3	2.57	.27	.28	.26	.42	3.80	.07	4.29
4	2.75	.37	.36	.26	.42	4.16	.08	4.69
5	2.93	.37	.34	.27	.42	4.33	.08	4.88
6	3.58	.38	.38	.26	.42	5.02	.07	5.64
7	3.97	.39	.35	.27	.42	5.40	.08	6.07
8	3.72	.36	.27	.26	.42	5.03	.08	5.66
9	4.17	.27	.31	.27	.42	5.44	.08	6.11
10	4.40	.30	.35	.27	.42	5.74	.08	6.45
11	4.52	.24	.26	.27	.42	5.71	.08	6.41
12	3.80	.27	.29	.27	.42	5.05	.08	5.68
13	3.62	.26	.27	.27	.42	4.84	.08	5.45
14	3.87	.27	.29	.27	.42	5.12	.08	5.76
15	4.25	.32	.28	.27	.42	5.54	.08	6.22
16	4.82	.23	.27	.27	.42	6.01	.07	6.73
17	4.75	.23	.26	.26	.42	5.92	.08	6.64
18	5.27	.37	.35	.27	.42	6.68	.09	7.50
Worst Possible Combination:								
	5.27	.39	.38	.27	.42	6.73	.09	7.55

Table XIV

Energy Usage for Mine with Worst Conditions for Maximum Energy

Grade - 10%
Rolling Resistance - 300#/Ton
Coefficient of Traction - .4

A	B	C	D	E	F	G	H	I
SHIFT	TRAMMING	WAITING AT CHANGE POINTS	WAITING FOR MINER	LOADING	DISCHARGE	TOTAL CAR ENERGY	FLYWHEEL LOSSES	TOTAL FLYWHEEL ENERGY
1	3.37	.21	.25	.27	.42	4.52	.07	5.08
2	3.65	.31	.26	.27	.42	4.91	.08	5.53
3	2.64	.27	.28	.26	.42	3.87	.07	4.36
4	2.87	.37	.36	.27	.42	4.29	.08	4.84
5	3.00	.37	.34	.27	.42	4.40	.08	4.95
6	4.08	.42	.37	.26	.42	5.55	.07	6.22
7	4.35	.26	.29	.27	.42	5.59	.08	6.28
8	4.29	.28	.30	.27	.42	5.56	.09	6.26
9	4.44	.22	.26	.27	.42	5.61	.08	6.30
10	4.43	.26	.31	.27	.42	5.69	.07	6.38
11	4.46	.34	.35	.26	.42	5.83	.08	6.54
12	4.01	.29	.29	.27	.42	5.28	.07	5.93
13	3.74	.27	.29	.26	.42	4.98	.08	5.60
14	4.01	.32	.33	.26	.42	5.34	.08	6.00
15	4.30	.32	.31	.27	.42	5.62	.08	6.31
16	5.38	.22	.27	.27	.42	8.56	.08	7.35
17	5.19	.34	.26	.26	.42	6.47	.08	7.25
18	5.60	.39	.34	.27	.42	7.02	.09	7.87
Worst Possible Combination:								
	5.60	.42	.37	.27	.42	7.08	.09	7.94

COL	ENERGY		CALCULATION
B	Tramming Energy/Trip	=	$\frac{\text{AMP-hrs (Simulator)} \times 2 \text{ Motors} \times 250 \text{ V}}{1000 \times \text{no. of Trips}}$
C	Waiting at Charge Point Energy	=	$\frac{\text{Change Time} \times 4.8 \text{ kW*}}{60 \times \text{no. of Trips}}$
D	Waiting for Miner Energy	=	$\frac{\text{Wait for Miner Time} \times 4.8 \text{ kW*}}{60 \times \text{no. of Trips}}$
E	Loading Energy	=	$\frac{\text{Sump and Shear Time} \times 17 \text{ kW*}}{60 \times \text{no. of Trips}}$
F	Discharge Energy	=	$\frac{\text{Discharge Time} \times 31 \text{ kW*}}{60 \times \text{no. of Trips}}$
G	Total Car Energy	=	B + C + D + E + F
H	Flywheel Losses	=	$\frac{\text{Shift time - Portal in and out} \times .007 \text{ kW hr}}{\text{no. of Trips}}$
I	Total Flywheel Energy	=	$\frac{\text{Total Car Energy}}{\text{Efficiency of F.W. Motor and Power Conditioning}} + \text{Flywheel Losses}$

* Results from Orient #4 and Mathies Mines
**Estimate - Must be confirmed with flywheel analysis

Figure 4. Energy Storage Requirements
Build Up of Flywheel kW hrs

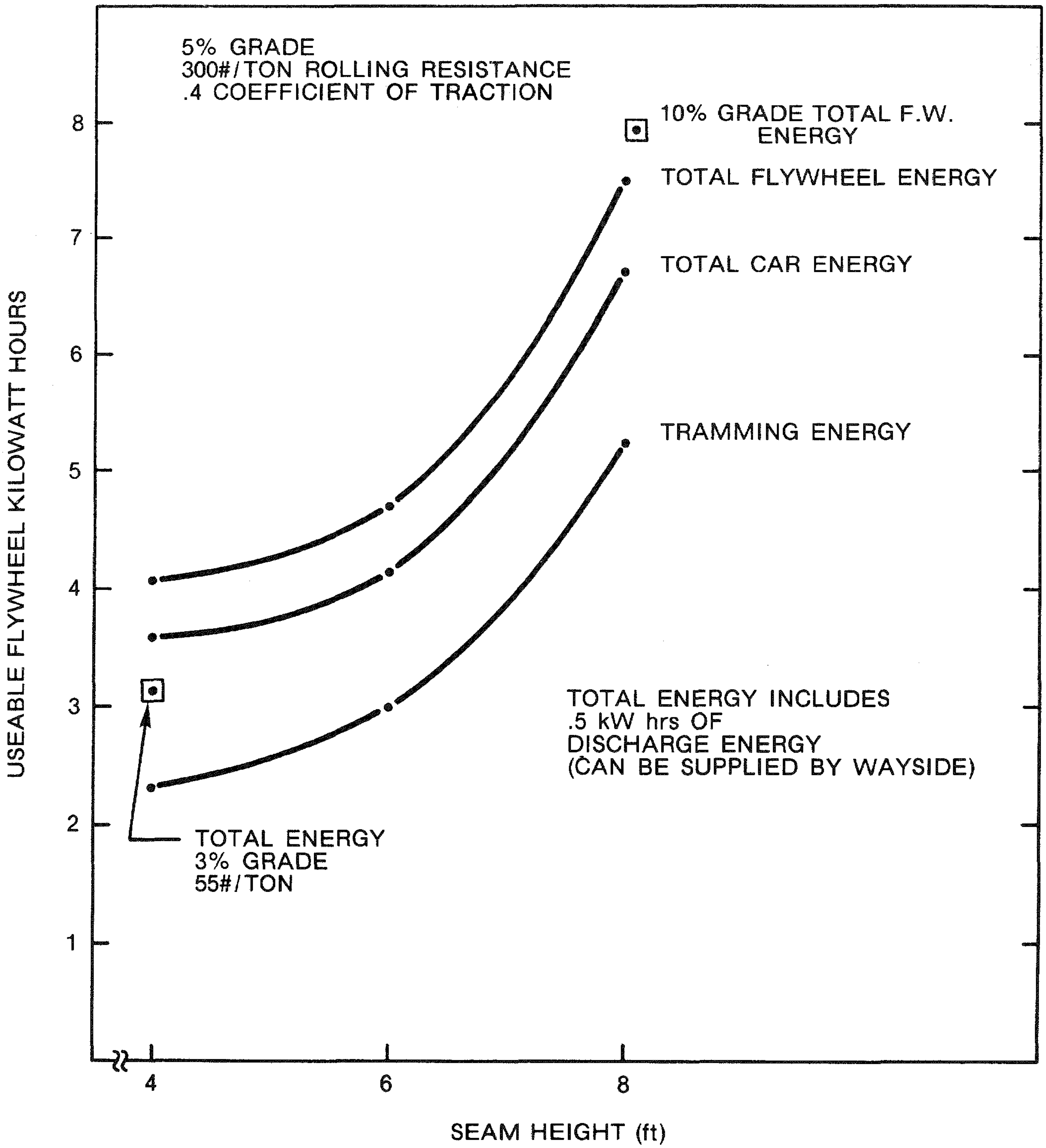


Figure 5. Required Flywheel kW hrs Versus Steam Height

Some other factors should be considered:

Waiting Time

The worst case wait times at the change out points from the simulator were used to calculate energy used by lights and pumps. Extended wait times due to breakdowns were not considered. However, if the control were designed to shut off the pumps and lights after an agreed on time at zero speed (when not discharging), it is possible that the energy requirements can be reduced. Table XI, for example, shows a loss attributed to wait time at the change point plus waiting for miner of .83 kW hrs. It is felt that this can be reduced.

Flywheel Losses

An estimate of 420 watts of spinning losses is based on an early estimate (before in-depth aerodynamic design is done). If this figure can be confirmed, a two-hour delay will lose less than 1 kW hr (if all loads are disconnected), and normal breakdowns and between shift layovers would not pose problems. However, a substantial change in these losses will increase the size of the flywheel and cause reconsideration of the operation between shifts.

Efficiency of Flywheel Driving Motor and On-Board Power Conditioning

A combined efficiency of about 90% was assumed prior to preliminary design of either the motor or the power conditioning. The actual values will have an effect on flywheel size.

Section 3

FLYWHEEL TECHNOLOGY

3.1 SUMMARY AND CONCLUSIONS

This section discusses the parameters important to flywheel design and develops a methodology for determining the appropriate flywheel size, shape, and material based on the application limitations. Composite flywheels are evaluated relative to their application to shuttle cars and their state of development for this application.

- It was concluded that the state-of-the-art of composites are not sufficiently advanced to design a flywheel with assurance of suitability for early demonstration of a flywheel shuttle car, although theoretical work indiates future promise of a weight saving for the tractor-trailer shuttle car application where diameter is not limiting.

- The work on steel flywheels indicated that an optimized conical shape made best use of material properties and available space at a diameter of the flywheel of approximately 43" with a speed of 10,000 rpm. The optimum utilization of the material is achieved by manufacturing the wheel without holes. This can be achieved by electron beam or friction welding the flywheel to the shaft, thus eliminating an expensive forging.

- For application to a conventional shuttle car, with restricted diameter, a cylindrical steel flywheel appears to be the optimum design.

3.2 CONSIDERATIONS IN FLYWHEEL DESIGN

Much has been published concerning the strength to density ratio that indicates a potential of storing 10 times as much energy in a pound of fiber composite as can be stored in steel. By the time fatigue life, shape factor, and manufacturability are considered it appears that the better steel wheels can store 12 watthours per pound, and in the near future the composites may achieve 30 - 40 W-hrs/lb (Reference 1). To allow preliminary exploration of flywheel configurations, a matrix was developed to display the parameters. From the matrix, one can more readily achieve a perspective of the optimum flywheel configuration and material, considering the application constraints. The matrix description discussed below is based on the above energy densities.

3.2.1 Matrix Description

Seldom will the governing criteria in a flywheel design be the energy storage per unit weight (specific energy). Usually, the space will be limiting, and so two questions must be answered: (1) For a given diameter, what are the material and shape options? (2) For a given energy storage and a fixed length, what are the material and shape options?

To deal with these questions, the basic flywheel parameters must be considered:

- Construction
- Material (Stress & Density)
- Energy
- Energy per lb
- Diameter
- Length
- Speed
- Weight

In order to reduce the number of variables to a manageable number, construction and material were considered independent variables, and the rest were reduced to five dependent variables:

- Energy per unit length
- Energy per unit weight
- Weight per unit length
- Diameter
- Speed

From these a table of parameter ratios, Figure 6, was developed that allows exploration of the shape and material options for a number of critical variables. The parameter ratios of Figure 6 are developed starting with the basic flywheel equations of Figure 7. The basic equation terms are defined in Figure 8. The parameter ratios of Figure 9 are then derived.

The four basic flywheel equations as organized in Figure 7 show some very interesting relationships regarding the constants that depend upon the configuration of a flywheel. The equations show that there are only three basic constants, and these pertain to volume (C_v), inertia (C_i), and stress (C_t). These three constants are related by Equation 8 to produce the conventional shape factor (C_s). Thus if the commonly used chart for shape factors (such as Table 6-2, Reference 2) was supplemented with the three additional constants, one would be supplied with adequate information for parameteric comparisons of flywheels.

Experience with the flywheel constants has shown that the anisotropic flywheel designs require the use of another factor that identifies the efficiency with which the fibers are employed.

	Equal Weight Per Unit Length	Equal Diameters	Equal Speed	Equal Energy Per Unit Length
Weight Ratio	1	VB	TVA	$\frac{B}{SA}$
Diameter Ratio	$\left(\frac{1}{VB}\right)^{1/2}$	1	$\left(\frac{TA}{B}\right)^{1/2}$	$\left(\frac{1}{SVA}\right)^{1/2}$
Speed Ratio	$\left(TVA\right)^{1/2}$	$\left(\frac{TA}{B}\right)^{1/2}$	1	$\left(\frac{TSVA^2}{B}\right)^{1/2}$
Energy Ratio	$\frac{SA}{B}$	SVA	$\frac{TSVA^2}{B}$	1
Material Energy Ratio	$\frac{SA}{B}$	$\frac{SA}{B}$	$\frac{SA}{B}$	$\frac{SA}{B}$

$$A = \frac{\sigma_2}{\sigma_1} \qquad S = \frac{\text{Shape Factor}_2}{\text{Shape Factor}_1}$$

$$B = \frac{\rho_2}{\rho_1} \qquad V = \frac{\text{Volume Factor}_2}{\text{Volume Factor}_1}$$

$$T = \frac{\text{Stress Factor}_2}{\text{Stress Factor}_1}$$

Figure 6. Parameter Ratios

1. $R^2\omega^2 = C_t \ (g)\left(\frac{\sigma}{\rho}\right)$

$\therefore R^2N^2 = C_t \left(\frac{60}{2\pi}\right)^2 (386.4) \left(\frac{\sigma}{\rho}\right)$

2. $\therefore R^2N^2 = 35{,}235 \ C_t \left(\frac{\sigma}{\rho}\right)$

3. $V = C_v V_{cyl} = C_v \ (\pi R^2 h)$

4. $W = C_v \ (\pi R^2 h) \ \rho$

5. $I = C_i \ I_{cyl.} = C_i \left(\frac{1}{2g}\right) (\pi R^2 h) \ R^2 \ \rho$

$KE = \frac{1}{31{,}850} \left(\frac{I\omega^2}{2}\right)$

$\therefore KE = \frac{C_i}{31{,}850} \left(\frac{1}{4g}\right) (\pi R^2 h) \ R^2\omega^2 \ \rho$

6. $\therefore KE = \frac{C_i C_t}{4} \left(\frac{1}{31{,}850}\right) (\pi R^2 h) \ \sigma$

$SE = \frac{KE}{W} = \frac{C_i C_t}{4 \ C_v} \left(\frac{1}{31{,}850}\right) \left(\frac{\sigma}{\rho}\right)$

7. $SE = \frac{C_s}{31{,}850} \left(\frac{\sigma}{\rho}\right)$

8. $C_s = \frac{C_i C_t}{4 \ C_v}$

9. $\therefore KE = \frac{C_s C_v}{31{,}850} (\pi R^2 h) \ \sigma$

○ = Key Equations

Figure 7. Basic Flywheel Equations

W = Weight, lbs

C_v = Swept Volume Factor (CYL = 1)

R = Max. Radius, in

h = Flywheel Length, in

ρ = Material Density, lb/in^3 (.29 for Steel, .05 for Fiberglass)

SE = Average Energy Density, Whrs /lb

C_s = Shape Factor (Constant Stress Wheel = 1)

σ = Maximum Stress, lb/in^2

N = Speed, rpm

C_t = Stress Factor

KE = Total Stored Energy, Whr

31,850 = in-lb/Whr

C_i = Swept Volume Inertia Factor (Cyl = 1)

V = Volume, in^3

ω = Speed, Radians Per Second

I = Inertia

Figure 8. Definitions for Basic Equations

10. $\frac{\sigma_2}{\sigma_1} = A$ = Stress Ratio

11. $\frac{\rho_2}{\rho_1} = B$ = Density Ratio

12. $\frac{\text{Shape Factor}_2}{\text{Shape Factor}_1} = S$

13. $\frac{\text{Volume Factor}_2}{\text{Volume Factor}_1} = V$

14. $\frac{\text{Stress Factor}_2}{\text{Stress Factor}_1} = T$

15. $\frac{(W/h)_2}{(W/h)_1} = \left(\frac{C_{v2}}{C_{v1}}\right)\left(\frac{R_2}{R_1}\right)^2\left(\frac{\rho_2}{\rho_1}\right) = VB\left(\frac{R_2}{R_1}\right)^2$ = Weight Ratio

16. $\left(\frac{SE_2}{SE_1}\right) = \left(\frac{C_{s2}}{C_{s1}}\right)\left(\frac{\sigma_2}{\sigma_1}\right)\left(\frac{\rho_1}{\rho_2}\right) = \frac{SA}{B}$ = Material Energy Ratio

17. $\frac{(KE/h)_2}{(KE/h)_1} = \left(\frac{C_{s2}}{C_{s1}}\right)\left(\frac{C_{v2}}{C_{v1}}\right)\left(\frac{R_2}{R_1}\right)^2\left(\frac{\sigma_2}{\sigma_1}\right) = SVA\left(\frac{R_2}{R_1}\right)^2$ = Energy Ratio

$$\left(\frac{R_2N_2}{R_1N_1}\right)^2 = \left(\frac{C_{t2}}{C_{t1}}\right)\left(\frac{\sigma_2}{\sigma_1}\right)\left(\frac{\rho_1}{\rho_2}\right) = \frac{TA}{B}$$

18. $\frac{R_2}{R_1} = \left(\frac{N_1}{N2}\right)\left(\frac{TA}{B}\right)^{1/2}$ = Diameter Ratio

19. $\frac{N_2}{N_1} = \left(\frac{R_1}{R_2}\right)\left(\frac{TA}{B}\right)^{1/2}$ = Speed Ratio

Figure 9. Parameter Ratios

This efficiency factor is then applied to the allowable fiber stress to arrive at the equivalent isotropic limiting working stress. For instance, the concentric ring flywheel of Figure 17 has an efficiency of 1.0 because the lay of the fibers is in the direction of the stress, so the full allowable working stress would be used in design calculations. For the Scotchply alpha lay plates the efficiency is estimated at 1/3 which severly depreciates its performance.

Five flywheel designs are characterized in Figure 10 by the above constants. Using these constants in the Equations 10 through 14 of Figure 9 provides the table of factors Figure 11 which lead to comparison ratios of the matrices of Figures 12, 13, 14, and 15. (For the ultimate in accuracy the stress constant C_t should be generated from a finite element analysis. Then with equally accurate C_v and C_i from simple geometry calculations the shape factor can be accurately determined by Equation 8 of Figure 7.) Figure 10 gives details on three additional configurations extending the background information.

For comparison of steel and composites, the stress ratio was derived from the quoted 12 W hrs/lb and 30 - 40 W hrs/lb from Reference 1. Reasonable shape factors for high performance flywheels will produce a working stress of 130,000 psi for steel and 190,000 psi for composites. These both seem optimistic but are used for calculating the matrix ratios since the primary purpose of the matrix is to develop a comparison. From these two stresses, the value for stress ratio A for the two composite wheels (b) and (c), Figure 10, related to the conical steel wheel (a) is as follows:

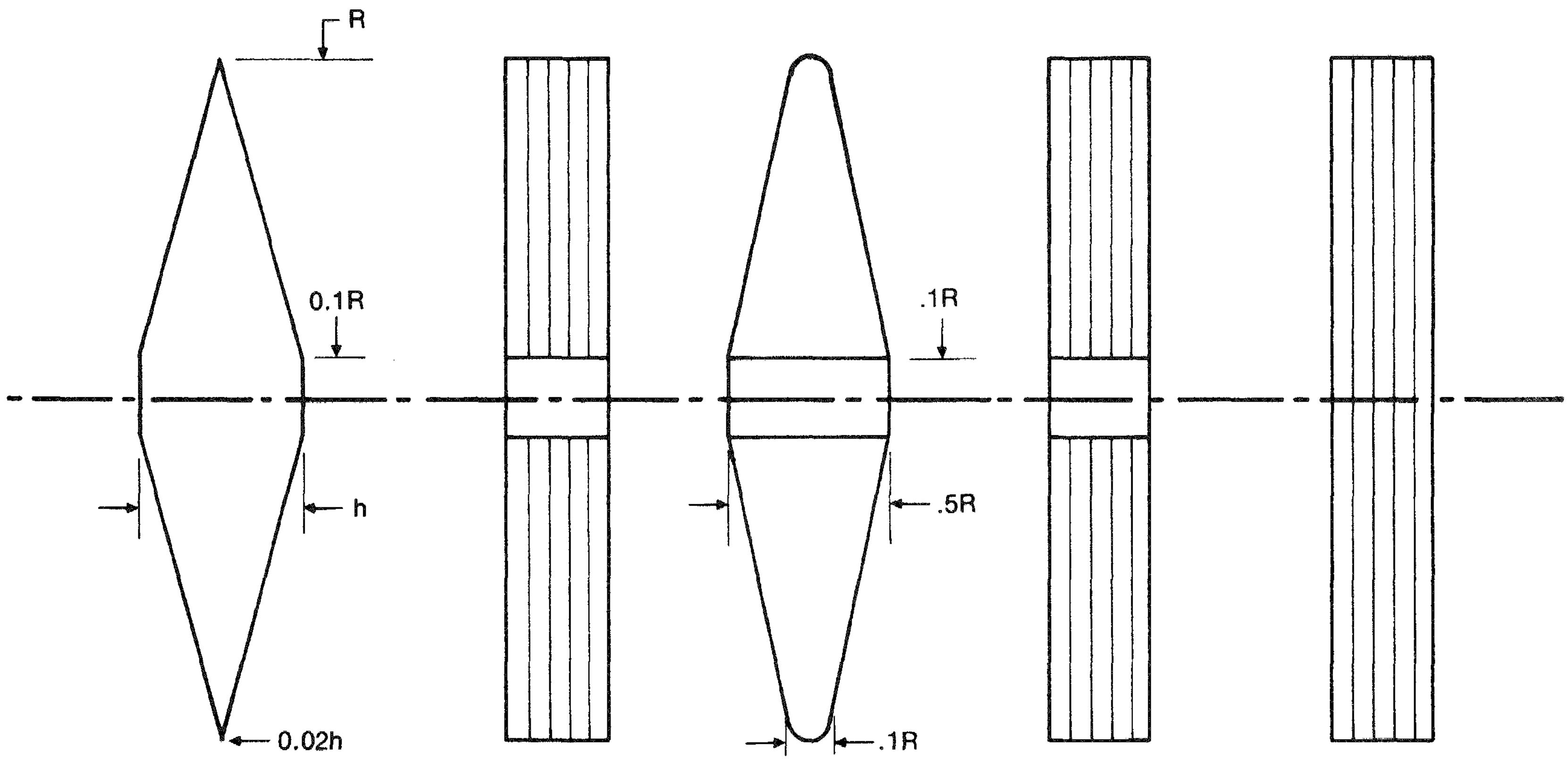
$$\frac{190{,}000}{130{,}000} \times \varepsilon = .49 \text{ and } 1.10 \quad (\varepsilon = \text{fiber use efficiency})$$

Figure 12 compares flywheels of equal speeds to a steel flywheel of modified conical shape with integral shafts, and can be used when the flywheel speed is determined by external factors such as bearings or a connected machine.

Figure 13 compares flywheels of equal weight to a steel flywheel of conical shape with integral shafts. This matrix can be used when weight is the governing criteria and diameter and speed can vary.

Figure 14 compares flywheels of equal diameter to a steel flywheel of modified conical shape with integral shafts and can be used when diametral space is limiting as in a conventional conveyor shuttle car.

Figure 15 compares flywheels of equal energy to a steel flywheel of modified conical shape with integral shafts and can be used where weight or diameter is flexible.



a. Conical Steel with Integral Shafts

b. Composite Disc with Shaft Hole (3M Scotch Ply)

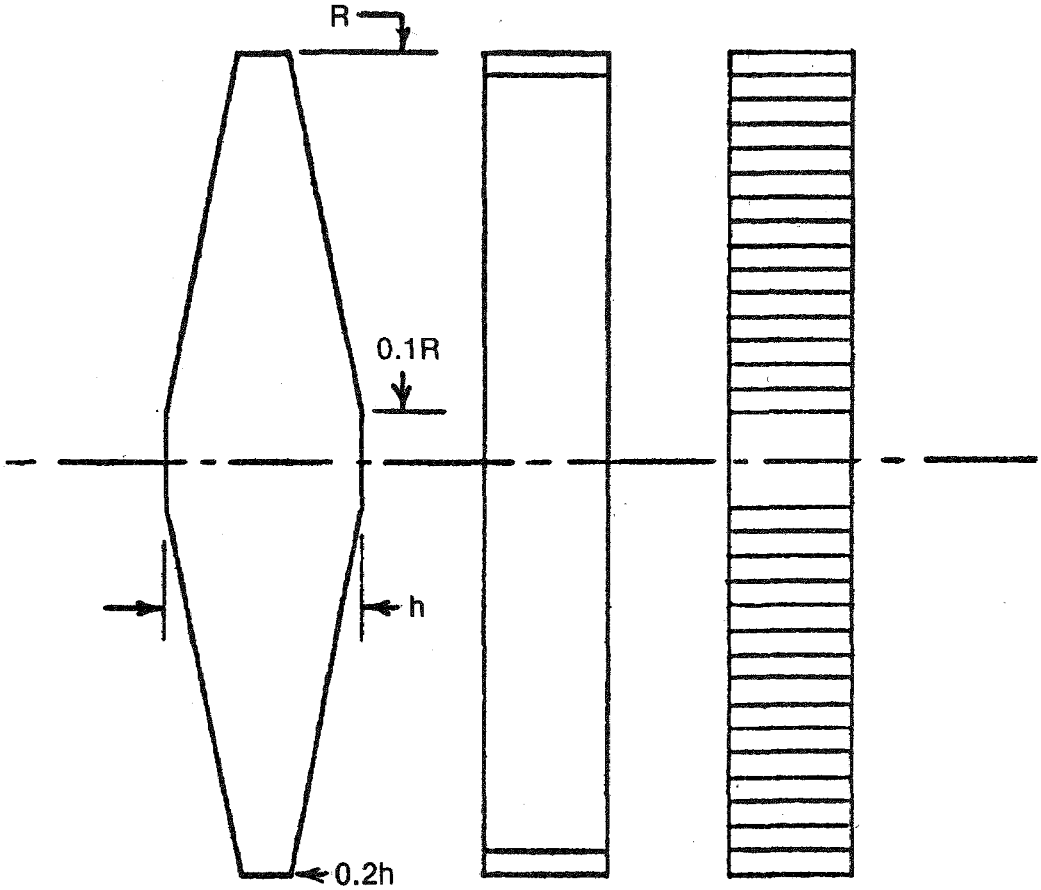
c. Composite Cone with Shaft Hole

d. Steel Disc with Shaft Hole

e. Steel Disc Without Shaft Hole

Literature Reference	Rockwell Ref. 4	Roark Ref. 5	Rockwell Ref. 4	Roark Ref. 5	Roark Ref. 5
Shape, C_s	0.83	0.31	0.38	0.31	0.61
Volume, C_v	0.38	1.0	0.50	1.0	1.0
Inertia, C_i	0.24	1.0	0.38	1.0	1.0
Stress, C_t	5.3	1.2	2.0	1.2	2.4
Fiber Use Eff.		0.33 est.	0.75 est.		

Figure 10. Flywheel Configuration Factors



	f. Conical Steel	g. Single Hoop	h. Composite Hoop
Literature Reference	Rockwell Ref. 4		
Shape, C_s	0.80	0.50	0.25
Volume C_v	0.50	$1-(\frac{r}{R})^2$	1.00
Inertia C_i	0.38	$1-(\frac{r}{R})^4$	1.00
Stress C_t	4.19	1.00	1.00
Fiber Eff.,	-	1.00	1.00

Figure 10. Flywheel Configuration Factors (Continued)

		A	B	S	V	T
		$\frac{\sigma_2}{\sigma_1}$	$\frac{\rho_2}{\rho_1}$	$\frac{C_{s2}}{C_{s1}}$	$\frac{C_{v2}}{C_{v1}}$	$\frac{C_{t2}}{C_{t1}}$
a.	Conical Steel with Integral Shafts*	-	-	-	-	-
b.	Composite Disc with Shaft Hole	0.49	0.17	0.37	2.60	0.23
c.	Composite Cone with Shaft Hole*	1.10	0.17	0.46	1.32	0.38
d.	Steel Disc with Shaft Hole	1.0	1.0	0.37	2.60	0.23
e.	Steel Disc Without Holes	1.0	1.0	0.73	2.60	0.45

*See Reference 4

Figure 11. Factors for Calculating Parameter Ratios

3.2.2 Use of Matrix for Comparison of Flywheel Types

The matrix figures offer a direct comparison of flywheel Types b, c, d, and e to the conical steel flywheel, Type a. However, a cross comparison between any two of Types b, c, d, or e can be made by directly comparing the corresponding matrix values of any of the tables. For instance, to compare flywheel Type b to flywheel Type d when the two have equal stored energy, use the values from Figure 15 as follows:

$$\text{Weight ratio} = \frac{0.93}{2.7} = 0.34$$

$$\text{Diameter ratio} = \frac{1.45}{1.02} = 1.4$$

$$\text{Speed ratio} = \frac{.57}{.47} = 1.2$$

$$\text{Material energy ratio} = \frac{1.07}{.37} = 2.9$$

3.2.3 Use of Matrix for Tractor-Trailer Car Flywheel Analysis

The pertinent factors for a preliminary look at a flywheel for a tractor-trailer car indicates that neither diameter nor weight is limiting, but the speed may be set by the motor/alternator or bearing limitations. For this case Figure 12 shows the following

A composite flywheel with 1.6 times the diameter of a modified conical steel flywheel can store 1.6 times the energy when the composite is stressed to 1.10 times the stress for steel. If this stress were achievable and if the state of development of composites were sufficiently advanced, this would be the direction to go. Unfortunately, these stress levels have not been achieved in practice. Section 3.3 discusses limitations of composites. Neither of the steel flywheels compares favorably with the conical steel wheel from an energy standpoint.

3.2.4 Use of Matrix for Conventional Conveyor Shuttle Car Flywheel Analysis

When considering a flywheel for a conventional shuttle car where both diameter and speed are limiting, the following reasoning applies.

Figure 14 shows that for equal diameters, neither composite wheel can achieve the energy of a conical wheel or a steel disc wheel without shaft hole, even with the conical composite wheel operating at 60% greater speed than the conical steel wheel. The steel disc flywheel without holes can achieve 1.9 times the energy of a conical wheel, but it is restricted to a lower speed but at a higher weight. When speed is held constant, Figure 12 shows the minimum diameter commensurate with optimum energy storage is a steel disc without holes.

Figure 15 shows another method of reasoning. If the design is for constant energy with restriction on diameter and speed, the minimum diameter consistent with reasonable utilization of material and near unity speed with some weight penalty indicates a steel disc without holes should be used.

3.3 COMPOSITE FLYWHEELS

A flywheel presents an unusual stress situation (Figure 16). At the outside diameter the stress is tensile since it is tangential to the outside and providing a hoop stress. Closer toward the center of rotation a radial tensile stress develops; at the center of a solid disc flywheel both of these stresses are equal and at a maximum. Note that these two stresses are 90° from each other, and this poses a special problem for the use of composites as flywheel materials. The problem comes from the fact that composites are strong only lengthwise along the fibers, and the allowable tensile stress 90° to the fibers is determined by the strength of the epoxy.

3.3.1 Specific Energy

Composites came into prominence with the development of high-strength glass fibers. The terms of flywheel equations can be organized to show that a governing factor in performance is stress/density or specific energy, Equation 7, Figure 7. On this basis there are high-strength fibers which have a stress/density ratio that is 10 times that of the best steel and consequently have the theoretical potential of storing 10 times the

	Equation	Composite Disc with Shaft Hole	Composite Cone with Shaft Hole	Steel Disc with Shaft Hole	Steel Disc Without Holes
Weight Ratio	TVA	0.29	0:55	0.60	1.2
Diameter Ratio	$\frac{TA}{B}^{1/2}$	0.81	1.6	0.48	0.67
Speed Ratio	1.0	1.0	1.0	1.0	1.0
Energy Ratio	$\frac{TSVA^2}{B}$	0.31	1.6	0.22	0.85
Material Energy Ratio	$\frac{SA}{B}$	1.07	3.0	0.37	0.73

$$A = \frac{\sigma_2}{\sigma_1} \qquad B = \frac{\rho_2}{\rho_1} \qquad V = \frac{\text{Volume Factor}_2}{\text{Volume Factor}_1}$$

$$S = \frac{\text{Shape Factor}_1}{\text{Shape Factor}_2} \qquad T = \frac{\text{Stress Factor}_1}{\text{Stress Factor}_2}$$

Figure 12. Comparing Flywheels of Equal Speed to a Steel Flywheel of Conical Shape with Integral Shafts

	Equation	Composite Disc with Shaft Hole	Composite Cone with Shaft Hole	Steel Disc with Shaft Hole	Steel Disc Without Holes
Weight Ratio	1.0	1.0	1.0	1.0	1.0
Diameter Ratio	$\left(\frac{1}{VB}\right)^{1/2}$	1.5	2.1	0.62	0.62
Speed Ratio	$(TVA)^{1/2}$	0.54	.74	0.77	1.08
Energy Ratio	$\frac{SA}{B}$	1.07	3.0	0.37	0.73
Material Energy Ratio	$\frac{SA}{B}$	1.07	3.0	0.37	0.73

$$A = \frac{\sigma_2}{\sigma_1} \qquad B = \frac{\rho_2}{\rho_1} \qquad V = \frac{\text{Volume Factor}_2}{\text{Volume Factor}_1}$$

$$S = \frac{\text{Shape Factor}_1}{\text{Shape Factor}_2} \qquad T = \frac{\text{Stress Factor}_1}{\text{Stress Factor}_2}$$

Figure 13. Comparing Flywheels of Equal Weight to a Steel Flywheel of Conical Shape with Integral Shafts

	Equation	Composite Disc with Shaft Hole	Composite Cone with Shaft Hole	Steel Disc with Shaft Hole	Steel Disc Without Holes
Weight Ratio	VB	0.44	0.22	2.6	2.6
Diameter Ratio	1.0	1.0	1.0	1.0	1.0
Speed Ratio	$\left(\frac{TA}{B}\right)^{1/2}$	0.81	1.6	0.48	0.67
Energy Ratio	SVA	0.47	.67	0.96	1.9
Material Energy Ratio	$\frac{SA}{B}$	1.07	3.0	0.37	0.73

$$A = \frac{\sigma_2}{\sigma_1} \qquad B = \frac{\rho_2}{\rho_1} \qquad V = \frac{\text{Volume Factor}_2}{\text{Volume Factor}_1}$$

$$S = \frac{\text{Shape Factor}_1}{\text{Shape Factor}_2} \qquad T = \frac{\text{Stress Factor}_1}{\text{Stress Factor}_2}$$

Figure 14. Comparing Flywheels of Equal Diameter to a Steel Flywheel of Conical Shape with Integral Shafts

	Equation	Composite Disc with Shaft Hole	Composite Cone with Shaft Hole	Steel Disc with Shaft Hole	Steel Disc Without Holes
Weight Ratio	$\frac{B}{SA}$	0.93	.34	2.7	1.4
Diameter Ratio	$\left(\frac{1}{SVA}\right)^{1/2}$	1.45	1.22	1.02	0.73
Speed Ratio	$\left(\frac{TSVA^2}{B}\right)^{1/2}$	0.57	1.28	0.47	0.85
Energy Ratio	1.0	1.0	1.0	1.0	1.0
Material Energy Ratio	$\frac{SA}{B}$	1.07	3.0	0.37	0.73

$$A = \frac{\sigma_2}{\sigma_1} \qquad B = \frac{\rho_2}{\rho_1} \qquad V = \frac{\text{Volume Factor}_2}{\text{Volume Factor}_1}$$

$$S = \frac{\text{Shape Factor}_1}{\text{Shape Factor}_2} \qquad T = \frac{\text{Stress Factor}_1}{\text{Stress Factor}_2}$$

Figure 15. Comparing Flywheels of Equal Energy to a Steel Flywheel of Conical Shape with Integral Shafts

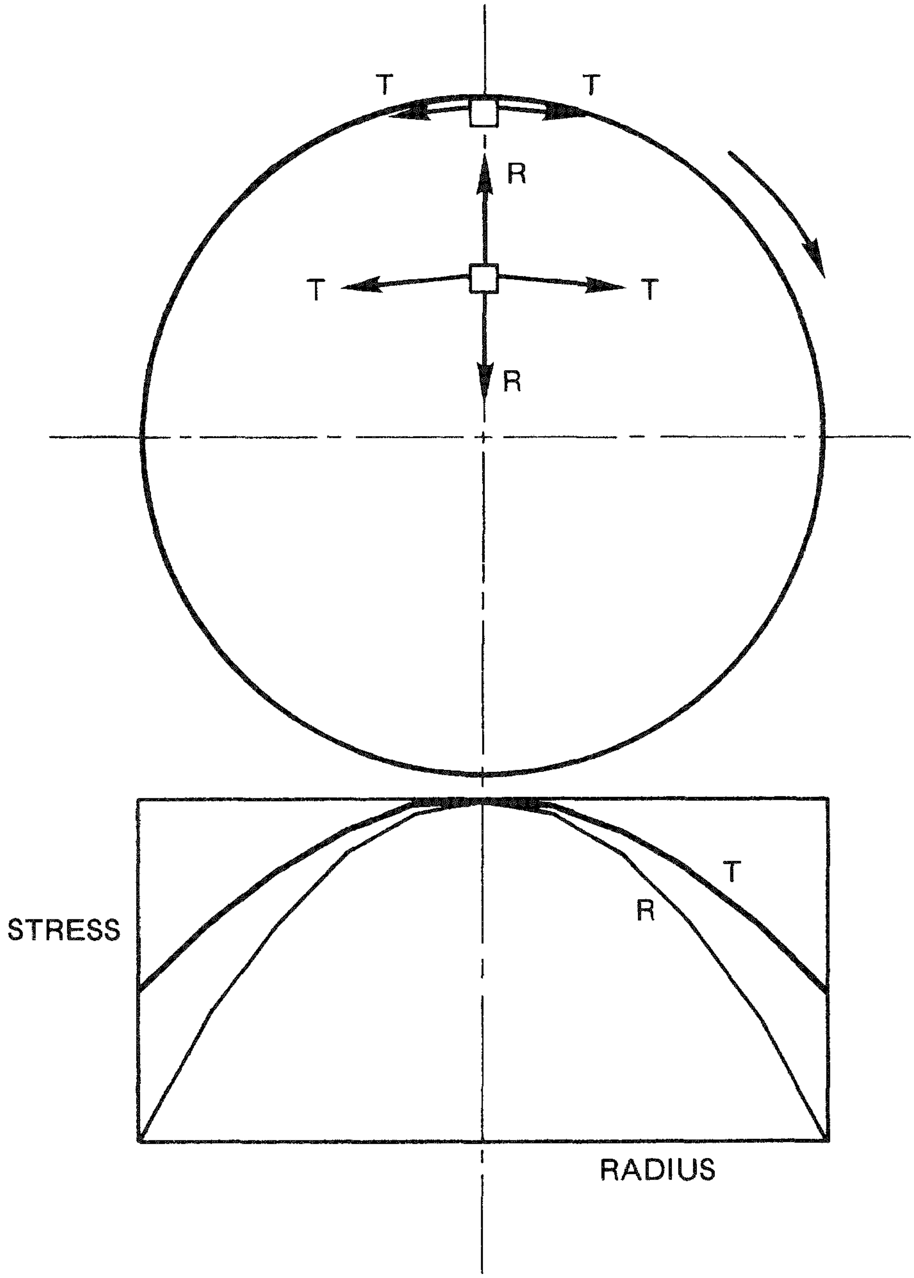


Figure 16. Orthogonal Stresses in a Solid Disc Flywheel

energy per pound as does steel. In actual flywheels, the construction determines the efficiency with which the material is used up to its capability. Good steel wheels realize 80% of the material capability, and with composites the number is in the vicinity of 40% due to their inability to carry stress normal to the fibers. Thus, the ratio of energy densities becomes 10 x .4/.8=5. Further, the fiber yarns fail somewhat like a chain in that the weaker fibers determine the strength (Ref. 3), and this decreases their potential strength to .8 of the maximum. In addition, the fiber strength is reduced by the volumetric ratio of the epoxy binder which is at best a .7 factor. So the 10 to 1 advantage in energy density is now cut to 5 x .8 x .7=2.8. This figure needs to be further adjusted for fatigue cycling of the stresses, which for steel might be 1/3 and for the composites 1/2. Therefore, 2.8 x 3/2=4 is an approximation of the possible energy density of composites relative to steel. This is a very respectable goal, especially if weight is important. Composites also save additional weight in containment since burst failure produces shreds instead of shrapnel.

The general conclusions in the preceding Section 3.2 indicated that composite flywheels could offer advantages for the tractor-trailer car if a stress level of 190,000 psi were achieved (specific energy of 30-40 Whrs/lb. Since the density of steel is approximately 6 times the density of composites, an achievable stress level in composites of only 1.5 times steel is required to achieve the fourfold improvement in energy density. See Equation 16, Figure 9.

3.3.2 Construction

To make maximum use of composites in view of the directional characteristic of fibers, one recourse is to lay up successive hoop layers with softer material between them. This eliminates the radial tensile stress and leaves the fibers in true tensile hoop stress (Figure 17). The opposite extreme is to bundle the fibers at the center of rotation and let them flare out radially--bound as a disc with epoxy. With epoxy having a lower modulus than the fiber, these fibers experience only radial tension (Figure 18). Between these two there are numerous weaves and layouts that are being explored.

3.3.3 State of Development

Currently the composites are actually in the development stage with the main thrust being to achieve with laboratory models a respectable amount of the theoretical capability predicted. The second step will be to explore composites in larger than laboratory sizes and to determine working stress ranges under cyclic fatigue conditions. The third and final step will have to be one of avoiding the hand labor involved in the construction and of controlling the curing processes to get uniform results. The state-of-the-art seems to be located in the second step for the composite wheels

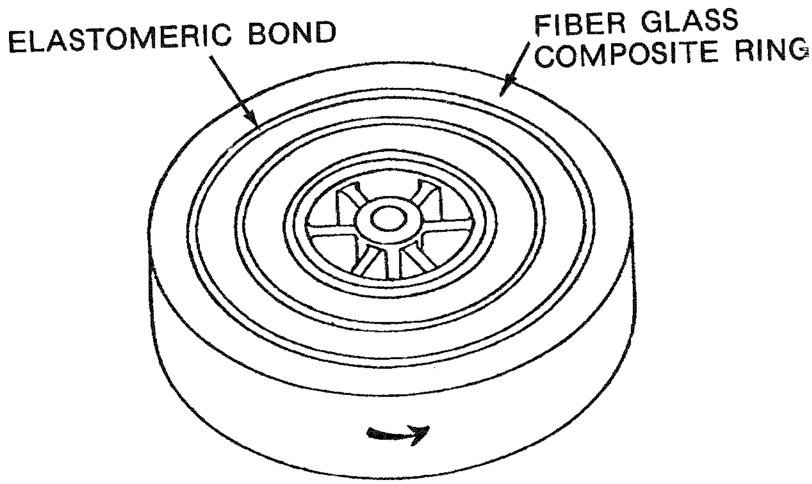


Figure 17. Multiple Ring Design with Fibers Circumferential

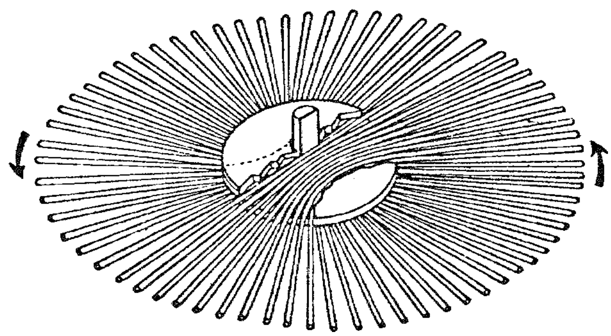


Figure 18. Radial Fiber Design

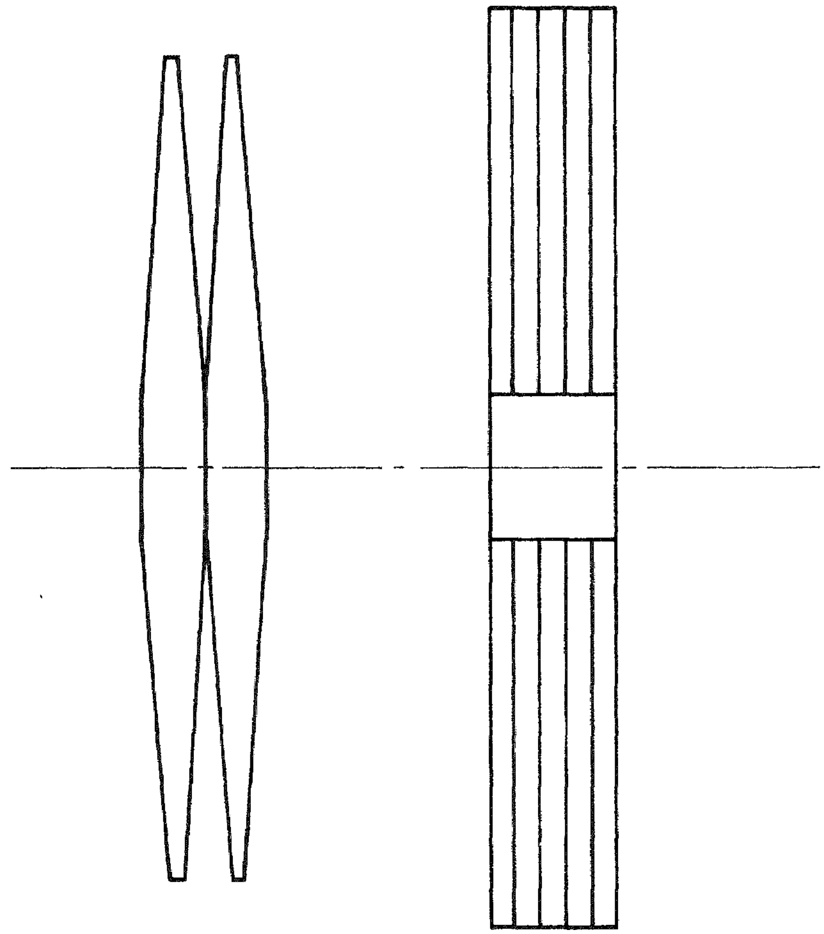
that are under development. The only type of construction that is immediately available for application appears to be a flat laminate disc built up of layers of E glass filaments with each layer oriented 60° from its neighbor layers (Scotchply 1002 isotropic sheets). Some work is presently being done with Kevlar which has a higher working stress which could reduce the required thickness of the flywheel.

3.3.4 Steel Versus Composite for the Tractor-Trailer Shuttle Car

Figure 19 directly compares a conical steel flywheel to a composite disc flywheel, both having the same energy capacity, same speed, and same thickness. Under these constraints, the composite wheel calculates to be storing 12 Whrs/lb which does not realize much of the potential capacity for fibers and as such does not offer much advantage over a conical steel wheel. Thus, the theoretical advantages of composites indicated in Section 3.2 are not realized due to the low stress level used. The 72,000 psi stress in the composite may be optimistic since two similar wheels--but with bolt holes that produce stress concentrations--burst at stresses of 65,000 and 70,000 psi. These wheels were constructed as part of a program at the General Electric Company sponsored by the Energy Research and Development Administration (ERDA). The low burst strength of the laminate discs comes as a surprise since this stress is below that for steel. This has given rise to the use of the "fiber use efficiency" mentioned on page and listed in Figure 10. The energy density calculates to be one-third that of estimates of the near state-of-the-art.

A very good rundown on many of the programs with composite flywheels is to be found in Reference 6. The authors anticipate that in the "near future" composite wheels will achieve 30 to 40 Whrs/lb (as compared to 10 to 12 Whrs/lb in steel). If so, there still would be the necessary two developmental steps in preparation for general application at these levels.

This study has shown that further composite flywheel development and evaluation is required to obtain the technology to design a flywheel for the shuttle car application. However, steel flywheels can be designed according to established engineering principles, and it is recommended that this approach be pursued for the shuttle car. This design avenue will lead to a more timely proof of the viability of a flywheel-powered shuttle car.



STEEL 4330	SCOTCHPLY E-GLASS
10,000 rpm	10,000 rpm
4.5 kWh (useable)	4.5 kWh (useable)
4.5'' THICK	4.5'' THICK
40.7'' O.D.	49.3'' O.D.
850 lb	430 lb
82,000 psi	72,000 psi
7.0 Wh/lb	14.0 Wh/lb

Figure 19. Comparison of Composite and Steel Flywheel

REFERENCES

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2. John M. Woods and Louis J. Lawson, "Energy Storage--Propelled Transit Vehicle Application Study," Final Report, April 30, 1975 for San Francisco Municipal Railway Improvement Corporation and U.S. Department of Transportation, Urban Mass Transportation Administration.

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4. "Economic and Technical Feasibility Study for Energy Storage Flywheels," Rockwell International Space Division for Energy Research and Development Administration, Division of Conservation Research and Technology Office of Conservation, Contract AT(04-3)-1066, December 1975, p. 5-11 to 5-23.

5. R.J. Roark, Formulas for Stress and Strain Fifth Edition, McGraw-Hill, pp. 566-567.

6. Amstutz, Cooper, Heckl, Zweig and D'Andrea, "Flywheels and Pulsed Power," preliminary information to final report by the United States Army (MERADCOM, Watervliet Arsenal, AMMRC) to DOT, 1977, September 1976.

Section 4

FLYWHEEL PACKAGE AND ELECTRICAL SYSTEM PACKAGE FOR SHUTTLE CAR

This section discusses the conceptual design of the major components of a flywheel energy storage system. The system is perceived to consist of: a 4.5 kW hr flywheel directly coupled to a motor/alternator, a suitable housing which provides flywheel containment and a partial vacuum environment, and flywheel recharging equipment located on the wayside and consisting primarily of a load commutating inverter. Included are discussions of:

- The flywheel motor/alternator and its enclosure including provisions for mounting in a vehicle
- Inductor motor/alternators capable of recharging 4.5 kW hrs of energy in 30 and 80 seconds
- The wayside and on-board electrical equipment for two configurations: an inductor motor/alternator-rectifier on the vehicle with a load commutating inverter at the wayside, and an inductor motor/alternator-LCI on the vehicle

4.1 SUMMARY AND CONCLUSIONS

The conceptual design of the flywheel package and the wayside electrical equipment has indicated that:

- A number of design areas require further investigation (see Sections 4.4.7, 4.5.7, 4.6.7, 4.7).
- A 4.5 kW hr usable energy flywheel package consisting of a conical steel flywheel directly coupled to an inductor motor/alternator is a feasible design approach.
- It is also feasible to build a multi-disc flywheel for greater energy storage (6.0 kW hr or more usable energy). With additional shuttle cars to compensate for longer tram time, this arrangement can use the same motor and provide extended tram distances with no sacrifice in productivity.
- The flywheel package should be evacuated to minimize windage losses.
- The flywheel package will require an auxiliary heat transfer system.

- Mounting means to accommodate the gyroscopic action of the flywheel must be provided.

- Since regeneration offers no significant benefit (Section 5.4.7) a relatively simple on-board rectifier may be used to convert flywheel motor/generator power to a dc voltage to power the vehicle.

- A wayside inverter can be designed to charge the flywheel.

The flywheel material and shape was determined by the analysis of flywheel technology in Section 3. The selection of the inductor/alternator-type machine for driving the flywheel was made based on an independent internal General Electric study comparing: synchronous, induction, and dc motors, with their attendant power conditioning, for flywheel drives. The principal factors favoring the inductor machine are its solid rotor with no windings and its high speed capability which eliminates gearing between the motor/alternator and flywheel. The inductor machine size was selected based on the system trade-offs in Section 5. The motor is sized to provide the peak torque required for acceleration. There is no significant penalty in size for the charging duty. Similarly, the wayside power supply equipment is not penalized because of the charging duty. The sizing of the wayside inverter is consistent with the duty cycle of the maximum charge time (80 seconds).

4.2 OVERALL FLYWHEEL PACKAGE DESCRIPTION

The complete package is composed of two main assemblies: the Capsule (Figure 20), which houses the rotating assembly and provides containment and the vacuum housing, and the Enclosure (Figure 21), which provides a spherical bearing to allow the Capsule to tilt and also provides protection from the operating environment.

Figure 20 illustrates the flywheel capsule design concept consisting of a conical steel flywheel constructed by electron beam joining to the shaft sections, and an inductor motor sized for accepting 4.5 kW hrs of energy in 80 seconds (203 kW).

The motor is used for spinning up the flywheel in the charging cycle and as an alternator to provide propulsion power during shuttle car operation. The motor has a solid rotor and all windings are on the stator making it possible to operate at the required 10,000 rpm top speed.

Duplex pair bearings illustrated are used in both ends of the shaft. Since the axis is vertical, the bearings will be loaded in thrust.

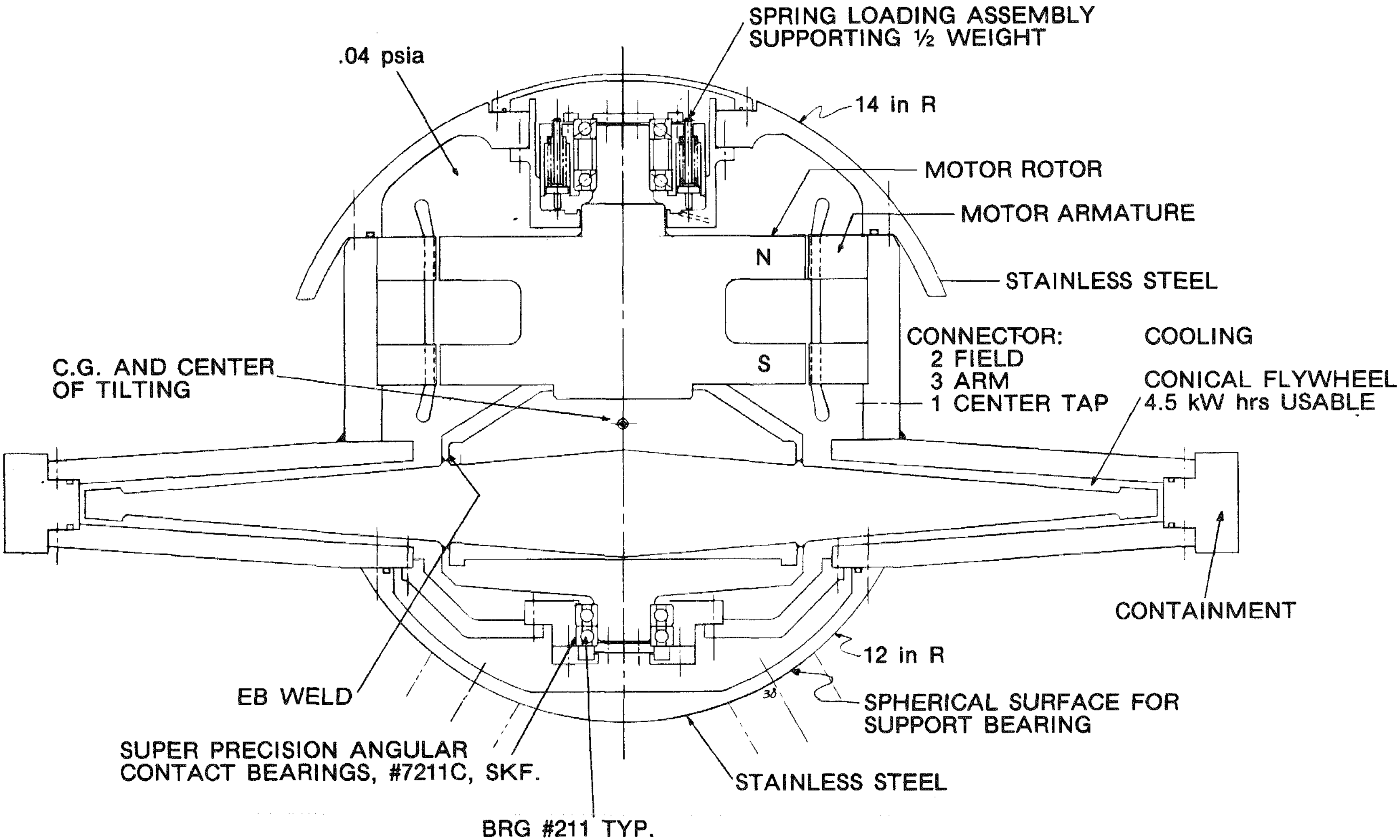


Figure 20. Flywheel Capsule

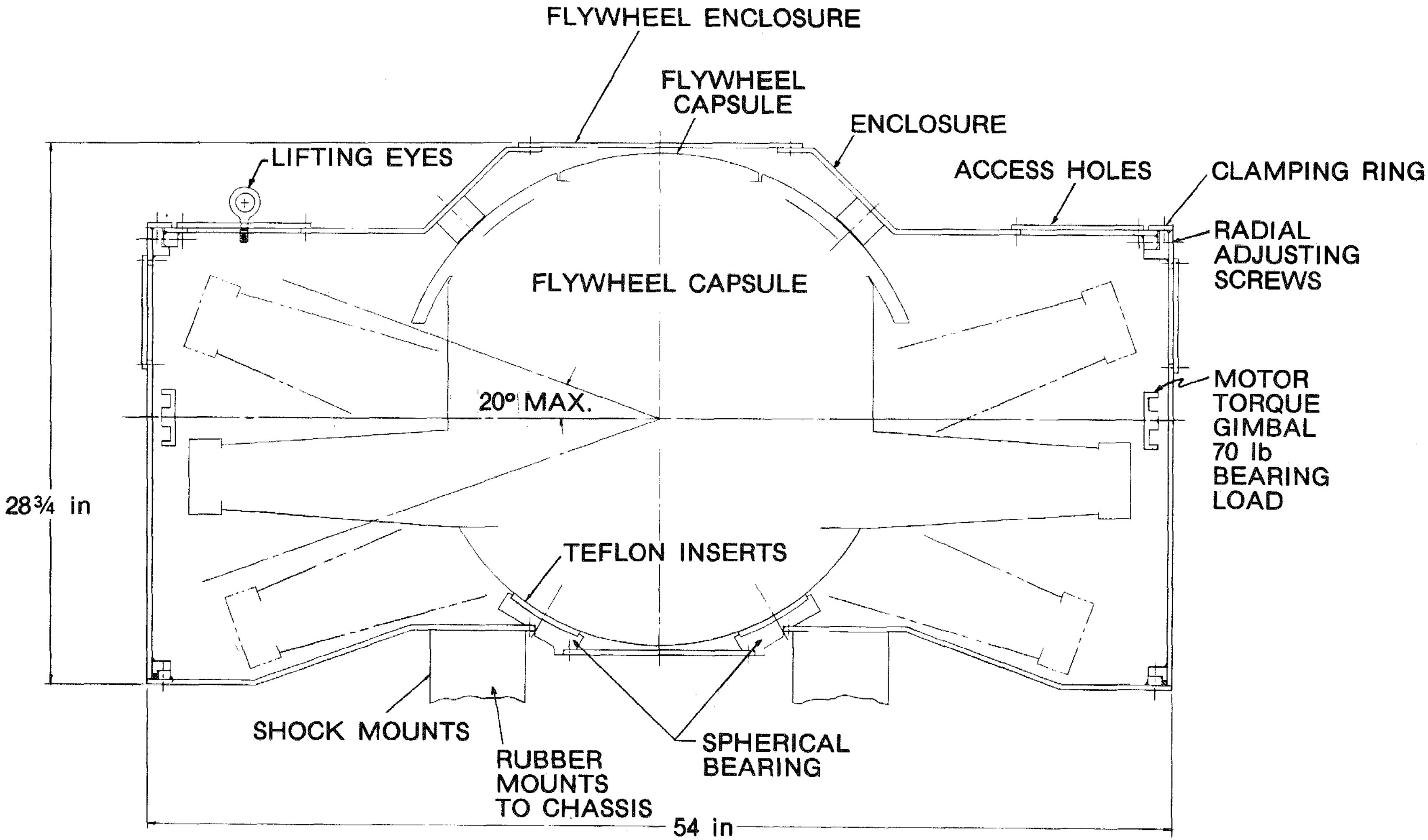


Figure 21. Flywheel Package (4.5 kW hrs)

The capsule is evacuated with a pressure of 0.04 psia. Static O ring seals are utilized. No rubbing seals are required.

The flywheel diameter is approximately 43 inches for 4.5 kW hrs of usable energy. A containment ring approximately 2 inches thick is required.

The package including the capsule is approximately 54 inches in diameter, 29 inches high, and weighs approximately 2,000 pounds.

The capsule is freely mounted in the enclosure (Figure 21) to minimize gyroscope forces on the flywheel bearings due to actions of both vehicle motion and motor torque.

External cooling means are required to accommodate the heat loss in the inductor motor rotor and the flywheel windage. This is accomplished by cooling tubes mounted on the outside of the capsule, Figure 45.

4.3 INDUCTOR MACHINE

This subsection describes the Inductor Machine and discusses two alternate approaches, a machine sized for a 30-second charge time for 4.5 kW hrs and a machine sized for peak torque requirement for operating the vehicle. This latter machine requires 80 seconds to charge 4.5 kW hrs. The trade-offs in machine design are discussed and alternate designs displayed.

4.3.1 Description of the Inductor Machine (motor/alternator)

The basic function of the inductor machine can best be explained by means of Figures 22 and 23. Figure 22 shows a dc field coil between two stacks of stator laminations. A dc current in the field coil will drive a magnetic flux as indicated through one stack of laminations into the rotor through the rotor center axially and radially out of the rotor through the second stack of laminations and the frame to close the loop. Large magnetic slots in the rotor will interrupt the flux at the air-gap and cause the flux through the ac windings to pulsate. This, in turn, generates an ac voltage in these windings. The windings are located in slots in the laminated stator stacks close to the air gap. To make the induced voltage in both stator stacks add properly, the magnetic rotor slots in both halves will be offset by one-half pole pitch.

4.3.2 Alternate Inductor Machines

Machines were studied which were sized by the peak torque requirements in operating the vehicle and by the requirement for a 30 second charge time for 4.5 kW hrs

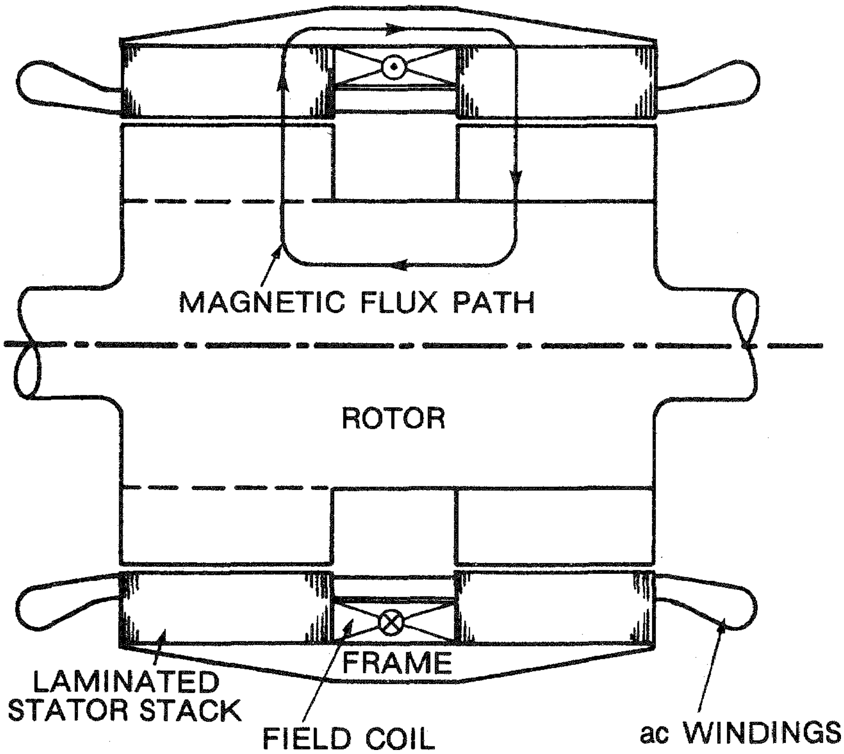


Figure 22. Cross Section of an Inductor/Alternator

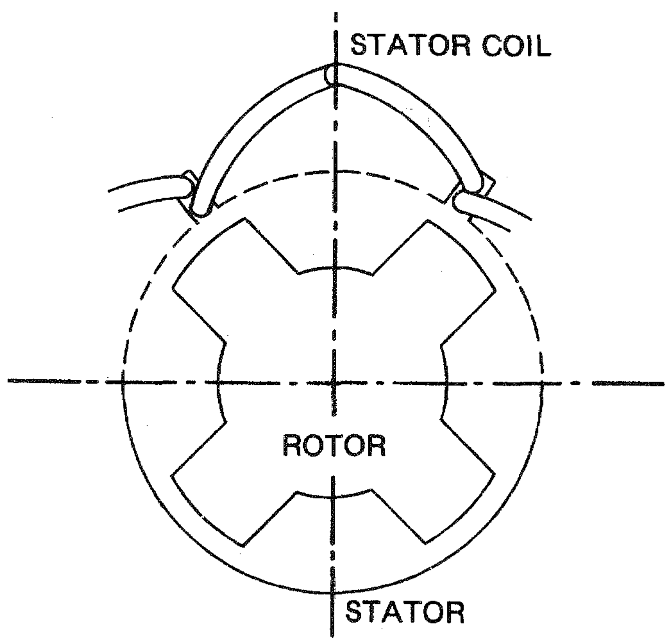


Figure 23. End View of an Inductor/Alternator

energy. The machines sized for the 30 second charge are considerably larger than the size determined by peak torque requirement. The primary trade-offs were made based on discharge efficiency, although consideration was given to frequency and machine reactance since these represent limitations on the charging inverter.

Inductor Machine Trade-Offs for Machine Sized for Charging Duty (540 kW Machine)

The peak alternator rating of this machine is 375 A (600 volts dc output from rectifier). This data was extrapolated for a 75 hp shuttle car traction motor from a 40 hp motor oscillogram (Section 2, Figure 1).

A charging requirement of 4.5 kW hrs in 30 seconds requires

$$4.5 \text{ kW hrs} \times \frac{3600 \text{ sec/hr}}{30 \text{ sec}} = 540 \text{ kW or } 724 \text{ hp}$$

When the principal design criterium is minimum charging time, maximum motor current is the limiting design parameter. If the machine size is determined by the maximum generating load requirements, then the minimum charge time is determined by the maximum current capability and the duty cycle of the shuttle car mission.

The efficiency of the machine at the above power level significantly influences the machine size and weight as shown below for two efficiency ratings.

All of the machines studied had to meet a given ratio of air gap voltage to dc current times commutating reactance. This ratio defines the subtransient short-circuit current capability. This is necessary to insure the load commutating capability of the machine when operating as a motor (during the charging process). This requirement automatically determines the maximum number of turns/phase within the machine. Independent parameters which were varied during these trade-off studies are:

Flux density in the stator yoke	(By)
Current density in the ac windings	(CD)
Base diameter	(Sta. Dia)
Number of pole pairs	(p)
Rotor speed	(rpm)

The results of the trade-off studies have shown that there is an optimum value of current density which results in minimum losses. This current density is below 10,500 amps/in^2. The total losses increase with increasing core flux density from By = 60 to 90 kL/in^2 for given copper losses. There is no optimum value of core flux density for minimum total losses.

The results of this study have to be considered with two limitations in mind. One limitation is given with respect to the fatigue stress limits in the rotor and flywheel. The stress limitation is determined by flywheel diameter, rotor diameter and rotor speed. The second limitation is the frequency limit for the inverter. The limit is determined by minimum turn-on, turn-off, and power dissipation characteristics of available power semiconductors and is around 1000 Hz. This represents the maximum realizable frequency at full speed (10,000 r.p.m.). Reasonable design margins coupled with the necessity for an integral number of motor poles lead to a maximum of 417 Hz at base speed (5,000 r.p.m.).

Table XV lists the final results for a constant base frequency of 417 Hz for two efficiency levels for normal operation:

Table XV

Machine Parameters for Two Efficiency Levels

Efficiency	.911	.911	.921	.921
Pole pairs	5	4	5	4
Rpm	5,000	6,200	5,000	-
Bore diameters	20"	20"	18"	-
Weight	1,517	1,499	1,620	- lbs
Current density	6	6	7.5	- KA/in^2
Core flux density	80	70	60	- KL/in^2

As is seen, a lower number of pole pairs and a higher rotor speed result in a lighter machine. The high efficiency, however, cannot be achieved by the 4-pole pair machine.

It should be noted that efficiency in this study is electrical efficiency only. No mechanical losses (windage, friction) are considered.

Table XVI shows the same results for a higher base frequency of 500 Hz.

Table XVI

Machine Parameters for Two Efficiencies

Efficiency	.911	.911	.911	.921	.921	.921
Pole pairs	4	5	6	4	5	6
Rotor speed (rpm)	7,500	6,000	5,000	7,500	6,000	5,000
Bore diameter (in)	21.4	19	19	-	18	19
Weight (lbs)	1,284	1,290	1,374	-	1,375	1,448
Current density (KA/in^2)	9	7.5	7.5	-	7.5	7.5
Flux density (KL/in^2)	65	75	80	-	60	60

It can be seen again that a low number of pole pairs combined with a high rotor speed results in the lowest weight machine. However, a thermal analysis is mandatory for the 7500 rpm base speed machine with a charging current density of 9000 A/in^2.

Table XVII shows results for two constant number of pole pairs, while Table XVIII shows the results for a constant rotor speed.

Table XVII

Machine Parameters for Two Efficiencies

a) <u>4 POLE PAIR MACHINES</u>

Efficiency	.911	.911	.911	.921	.921	.921
Rotor speed	5,000	6,250	7,500	5,000	6,250	7,500
Bore diameter (in)	21.0	20.0	18.25	-	-	-
Weight (lbs)	1,836	1,499	1,284	-	-	-
Current density (KA/in^2)	6	6	9	-	-	-
Yoke flux dens. (KL/in^2)	70	70	65	-	-	-
Frequency (Hz)	333	417	500	333	417	500

b) <u>5 POLE PAIR MACHINES</u>

Efficiency	.911	.911	.921	.921
Rotor speed	5,000	6,000	5,000	6,000
Bore diameter (in)	20	19	18	18
Weight (lbs)	1,517	1,290	1,620	1,375
Current density (KA/in^2)	6	7.5	7.5	7.5
Yoke flux dens. (KL/in^2)	80	75	60	60
Frequency (Hz)	417	500	417	500

Table XVIII

Machine Parameters for Two Efficiencies (rpm = 5000)

Efficiency	.911	.911	.911	.921	.921	.921
No. of pole pairs	4	5	6	4	5	6
Frequency (Hz)	333	417	500	333	417	500
Bore diameter (in)	21.0	20.0	19.0	-	18.0	19.0
Weight (lbs)	1,836	1,517	1,374	-	1,620	1,448
Current density (KA/in^2)	6	6	7.5	-	7.5	7.5
Yoke flux dens. (KL/in^2)	70	80	80	-	60	60

These tables show that the machine size and weight can be reduced by raising speed, and furthermore the combination of high speed and low frequency is desirable from a weight standpoint. Also higher efficiency generally means increased size and weight.

For purposes of sizing the appropriate machine, computer runs were made of the machines in the first two columns of Table XV and shown on the oscillograph printouts Figures 26 and 27. The sketch (Figure 24) provides detailed information of machine dimensions and parameters. The code for computer printout (Figure 25) and the sketch (Figure 24) define terms used in the computer printout.

The 5000 rpm machine is 18.66 inches long, with a bore diameter of 20 inches, an outer diameter of 27.43 inches, and a weight of 1517 pounds.

Inductor Machine Trade-offs for Machines Sized for Peak Current Requirements During Operation

The machine sized for 4.5 kW hrs charge in 30 seconds is heavy (1517 pounds) and seriously impacts on the wayside power substation costs and the required charging inverter. It may even affect the high voltage mine distribution system. The mine productivity analysis (Section 5.3) indicates that longer charge times are permissible, and thus design trade-offs have been made for a machine which is sized for the maximum required torque represented by 375 amps and 600 volts dc. The charging time required for these designs is in the neighborhood of 80 seconds and results in a weight of between 700 and 800 pounds.

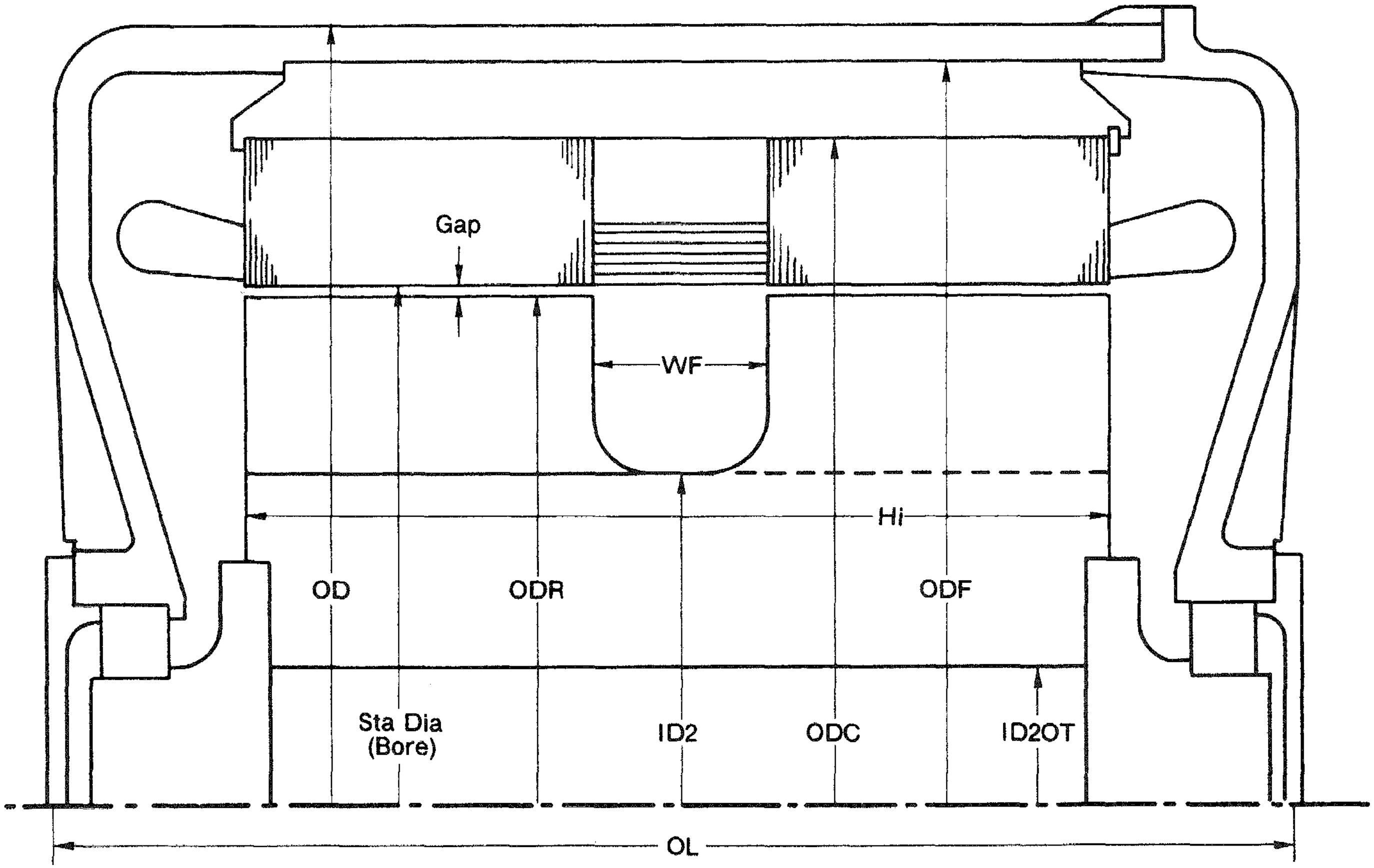


Figure 24. Inductor/Alternator Cross Section

XC/XCOMM	COMMUTATING REACTANCE
XAD	SYNCHRONOUS REACTANCE IN D AXIS
XLEAK	LEAKAGE REACTANCE
XD	= XAD + XLEAK
WS	SLOT WIDTH
WT	TOOTH WIDTH
H14	SLOT DEPTH
HYC	CORE HEIGHT

WCU	COPPER (ac & FIELD)	WEIGHT
WC	CORE	
WFR	FRAME (dc)	
WH	HOUSING	
WROT	ROTOR (INCL. BEARINGS)	
WTOT	TOTAL	

PCU	I^2R (FIELD & AO)	LOSSES
PFE	IRON	
PTOT	TOTAL	

Figure 25. Code for Computer Printout

Based upon background work done on flywheel energy storage units, a Five-Pole Pair Machine operating between 5000 and 10000 rpm at a frequency range of 417 to 834 Hz is the most likely candidate to be used. This is based upon the diameter speed combination and the resulting rotor fatigue life, as well as upon the fact that 834 Hz maximum frequency is more readily handled by the power inverter. Trade-off studies have been performed for this machine for equal charging and generating rates. For different efficiencies the following optimum bore diameters were found:

Table XIX

Machine Sized for Torque Requirements

MACHINE	EFFICIENCY GENERATING (Mg)	EFFICIENCY CHARGING (MCH)	DIAMETER BORE - in	WEIGHT lbs	CHARGE TIME sec
1	.920	.915	13.1	794	78.6
2	.910	.903	15.0	716	79.7
3	.900	.891	15.0	684	80.7

Also given in Table XIX are the times necessary to charge the 4.5 kW hr flywheel at a constant power rate as determined by the 375 ampere maximum current rating and the calculated charging efficiency. Detailed dimensions may be found in the attached computer printouts (Figures 28, 29, and 30. Also shown is a curve sheet showing the function of machine weight versus diameter for constant efficiency (Figure 31).

The trade-offs clearly show that improvements in efficiency result in added weight. For the shuttle car application the 110 pound difference between .891 efficiency and .915 efficiency is not significant. Therefore in the interests of increased efficiency the heavier machine is recommended as a base for subsequent design work.

15000 RPM Machine

A preliminary study of a 15,000 rpm machine for an 80 second charge time indicates that a design is feasible. There may be a weight penalty of about 15 percent because a high efficiency machine which requires more copper and iron must be used to reduce the diameter. This means longer length and smaller diameter. The rotor material must be vacuum melted 4340 steel rather than common 4340 steel which is planned in the baseline machine.

```
INDUCTOR ALTERNATOR TRADEOFF FOR RECTIFIED OUTPUT

DC OUTPUT    U = 600.00     I =  980.5

    ESTIMATED DATA
     DIODE DROP     = 2.00 VOLT
     PHASE RESIST.=0.007883 OHM
     XC/XAD         =0.700

BASIC MACHINE DATA
 STA DIA    = 20.000 INCH   SPEED      =  5000. RPM
 POLEPAIRS =        5.       FREQUENCY =  416.7 HZ
 PUPA       =  0.500         HSC        =  0.065 INCH
 PHASES     =       3.
   AC WDG    CIR      3.  TPC 1.000  FILL 0.660  CD  6000.
   DC WDG    CDF   4000.  APF 1.600  FILL 0.660
   FLUXDENSITIES IN KL/IN2
     GAP     70.00  CORE    80.00  FRAME  80.00
     ROTOR   95.00  TOOTH 120.00
     STACK FACT  0.92

OUTPUT
GAP       0.210
NEFF     17.270
XAD     0.24800
XLEAK     0.042
XD        0.290
XCOMM     0.215
HI        8.289
SI      155.43        WEI  1.00  WEX 1.00  WX  0.49
WS        0.148       UDG  600.  IDG 375.
WT        0.256       EI/XC/IDC   1.661
H14       0.520       EFFCH       0.9177
HYC       2.272       EFFGE       0.9112
ODC      25.584
ODR      19.580
ID2      11.181
IDROT     6.336
BC       94.500
ODF      27.473
WF        2.573
ETL       5.802
```

Figure 26.

Inductor Alternator Machine-5000 rpm Base Speed for 30 Second Charge Duty

```
ETE       6.802
OL       18.664
OD       28.673
WCU        135.
WC         420.
WFR        251.
WH         152.
WR         558.
WTOT      1517.
PCU      27657.
PFE      20745.     PFT    4835.    PFC    15910.
PTOT     48401.
NSECT         1
SALNY         1
RA      0.00798
```

```
DC OUTPUT    U = 600.00      I =   981.7

     ESTIMATED DATA
      DIODE DROP    = 2.00 VOLT
      PHASE RESIST.=0.008093 OHM
      XC/XAD        =0.700

BASIC MACHINE DATA
 STA DIA    = 20.000 INCH    SPEED      =  6250. RPM
 POLEPAIRS =      4.         FREQUENCY =  416.7 HZ
 PUPA       =   0.500        HSO        =  0.065 INCH
 PHASES     =      3.
   AC WDG   CIR      3.   TPC 1.000   FILL 0.660   CD   6000.
   DC WDG   CDF   4000.   APF 1.600   FILL 0.660
   FLUXDENSITIES IN KL/IN2
     GAP     70.00  CORE    70.00  FRAME   80.00
     ROTOR   95.00  TOOTH 120.00
     STACK FACT  0.92

OUTPUT
GAP       0.264
NEFF     16.309
XAD     0.23440
XLEAK     0.051
XD        0.286
XCOMM     0.215
HI        7.028
SI      146.78      WEI   1.00   WEX 1.00   WX   0.50
WS        0.157     UDG   600.   IDG 375.
WT        0.271     EI/XC/IDC    1.661
H14       0.495     FFFCH        0.9166
HYC       3.245     FFFGE        0.9108
ODC      27.481
ODR      19.472
ID2       8.902
IDROT     2.701
BC       94.500
ODF      28.986
WF        2.252
ETL       7.244
ETE       8.244
OL       18.523
OD       30.186
WCU        161.
WC         507.
WFR        182.

WH         161.
WR         488.
WTOT      1499.
PCU      29851.
PFE      19256.     PFT    3904.    PFC    15353.
PTOT     49108.
NSECT         1
SALNY         1
RA      0.00909
```

Figure 27.

Inductor Alternator Machine-
6250 rpm Base Speed for 30
Second Charge Duty

```
BASIC MACHINE DATA
 STA DIA    = 13.100 INCH    SPEED      =  5000. RPM
 POLEPAIRS  =      5.        FREQUENCY  =  416.7 HZ
 PUPA       =   0.500        HSO        =  0.065 INCH
 PHASES     =      3.
   AC WDG   CIR      3.  TPC 1.000  FILL 0.660  CD  4500.
   DC WDG   CDF  4000.  ARF 1.600  FILL 0.660
   FLUXDENSITIES IN KL/IN2
     GAP    70.00  CORE   60.00  FRAME  80.00
     ROTOR  95.00  TOOTH 120.00
     STACK FACT  0.92

OUTPUT
GAP       0.140
NEFF     28.546
XAD     0.62491
XLEAK     0.117
XD        0.742
XCOMM     0.554
HI        7.645
SI      256.91       WEI  1.00   WEX 1.00   WX  0.46
WS        0.059      UDG  600.   IDG 380.
WT        0.102      EI/XC/IDC   1.662
H14       0.659      EFFCH       0.9150
HYC       1.984      EFFGE       0.9207
ODC      18.385
ODR      12.820
ID2       7.221
IDROT     1.000
BC       94.729
ODF      19.964
WF        1.930
ETL       3.890
ETE       4.890
OL       15.575
OD       21.164
WCU         72.
WC         249.
WFR        134.
WH          88.
WR         251.
WTOT       794.
PCU      10450.
PFE       8922.      PFT    3698.   PFC    5224.
PTOT     19372.
NSECT         1
SALNY         1
RA      0.01930

INDUCTOR ALTERNATOR TRADEOFF FOR RECTIFIED OUTPUT
```

Figure 28. Inductor Alternator
(Machine 1, Table XIX

```
INDUCTOR ALTERNATOR TRADEOFF FOR RECTIFIED OUTPUT

DC OUTPUT    U = 600.00      I =  380.0

     ESTIMATED DATA
      DIODE DROP     = 2.00 VOLT
      PHASE RESIST.=0.022069 OHM
      XC/XAD         =0.700

BASIC MACHINE DATA
 STA DIA    = 15.000 INCH    SPEED       =  5000. RPM
 POLEPAIRS =       5.        FREQUENCY =  416.7 HZ
 PUPA       =  0.500         HSO         =  0.065 INCH
 PHASES     =      3.
   AC WDG    CIR      3.  TPC 1.000  FILL 0.660   CD   4500.
   DC WDG    CDF   4000.  ARF 1.600  FILL 0.660
   FLUXDENSITIES IN KL/IN2
     GAP      70.00  CORE     80.00  FRAME   80.00
     ROTOR    95.00  TOOTH 120.00
     STACK FACT  0.92

OUTPUT
GAP        0.161
NEFF      31.990
XAD      0.61398
XLEAK      0.127
XD         0.741
XCOMM      0.557
HI         5.981
SI        287.91      WEI   1.00   WEX 1.00   WX   0.48
WS         0.060      UDG  600.    IDG 380.
WT         0.104      EI/XC/IDC     1.660
H14        0.646      EFFCH        0.9029
HYC        1.704      EFFGE        0.9105
ODC       19.699
ODR       14.679
ID2        8.249
IDROT      4.702
BC        94.500
ODF       21.031
WF         2.573
ETL        4.424
ETE        5.424
OL        14.978
OD        22.231
WCU          85.
WC          189.
WFR         107.
WH           92.

   WR           242.
   WTOT         716.
   PCU        12249.
   PFE         9895.     PFT      3249.    PFC      6646.
   PTOT       22144.
   NSECT           1
   SALNY           1
   RA        0.02207
```

Figure 29. Inductor Alternator
(Machine 2, Table XIX)

```
INDUCTOR ALTERNATOR TRADEOFF FOR RECTIFIED OUTPUT

DC OUTPUT    U = 600.00      I =  380.0

    ESTIMATED DATA
     DIODE DROP     = 2.00 VOLT
     PHASE RESIST.=0.029862 OHM
     XC/XAD         =0.700

BASIC MACHINE DATA
 STA DIA    = 15.000 INCH    SPEED       =  5000. RPM
 POLEPAIRS =       5.        FREQUENCY =  416.7 HZ
 PUPA       =  0.500         HSO         =  0.065 INCH
 PHASES     =      3.
   AC WDG   CIR       3.   TPC 1.000   FILL 0.660   CD   6000.
   DC WDG   CDF   4000.   ARF 1.600   FILL 0.660
   FLUXDENSITIES IN KL/IN2
     GAP     70.00   CORE    85.00   FRAME   80.00
     ROTOR   95.00   TOOTH 120.00
     STACK FACT  0.92

OUTPUT
GAP        0.161
NEFF      32.537
XAD      0.63030
XLEAK      0.121
XD         0.751
XCOMM      0.562
HI         5.935
SI        292.84     WEI   1.00   WEX 1.00   WX   0.48
WS         0.059     UDG   600.   IDG 380.
WT         0.102     EI/XC/IDC    1.661
H14        0.508     EFFCH        0.8907
HYC        1.603     EFFGE        0.9006
ODC       19.223
ODR       14.678
ID2        8.246
IDROT      4.735
BC        94.500

  ODF       20.576
  WF         2.734
  ETL        4.385
  ETE        5.385
  OL        15.054
  OD        21.776
  WCU          77.
  WC          168.
  WFR         108.
  WH           90.
  WR          242.
  WTOT        684.
  PCU       15567.
  PFE        9352.     PFT     2536.    PFC     6816.
  PTOT      24919.
  NSECT          1
  SALNY          1
  RA       0.02986
```

Figure 30. Inductor Alternator (Machine 3, Table XIX)

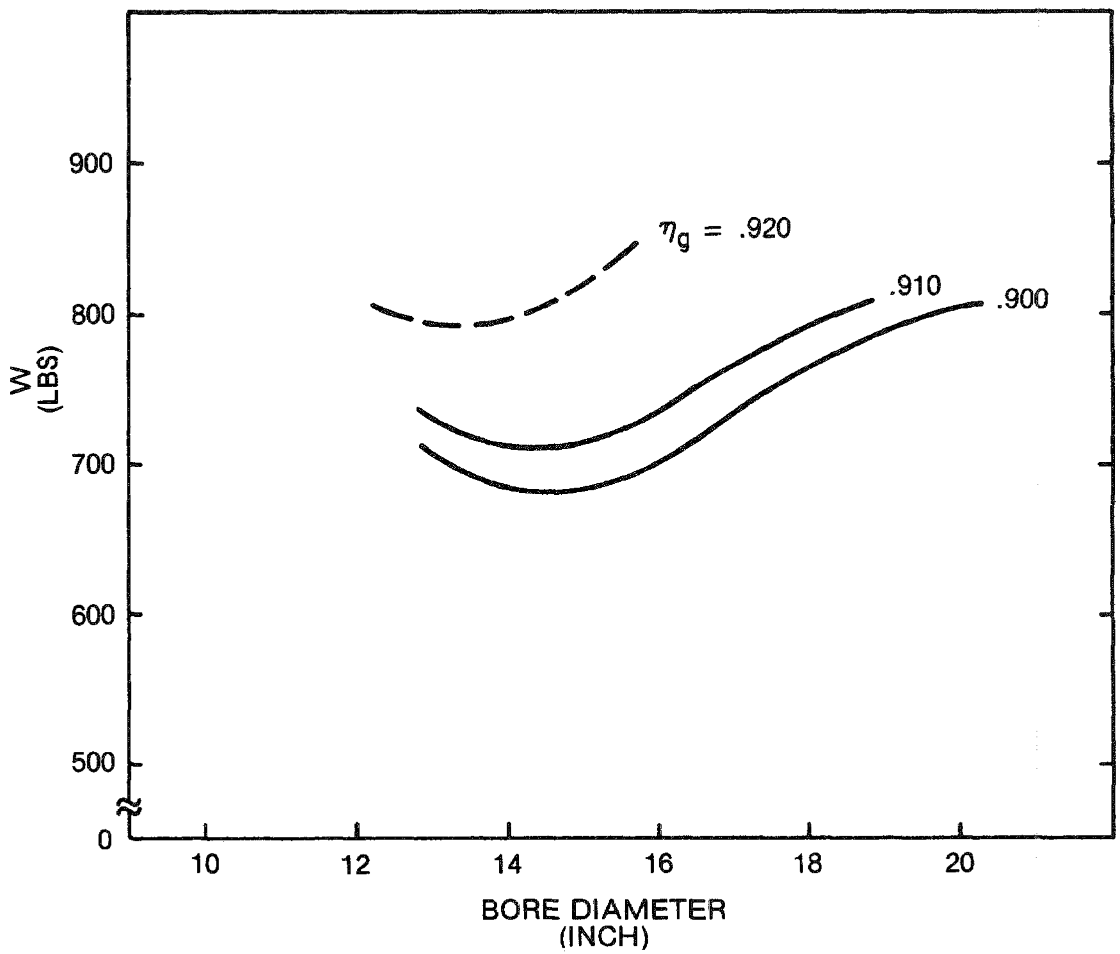


Figure 31. Weight Versus Bore Diameter for Constant Efficiency - Five Pole Pairs; 5,000-10,000 rpm; Ide = 375 A

4.4 FLYWHEEL

This subsection discusses the design of the flywheel itself, using basic equations derived in Section 4 on Flywheel Technology, Figures 7 and 8. For the present purpose, the conical steel flywheel, Style a of Figure 10 was used for the tractor-trailer shuttle car since diametral space is not a problem, but length is of concern. Topics in this subsection include Flywheel Construction, Allowable Working Stress Determination, Flywheel Size for Tractor-Trailor Car and Conventional Shuttle Car, Rocketdyne Failsafe Design Comments, and a Double Disc Concept.

4.4.1 Wheel Construction

Forgings are to be avoided, if possible, because of tooling expense, long delivery cycle, and their potential for non-uniformity and inclusions in the material. Plate stock is ideal in those regards. The most practical plate stock steel available for this application is "HY-Tuf" Alloy Steel by Crucible Steel Company of America. This is aircraft quality steel available as produced under carefully controlled conditions. It is a through-hardening steel, and has very good fatigue life characteristics. The composition of this material is:

94.40%	Iron
0.25%	Carbon
1.80%	Nickel
1.30%	Manganese
0.40%	Molybdenum
1.50%	Silicone
0.35%	Chromium

The analysis of Section 3.2 shows the severe stress penalty due to the existence of a shaft hole in a disc flywheel. To avoid this handicap, the flywheel design is based on a conical contour disc with electron beam welding[1] to attach the shaft elements. The diameter of attachment is kept large in order to be out where the stresses are lower and also to provide maximum stiffness to keep the critical speeds higher. Conventional brazing may be used to join the shaft elements to the inductor motor rotor. The order of assembly would be to first braze the motor assembly, fine balance the three components to be EB welded (rotor assembly, flywheel, and stub shaft), EB weld the flywheel joints, and lastly, turn the complete assembly on centers to finish-grind the bearings and finish-machine other surfaces as necessary. Final balancing can be done by removing stock from the rim since that is the lowest stress area.

4.4.2 Working Stress Determination

The stress in the flywheel is a function of the rotational speed. Consequently, each spin-up cycle constitutes a stress cycle. Over an adequate life of a flywheel package, the total spin-up cycles accumulate to a large number, so it becomes mandatory to design for stresses well below the endurance limit of the flywheel steel.

Figure 32 shows the endurance limit curves for "HY-Tuf." To plot these curves the Crucible Steel test data has been reduced by two factors. A factor of .9 has been applied to account for test samples being polished,

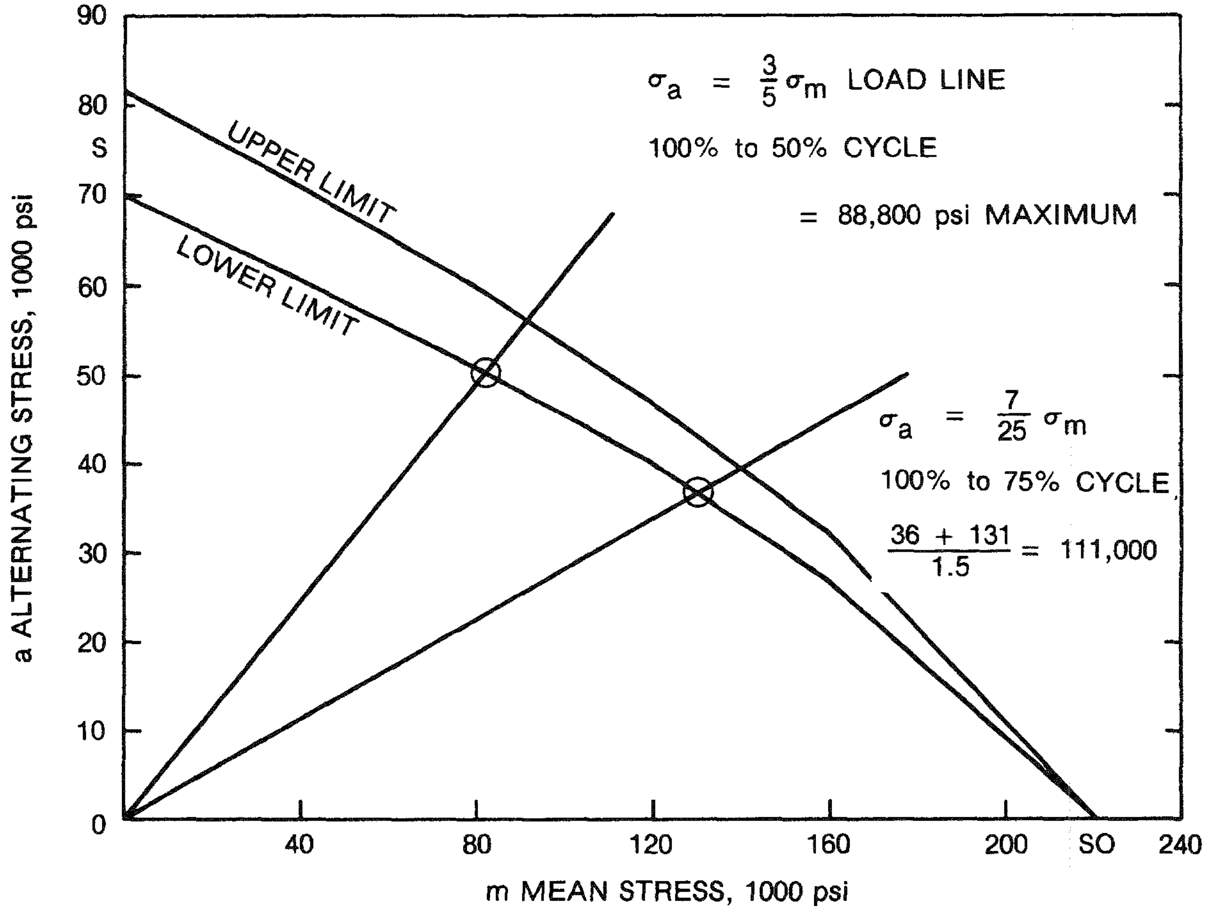


Figure 32. Fatigue Limit for HY-Tuf. Alternating Stress Data Reduced by Factor of .81 per Text

and another .9 factor has been applied to take care of size effects arising from small test samples. Superimposed on the HY-Tuf curves is a load line that represents fly-wheel stress conditions when operating from full speed down to half speed. At half speed the stresses are at 1/4 of maximum; the total stress swing is 3/4 of the maximum; the half-amplitude is 3/8 of the maximum; and the mean stress is (1 - 3/8) = 5/8 of maximum. Thus, the ratio of cyclic stress amplitude to mean stress is 3/8 to 5/8 so $\sigma\alpha = 3/5 \times \sigma\eta$.

The intersection of the lower stress limit curve and the flywheel load line produces an operating point of 50,000 psi alternating stress superimposed on 83,300 psi mean stress, giving a maximum stress at top speed of 133,300 psi. If a safety factor of 1.5 is applied, the maximum working stress is reduced to 88,800 psi. This stress is used for the conceptual design calculations.

A flywheel application in a shuttle car will not be operating consistently down to half speed since it will be recharged at the convenience of the system (Section 5). Fortunately, the load line for a shorter use cycle crosses at a higher allowable stress, so in effect the flywheel will actually be operating with a larger safety factor than 1.5.

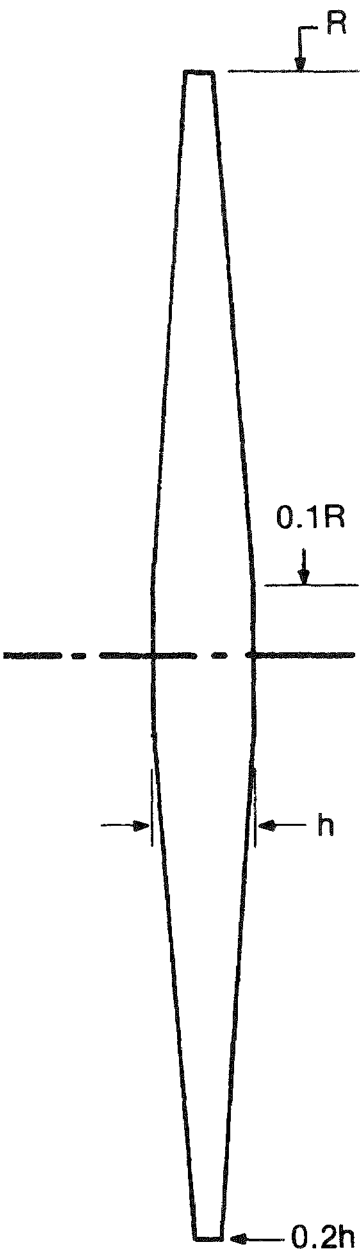
An assumption involved in the use of the working stress derived above is that the failure mode of concern is fatigue due to cycling. Many flywheel applications have overspeed problems due to the nature of the spin-up energy source. With the Load Commutated Inverter the power supply frequency is a direct indication of flywheel speed, so it will be assumed that adequate precautions and redundancy will be provided to assure that the flywheel speed never exceeds 10,000 rpm.

Figure 33 lists the input data and outlines the calculations to arrive at the size of the flywheel. Mission analysis (Section 2) indicates that the usable energy should be 4.5 kW hrs for operation from full speed down to half speed and this requires 6.0 kW hrs total energy storage capability if operated to zero speed.

The calculations actually permit the total flywheel to be built up of one, two, or more conical flywheels depending upon the available sizes for the steel. There is some possibility that multiple discs would moderate the containment problem, but, on the other hand, multiple discs impede the removal of heat from the rotating element (cooling coils on the case covers of the flywheel; see Section 4.5.5, Cooling).

4.4.3 Flywheel Size Determination for Conventional Car

Since there may be economic benefits to using a lower energy flywheel in a conventional shuttle car, a preliminary calculation, Figure 34 was made for an 18 inch diameter cylindrical flywheel for this car. The speed was arbitrarily raised to 15,000 rpm, and the energy capacity was lowered to 3.0 kW hrs. The stress then is low, resulting in a 2.75 safety factor. It is doubtful that there is enough room for spring-mounting the package on the vehicle.



INPUT DATA:

$C_s = .80$

$C_v = .496$

$C_i = .378$

$C_t = 4.19$

$\sigma = 88{,}800$ (S.F. = 1.5) psi

$\rho = .29$ lb/in^3

KE = 4.5/.75 = 6.0 kW hrs (TOTAL)

N = 10,000 rpm (MAX.)

CALCULATIONS: (SEE EQUATIONS FIGURES 7, 8)

$$R^2N^2 = C_t\left(\frac{\sigma}{\rho}\right) 35{,}235$$

R = 21.26 in, D = 42.5 in

$$KE = C_sC_v\,(\pi R^2 h)\,\frac{\epsilon}{31{,}850}$$

h = 3.82 in, .2h = .760 in

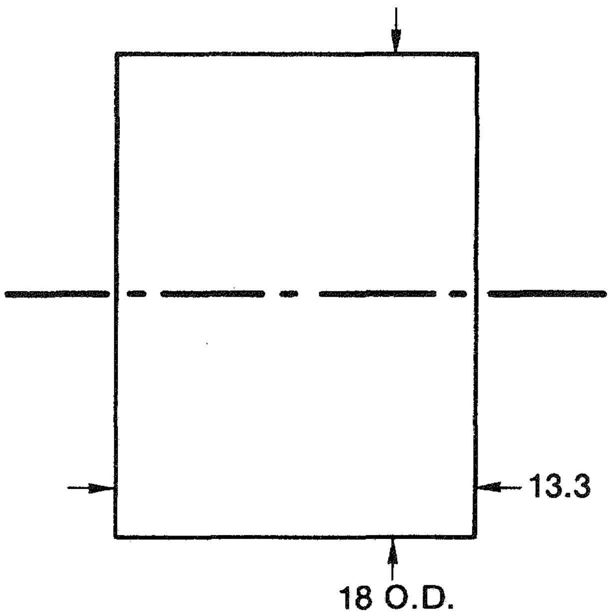
$$W = C_v\,(\pi R^2 h)\rho$$

W = 780 lbs

$$SE = C_s\left(\frac{\sigma}{\rho}\right)\frac{1}{31{,}850}$$

SE = 7.7 W hrs/lb

Figure 33. Size Determination for Conical Steel Flywheel for 4.5 kW hrs Usable Energy



INPUT DATA:

$C_s = .78$

$C_v = 1$

$C_i = 1$

$C_t = 3.11$

$R = 9$ in

$N = 15{,}000$ rpm

$KE = 3/.75 = 4$ kW hrs

$\rho = .29$

CALCULATIONS:

$$R^2N^2 = C_t\left(\frac{\sigma}{\rho}\right) 35{,}235$$

$$= 48{,}300 \text{ psi}$$

$$KE = C_s C_v\ (\pi R^2 h) \frac{\sigma}{31{,}850}$$

$$h = 13.3 \text{ in}$$

$$W = C_v\ (\pi R^2 h)\rho$$

$$W = 980 \text{ lbs}$$

$$SE = C_s\left(\frac{\sigma}{\rho}\right)\frac{1}{31{,}850}$$

$$SE = 4.1 \text{ W hr/lb}$$

Figure 34. Size Determination for Cylindrical Steel Flywheel for 3.0 kW hrs Usable Energy

4.4.4 Rocketdyne Failsafe Flywheel

There is an alternate flywheel design which could be considered in further design studies. Rocketdyne* Division of Rockwell International has developed and patented a "failsafe" flywheel of the type shown in Figure 35, Configurations 0, 7, 10, and 17. The outside rim is stressed to separate first in a failure mode, and this relieves the stresses on the main body of the flywheel so as to avoid its failure. The configuration of the flywheel is not appreciably different from the conical flywheel of Figure 20, so that a change to the failsafe flywheel is a possible design direction.

There are two concerns about the application of the failsafe design. The stresses at the rim (Figure 35) are in effect in the "fuse" that blows before the trouble develops toward the center. Naturally, the fuse should be designed not to blow in normal use, so the fuse stresses will be conservative. To make sure that the fuse blows first, the stresses in the body of the wheel must be considerably lower than in the fuse to separate their respective failure points. It may be just as well to retain the conical geometry but design to an intermediate stress level. This is especially true if overspeed is eliminated as a possible failure mode.

In addition, since fatigue is the mode of failure, the fatigue properties of the fuse must be equal to those of the main body of the flywheel. Thus, extreme care must be taken to insure uniformity of the basic material and, in machining and heat treating, to preserve uniformity of the endurance limits.

4.4.5 Early Concept of Cylindrical Flywheel

Figure 36 illustrates an early concept of a cylindrical flywheel shown with an inductor machine sized for 4.5 kW hrs charge in 30 seconds. The flywheel is composed of discs cut from plate stock. By shrink fitting the discs to the shaft it is possible to compensate for part of the 2 to 1 stress penalty due to the shaft hole (Section 3). It is also theoretically possible to prespin the discs beyond their yield stress to set up compressive stresses at the shaft hole and thus further help the stress picture. The cylindrical approach was abandoned in favor of the conical flywheel without a shaft hole for the tractor-trailer application because of the saving in length, made possible by the large diameter available. In the case of the conventional shuttle car, electron beam welding makes this concept of multiple discs valid without a shaft hole.

*Proceedings of the 1975 Flywheel Technology Symposium, p. 117-122.

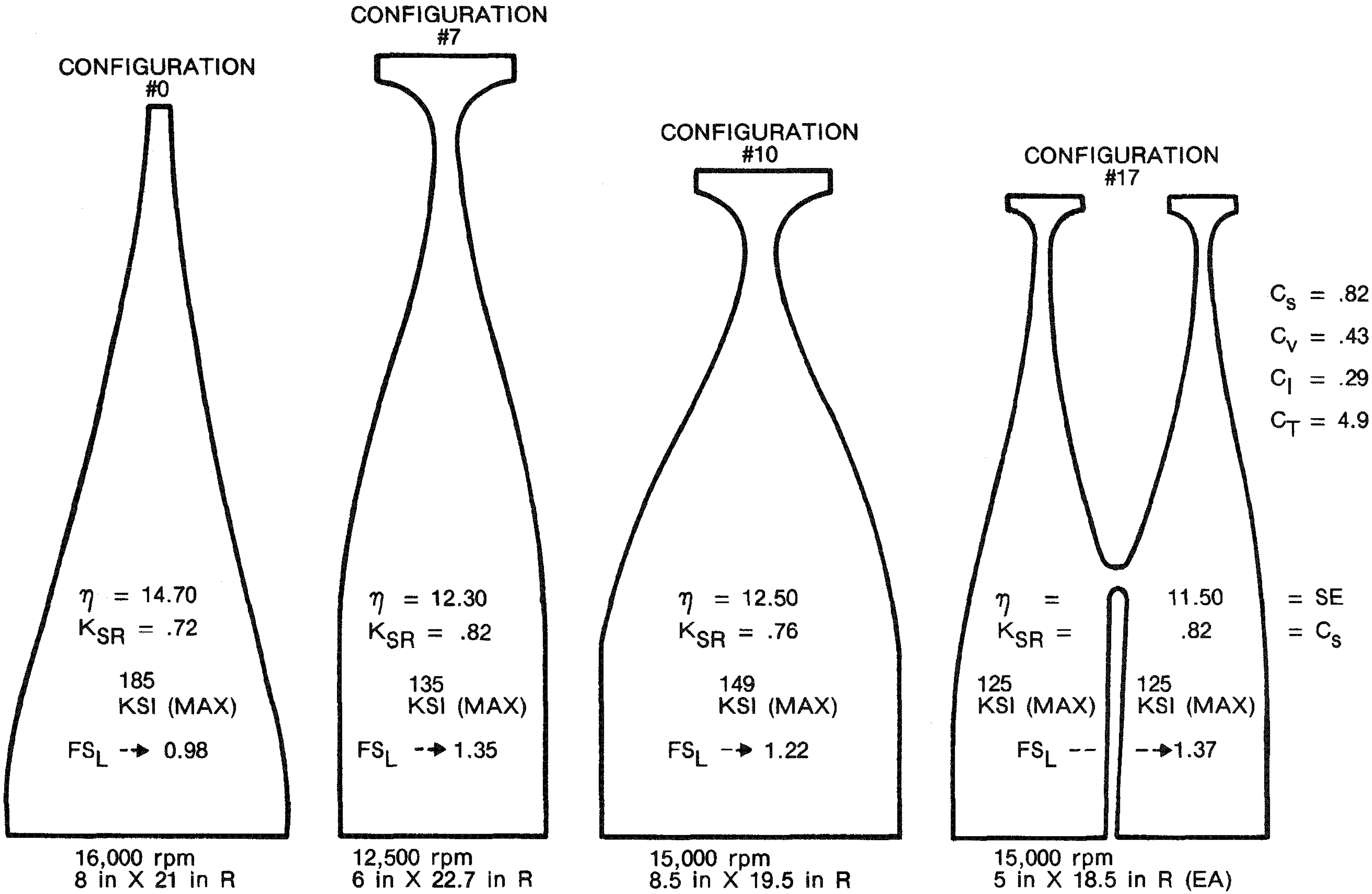


Figure 35. Rocketdyne Family of Flywheels from the Proceedings of the 1975 Flywheel Technology Symposium, Berkeley

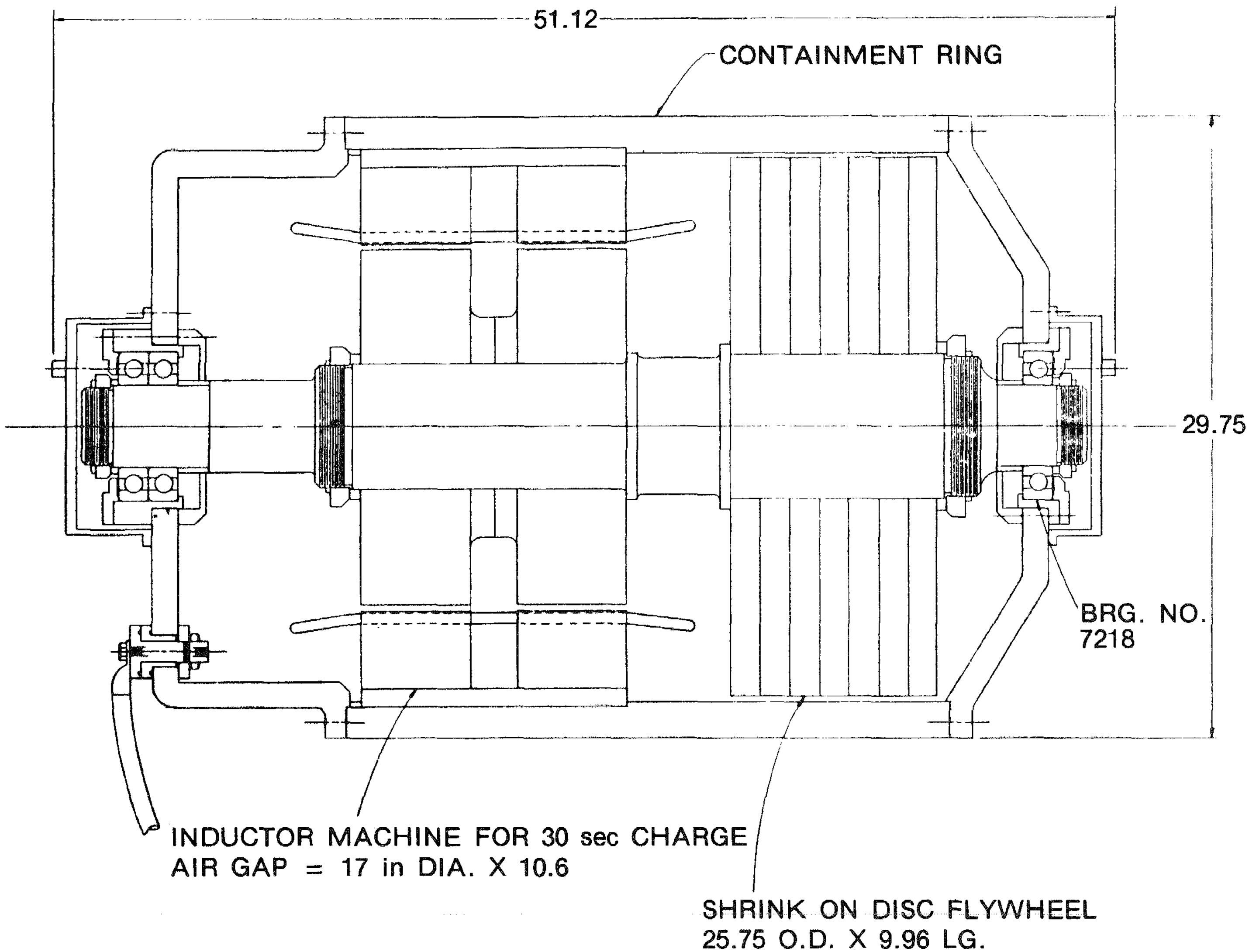


Figure 36. Flywheel Inductor Machine

4.4.6 Six kW hr Double Disc Concept

For one iteration of the conceptual design process, it was assumed that flywheel plate stock would only be available in thin sizes which would necessitate using two identical conical flywheels. Drawings for this version are presented in Figure 37 and Figure 38. The flywheels shown are 3-inch thick finished dimension, and their total usable capacity is 6 kW hrs. Everything else is the same as the baseline design shown in the overall flywheel package description Figure 20 and Figure 21. If it should subsequently appear desirable to provide extended tram distance capability (greater than 500 feet), this multiple disc concept could be pursued. The design penalty for multiple discs is increased flywheel package height which impacts on visibility of the operator (Appendix D). This double disc 6.0 kW hr design is 3-3/4 inches higher than the single disc 4.5 kW hr design. The motor is unchanged, and therefore the charge time for extended tram distances is lengthened. Section 5.4.6 discusses this further.

4.4.7 Design Areas for Further Work

There are several significant design areas to be investigated in depth for the design of a prototype flywheel.

- Flywheel stress calculations by means of the equations shown here are simplistic. Experience has shown that the computerized finite element analysis program provides information much closer to the real world. This would be particularly true for stresses in the area of the welds.

- The finite element stress program must incorporate provisions to check the wheel stresses due to vertical acceleration loads transmitted to the flat disc from the suspension system of the whole flywheel package.

- EB welding in this size is not a "shelf item," so it will take considerable effort to locate facilities and define the processes, or prove the weld schedule, using pull test samples and other techniques.

- Critical speeds are a function of both the rotating element mass distribution and the stiffness of the bearing supports. Computer programs are available to identify criticals, and extensive design effort is required on the details of the shaft elements to achieve a satisfactory combination.

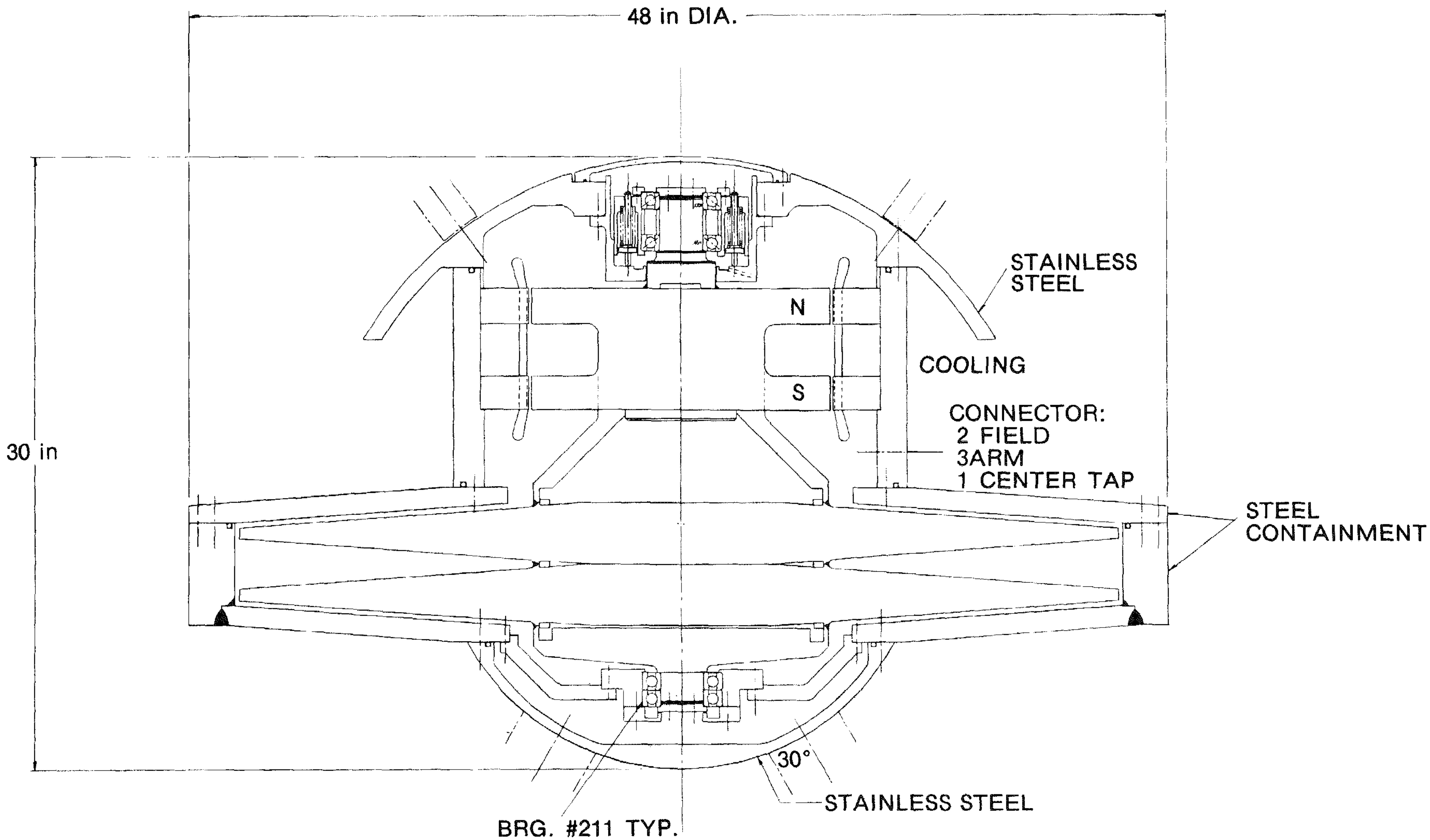


Figure 37. Flywheel Capsule (6 kW hr Flywheel)

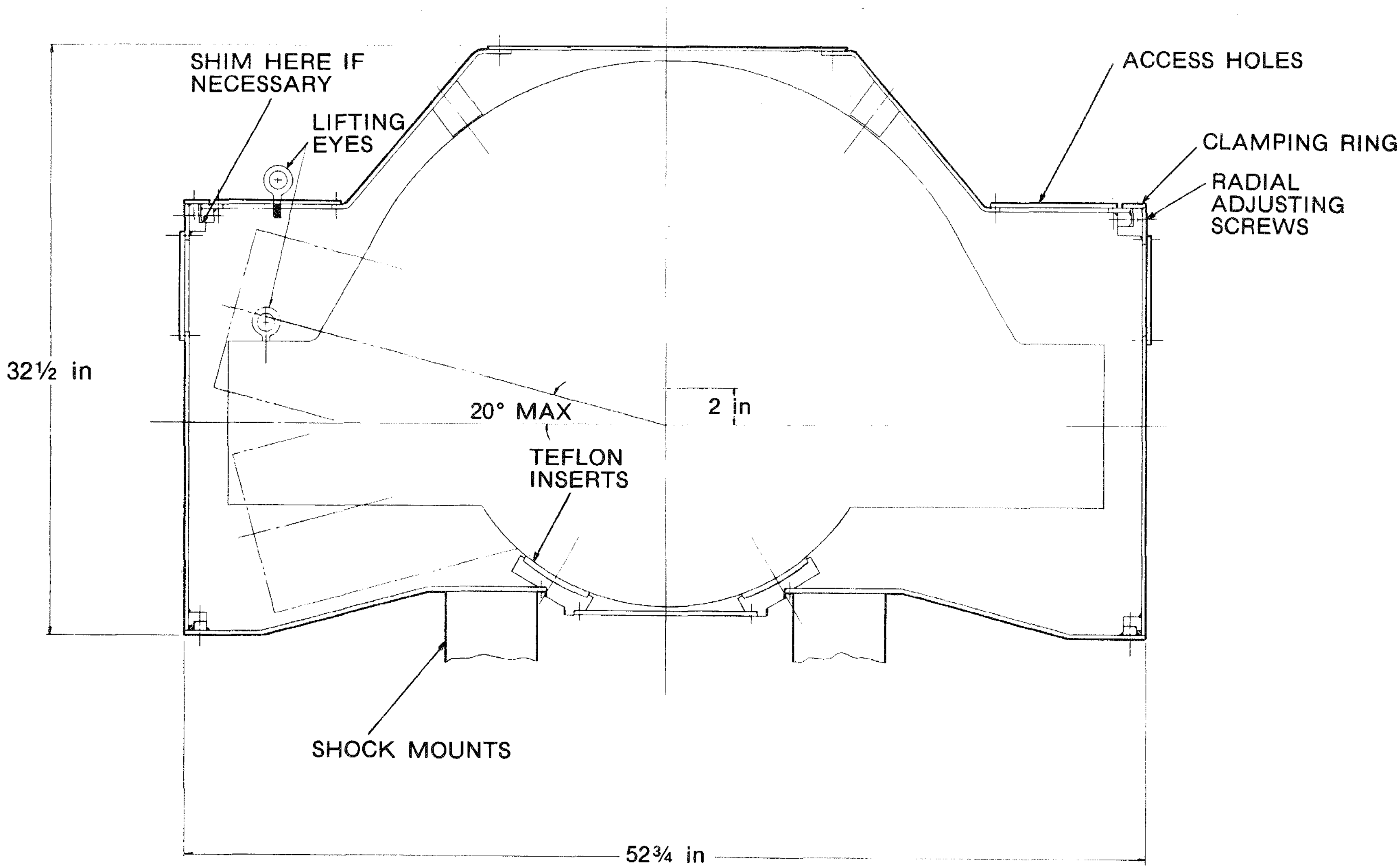


Figure 38. Flywheel Package (6 kW hr)

- Since the only exit for removing heat from the rotating element is through the flywheel out to the top and bottom covers, the conical shaft element mating the motor and the flywheel may be required to have good heat conduction into the flywheel. This impacts on stiffness and on local stresses at the weld and is another requirement in the detailed design.

4.5 DISCUSSION OF FLYWHEEL CAPSULE DESIGN FEATURES

Several features of the design of the flywheel capsule have been considered to an extent consistent with the conceptual design illustrated previously in Figure 20.

These features include bearings and lubrication, containment, and flywheel windage losses, vacuum, and cooling.

4.5.1 Bearings and Lubrication

Two approaches are discussed. In the first, the bearings are assumed to be "duplex pairs" with the static load equally divided between the top and bottom pairs. Duplex pairs are set up so that the load is divided equally between the two bearings in a pair. The second case assumes it is possible to shim the two bearings of a duplex pair so that the load division is changed to 90% and 10%. The 90% loaded bearing would be the lower of the two and will fail first. Adequate isolation (shielding) would protect the 10% bearing in the event of failure of the 90% bearing. Thus, the 10% bearing provides essentially full redundancy. Failure of the first bearing should audibly manifest trouble in time to avoid a catastrophic failure. The price for this redundancy is a net shortening of the minimum life of the spin axis bearings before failure. The rationale of the 90%/10% approach is the basis upon which the 7211 bearings were selected, and these are depicted in Figure 20.

Duplex Pair Bearing Calculations

The choice of bearings is a complex design problem. The flywheel will be operating with its axis vertical to avoid generating gyroscopic moments in maneuvering the shuttle car. Thus, the bearing loads are mainly thrust loads. This is fortunate for ball bearings since all the balls will be loaded and scuffing will be minimized. At the high operating speeds of the flywheel it will be necessary to use special high-speed bearings - "super precision angular contact ball bearings."

Further, it is anticipated that a duplex pair of bearings be used at the top and at the bottom of the flywheel, and that the case structure be provided with means to divide the load equally between the two pairs. This allows the bearing sizes to be small to minimize the bearing losses. However, decreasing the bearing size decreases the working life of the bearing. Duplex bearings thus reduce the bearing loads and extend the life. The redundancy of two bearings also eliminates the need of shaft space required by an emergency rub collar in the event of a bearing failure.

Bearing life calculations are outlined in Figure 39 based on SKF information. On the assumption of zero radial load, the thrust load is used as the equivalent load. The "Basic Dynamic Load Rating" is tabulated in the catalog for each bearing size and is given for a duplex pair incorporating a safety factor for load division between the two bearings in a pair.

Changing Equation (1) to result in years of operation instead of revolutions produces Equation (2). Equation (2) is obtained using 5000 hours of operation per year and an effective speed of 9000 rpm. Both of these numbers are conservative for the shuttle car application.

Bearing thrust load is half of the total of 780 lbs for the flywheel; 242 lbs for the motor rotor; and 78 lbs for the shafts. This gives P = 550#. SKF also applies their bearings under "favorable lubrication" expecting to get a minimum life of six times the calculated value obtained from Equation (1) which is the industry standard. Putting these two values into Equation (2) yields a working relationship (Equation (3)) between bearing capacity and bearing life.

Figure 40 presents the calculated minimum life for bearings having a maximum operating speed of 10,000 rpm or above. Using this figure as a basis, the 7207C bearings are selected due to their minimum life of eight years. The 7207C bearing was selected over the 7208C bearing in deference to bearing losses.

90%/10% Bearing Pair Calculations

For this arrangement, 90% of the rotating element weight is used in Equation (3) giving:

$$(4) \quad \gamma 10 = \left(\frac{C}{13.9 \times 550 \times .9}\right)^3 \times 6 = \left(\frac{C}{8132}\right)^3$$

(1) $L_{10} = \left(\frac{C}{P}\right)^3 \times 10^6$

L_{10} = MIN. LIFE IN REV. (10% FAILURES)

C = BASIC DYNAMIC LOAD RATING

P = EQUIVALENT LOAD = THRUST LOAD

(2) $Y_{10} = \left(\frac{C}{13.9 \times P}\right)^3$

Y_{10} = MIN. LIFE IN YEARS (10% FAILURE)

(3) $Y_{10} = \left(\frac{C}{13.9 \times 550}\right)^3 \times 6 = \left(\frac{C}{4200}\right)^3$

550 = HALF OF ROTATING ELEMENT WEIGHT

6 = SKF LIFE FACTOR

Figure 39. Bearing Life Calculations

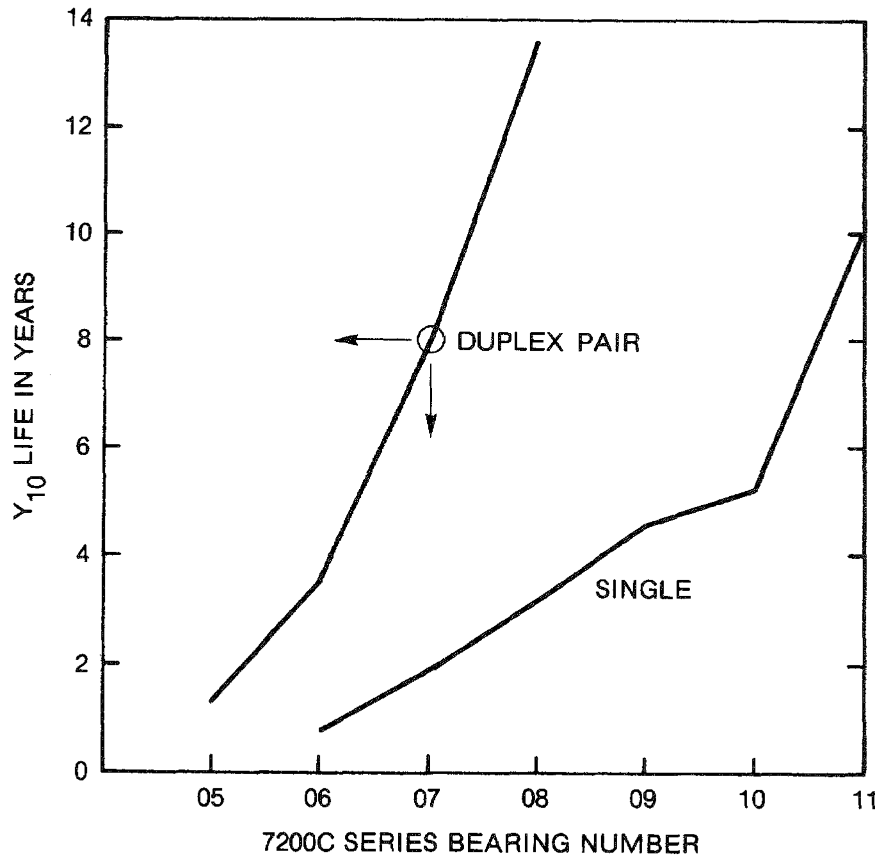


Figure 40. Minimum Life of Bearing Versus Bearing Number (9,000 rpm, 5,000 hrs/ year, 550 lb)

From this relationship a new curve of minimum life versus bearing size is plotted as Figure 41. The size 7211 bearings have a minimum life of two years and there are two per shaft assembly.

Top Bearing Assembly

The bearing calculations are based upon equal load division between the top and bottom pairs. The outside case of the flywheel capsule is not rigid, and its deflections are affected by the internal pressure which varies from one atmosphere to 0.003 atmosphere. Hence, it is necessary to resort to springs to control the load applied to the top bearing pair. This means that all shock loading will be applied only to the bottom pair so that ultimately the load division might be chosen to favor the bottom bearing with a load split of 40 - 60.

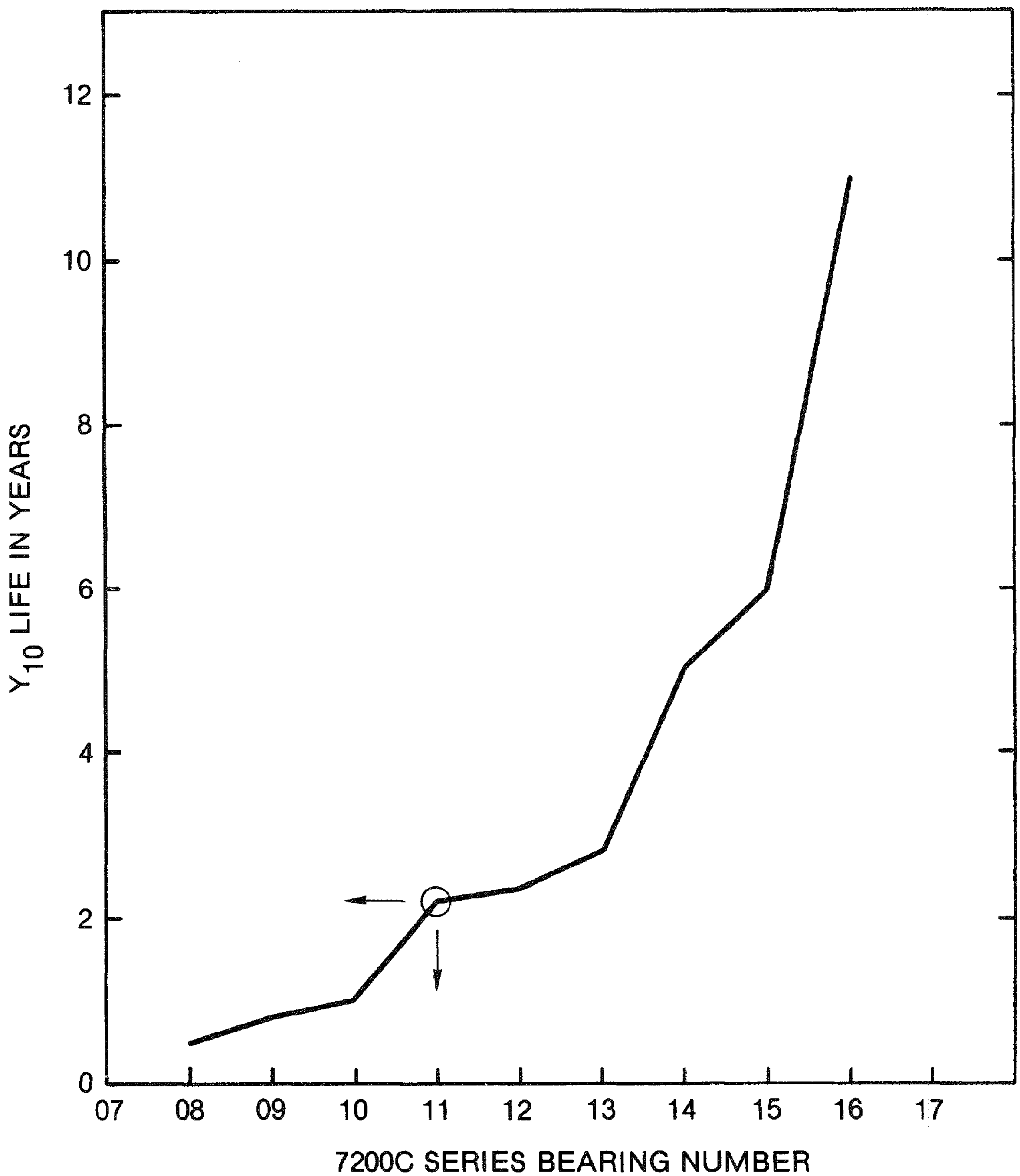


Figure 41. Minimum Life of Bearing Versus Bearing Number for 90%/10% Load Division Between Adjacent Pair

Next, the top bearing assembly (see Figure 20) was expressly arranged so that no cocking action can occur and so that the outer races cannot creep in service. This assembly would be mounted to the shaft after balancing the flywheel and thereafter remain an integral part of the flywheel-motor assembly. Not having to touch the bearings each time the flywheel capsule is disassembled should serve to maintain good bearing reliability. The bottom bearings also are to be considered a part of the flywheel-motor assembly and are not to be removed for capsule disassembly.

Lubrication

"Favorable lubrication" is defined by SKF by a curve of minimum oil viscosity at operating temperature versus the product of rpm times ball pitch diameter. For the 7207C bearings this means a viscosity above 50 SSU which is equivalent to #10 SAE oil at 180°F.

The quantity of oil delivered is critical. Too much oil results in significant heat generation and attendant slowing of the flywheel. The oil must be delivered in the quantities used in "mist lubrication" of high speed spindles. Oil is transported by air in mist lubrication systems, but the flywheel bearings operate in a vacuum which eliminates air as a carrier. Therefore, the oil must then be broken up into fine particles by a mechanical means, probably an impellent located adjacent to each bearing. This necessitates the oil being dispensed by drops to each bearing and 14.7 psia is available to deliver the oil.

There is a project sponsored by DOE/US Postal Services* to evaluate a flywheel application to an electric jeep delivery vehicle. Evaluation data from this project should be directly applicable to the lubrication problem of the shuttle car flywheel capsule. The postal services vehicle has been inspected and can be used as a base point for the mine vehicle flywheel package design of vacuum and lubrication systems.

*Garret AiResearch Manufacturing Company, Torrence, California.

4.5.2 Containment

A General Electric report is quoted[2]:

"Any analysis of the effects of a flywheel burst on a containment housing must be considered as an estimate since many factors (such as the number of fragments produced, their energies, the amount of energy dissipated in frictional heating and plastic deformation, the exact shape of the fragments as they impact the housing, etc.) cannot be evaluated. However, reasonable and conservative estimates can be made using proper assumptions and results of previous experimental and analytical work.

"Major points in the analysis of a flywheel burst are:

- It is assumed that no energy is required to form the initial fracture surfaces. Thus, assuming that the wheel bursts into "n" equal fragments, each possesses an initial kinetic energy (both translational and rotational) equal to the energy of the flywheel at time of burst divided by the number of fragments.
- The fragments leave the wheel with both tangential and radial components of velocity, and hence strike the housing at an oblique angle. The penetrating capability of the fragment is dependent on the radial component of velocity at the time of collision with the housing; tangential components result in energy dissipation due to friction and gouging of the surfaces. The radial component of velocity can be minimized by making the clearance between the wheel and housing small (Reference 3). If the clearance is very small, only one-third of the total energy of the fragment is associated with the radial velocity component, while if the clearance is 10 percent of the wheel radius, almost one-half of the total energy of the fragment must be dissipated by radially directed forces in the housing (Reference 4).

- Two models (Stanford and Ballistics Research Lab) of missile impacts and penetration have been presented in the literature (Reference 2). These are empirical and must be used with some caution when the geometrical and dynamic parameters lie outside the ranges tested (as they do in the case of the flywheel); nevertheless, they give some idea of the thicknesses of housing to contain a fragment. According to these models, this thickness (in inches) is given by:

  $$t = -\ .0118L + (.000138L^2 + \frac{2.9E}{DU})^{1/2} \qquad \text{Stanford}$$

  $$t = \left(\frac{E}{17400K^2D^{3/2}}\right)^{2/3} \qquad \text{Ballistics Research Laboratory}$$

  where E (ft/lb) is the energy associated with radial velocity at impact, D (in) is an effective diameter of the fragment, U (lb/in^2) is the ultimate strength of the housing, L (in) is a dimension which can be taken as the wheel length and K is a material constant which is about unity for steel."

Assuming a wheel fragments into thirds and with 1/3 of the energy of each fragment associated with radial velocity, the value for E will be 1/9 times the total kW hrs of the flywheel or 1.77×10^6 ft-lb. For the effective diameter (D) of a fragment, the wheel radius of 21.5 inches is a reasonable assumption. Assuming the containment ring to be heat-treated steel with an ultimate of 200,000 psi, the thickness from the above equations is:

Stanford	1.048 inch
BRL	1.013 inch

Since weight and space are not critical, a 2-inch thick ring weighing 330 lbs will provide a safety factor of 2 to 1.

The side plates of the enclosure that are located adjacent to the flywheel are purposely made heavy so they participate in the matter of containment. If the containment ring was made of two rings, each welded to a side plate and heavy bolting used to join the two halves, it should be possible to handle the containment without special heat treatment steel, etc.

In addition to the area of the containment ring, there is a second problem in the event of a disc rupture. Much of the disc momentum is transferred to the containment ring and capsule, and this resulting rotation must be handled. The simplest solution is to make the "Enclosure" to meet mine safety regulations for electric arcs, etc. and strengthen the structure adequately for keeping the capsule trapped at the spherical surfaces. It is not expected that a brake band around the capsule (at the containment ring) will be necessary because of the enclosure around the capsule.

4.5.3 Flywheel Windage Losses

This discussion presents preliminary calculations of the windage losses of the flywheel. The gas around the flywheel is assumed to be air although in a matter of hours after pumping the pressure down, the air will probably be a mixture of hydrocarbons derived from the lubricating oil.

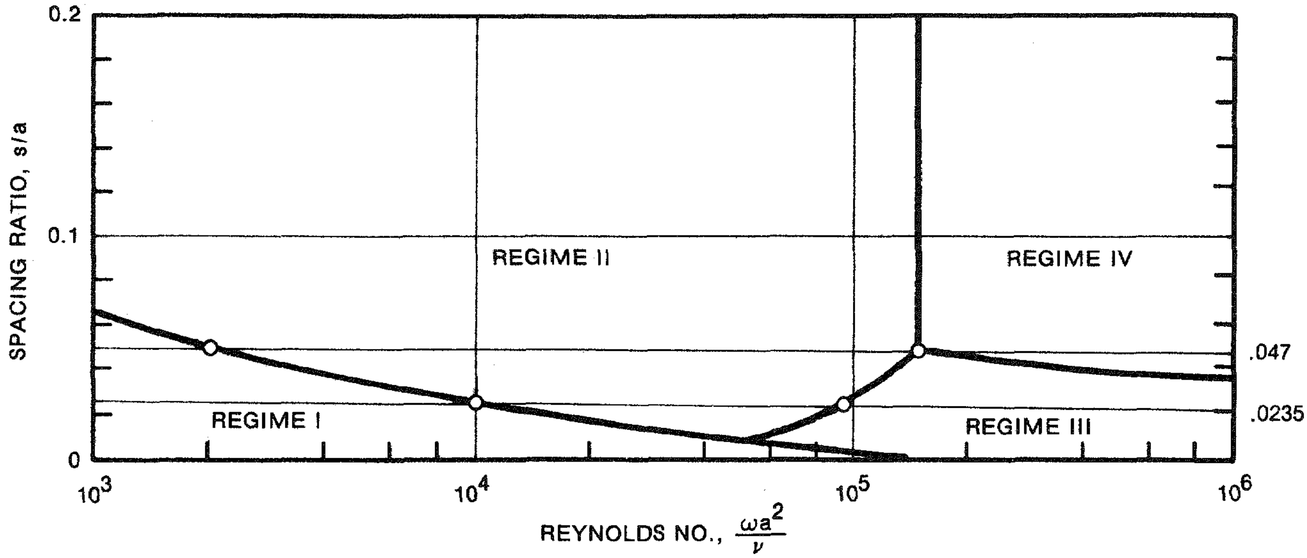
Figure 44 presents the losses based on a 125°C temperature, which is selected on the high side to highlight the problem of cooling the flywheel-motor assembly. To keep the rotating losses below 1000 watts it will be necessary to keep the pressure below .04 psia. It is not expected that this will be difficult. Calculations show that at these pressures the flywheel will be operating entirely within the flow Regimes I and II of Figure 7.5-4, which is in the low loss end of the spectrum.

Flywheel Windage Loss Calculation

Windage losses are calculated in terms of the drag torque M_1:

(1) $$M_1 = C_{m1} \times \frac{1}{2} \times \frac{\rho 2}{g} \times \omega^2 R^5, \text{ in ft/lbs}$$

It is seen that the torque is a function of the radius (R), rotational speed (ω), air density (ρ), and torque coefficient (C_{m1}). The torque coefficient in turn is a function of an empirically derived formula which includes a Reynolds number (N_{Re}), the air viscosity (ν), and the spacing (S), between the sides of the flywheel and the case. There are four distinct regimes of gas behavior, and for each regime there is a separate equation for the relationships of the terms within the torque coefficient C_{m1}. These regimes are identified in Figure 42. It is probably most convenient to proceed through the calculations in logical order, starting with the basic variables.



For the "free flow" disks rotating near a solid surface the mode of flow in the fluid between the disk and stator depends upon the axial distance ratio, s/R, the presence of four different flow regimes has been experimentally verified. These are:

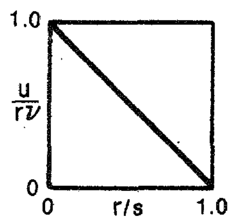


Figure 1

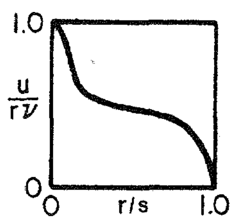


Figure 2

REGIME I. Laminar Flow, Close Clearance:
Boundary layers on the rotor and stator are merged. A relatively constant gradient in the tangential velocity across the axial gap, s, exists. (Figure 1)

REGIME II. Laminar Flow, Separate Boundary Layers:
The combined thickness of the boundary layers on the rotor and stator is less than the axial gap, s. Between the boundary layers is a core region in which relatively little or no change in the tangential velocity is expected to occur. (Figure 2)

REGIME III. Turbulent Flow, Close Clearance:
The turbulent counter part of Regime I where the flow in the boundary layers is turbulent (occurs at Reynolds number higher than Regime I)

REGIME IV. Turbulent Flow, Separate Boundary Layers:
The turbulent counter part of Regime II.

Figure 42. Operating Flow Regimes for Rotating Disks

Air Density - Air density at 32°F and 14.7 psi is given* as

$$\rho 1 = 0.0807 \text{ lb/ft}^3$$

The air density ($\rho 2$) in the capsule is calculated as follows:

(2) $$\rho 2 = \frac{V_1}{V_2} \rho 1$$

(3) also $$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \quad \text{or} \quad \frac{V_1}{V_2} = \frac{P_2}{P_1} \times \frac{T_1}{T_2}$$

therefore: $$\rho_2 = \rho_1 \left(\frac{P_2}{P_1} \times \frac{T_1}{T_2} \right)$$

and substituting known values:

(4) $$\rho_2 = 0.0807 \left(\frac{P_2}{14.7} \times \frac{273°}{273° + T_2} \right)$$

$$\rho_2 = 1.50 \left(\frac{P_2}{273 + T_2} \right)$$

where P_2 is the capsule pressure in psia and T_2 is the capsule temperature in °C

Air Viscosity

Air viscosity is given in terms of dynamic viscosity in tabulated form which is plotted as Figure 43. The local area of interest can be represented by the following equation that extrapolates between points:

(5) $\mu = (0.344\ T_2 + 185) \times 10^{-6}$ poise*
and this must be converted to kinematic viscosity:

(6) $$\nu = \frac{\mu \times g}{478.2 \times \rho_2}, \text{ in ft}^2\text{/sec}$$

*Robert C. Weast, Editor, Handbook of Chemistry and Physics, CRC Press, The Chemical Rubber Company, Cleveland, Ohio.

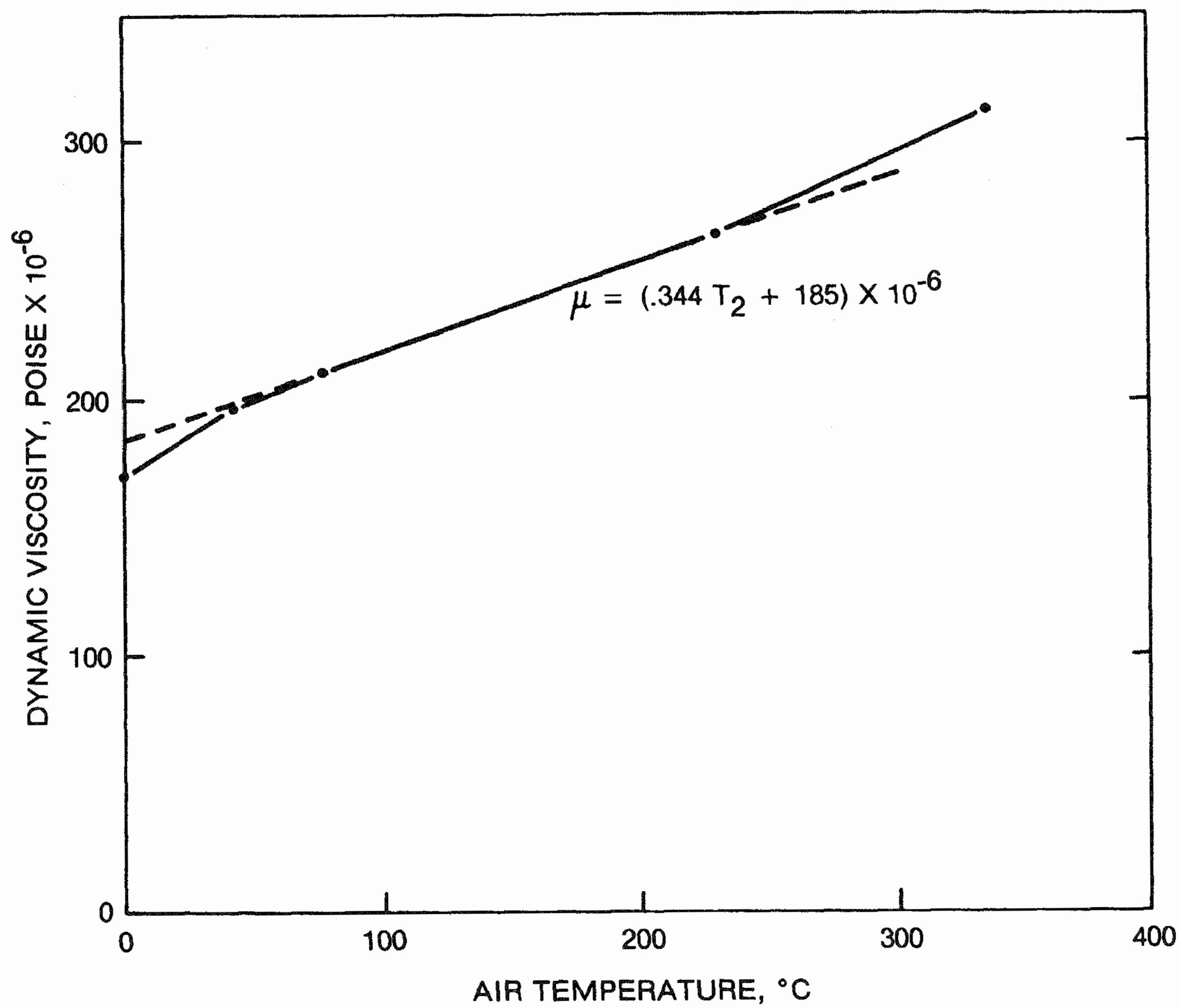


Figure 43. Air Viscosity (data from _Handbook of Chemistry and Physics_, CRC Press, Cleveland, Ohio)

$$(7) \quad \nu = \frac{\mu \times 32.17}{478.2 \times \rho_2} = 0.0672 \ \frac{\mu}{\rho_2}$$

<u>Reynolds Number</u>

The Reynolds number applicable to this case is defined as:

$$(8) \quad N_{Re} = \frac{\omega R^2}{\nu}$$

where: $\omega = \frac{2\pi \times 10{,}000 \text{ rpm}}{60} = 1047$ radians/sec

$R = 21.26$ in $= 1.77$ ft

ν = kinematic viscosity in ft^2/sec

and substituting:

$$(9) \quad N_{Re} = \frac{1047 \times 1.77^2}{\nu} = \frac{3280}{\nu}$$

For Regime I:

$$C_{m1} = \frac{\pi}{(S/R)\ N_{Re}} \ *$$

and substituting Equation 9 for N_{Re}:

$$(10) \quad C_{m1} = \frac{21.26 \times \pi \times \nu}{S \times 3{,}280} = 0.0204 \ \frac{\nu}{S}$$

For Regime II:

$$C_{m1} = \frac{1.85\ (S/R)^{0.1}}{(N_{Re})^{0.5}} \ *$$

Again substituting Equation 9 for N_{Re}:

$$(11) \quad C_{m1} = \frac{1.85 \times S^{0.1} \times \nu^{0.5}}{(21.26)^{0.1} \times (3{,}280)^{0.5}} = 0.0238\ S^{0.1}\nu^{0.5}$$

Now it is possible to return to Equation (1) and insert all of the necessary values to build up the relationship between pressure and windage loss in watts:

$$(1) \quad M_1 = C_{m1} \times \frac{1}{2} \times \frac{\rho_2}{g} \times \omega^2 R^5, \text{ in ft/lbs for one side of the disc.}$$

*From General Electric proprietary Data Books (EMPIS)

(12) $M_1 = C_{m1}\rho_2 \times \frac{1}{2} \times \frac{1}{32.17} \times (1,047)^2 \times (1.77)^5$

$= 2.96 \times 10^5\ C_{m1}\rho_2$

The amount of energy loss in watts is given by the formula:

(13) $E = 746 \times \frac{2\pi N}{33,000} \times 2\ M_1$

and substituting N = 10,000 rpm and Equation (12)

(14) $E = 746 \times \frac{2\pi \times 10,000}{33,000} \times 2 \times 2.96 \times 10^5$

$= 8.41 \times 10^8\ C_{m1}\rho_2$, in watts.

Assuming the air temperature in the case is 125°C and substituting in the earlier formulas:

(4) $\rho_2 = \frac{1.5\ P_2}{(273 + 125)} = 3.77 \times 10^{-3}\ P_2$

(5) $\mu = (0.344 \times 125 + 185) \times 10^{-6}$

$= 228 \times 10^{-6}$ poise

(7) $\nu = 0.0672\ \frac{N}{\rho_2} = \frac{0.0672 \times 228 \times 10^{-6}}{3.77 \times 10^{-3}\ P_2}$

$= \frac{4.064}{P_2} \times 10^{-3}\ ft^2/sec$

(9) $N_{Re} = \frac{3280}{\nu} = \frac{3280\ P_2}{4.06 \times 10^{-3}} = 8.07 \times 10^5\ P_2$

And finally for Regime I:

(10) $C_{m1} = 0.0204\ \frac{\nu}{S} = \frac{0.0204 \times 4.064 \times 10^{-3}}{S\ P_2}$

$= \frac{8.29 \times 10^{-5}}{S\ P_2}$

(14) $E_I = 8.41 \times 10^8 \; C_{m1}\rho_2$

$$= \frac{8.41 \times 10^8 \times 8.29 \times 10^{-5} \times 3.77 \times 10^{-3} \; P_2}{S \; P_2}$$

$$= \frac{263}{S} \text{ watts}$$

For Regime II:

(11) $C_{m1} = 0.0238 \times S^{0.1} \times \nu^{0.5}$

$$= 0.0238 \times S^{0.1} \left(\frac{4.064 \times 10^{-3}}{P_2}\right)^{0.5}$$

$$= \frac{1.52 \times 10^{-3} \; S^{0.1}}{P_2^{\;0.5}}$$

(14) $E_{II} = 8.41 \times 10^8 \; C_{m1}\rho_2$

$$= \frac{8.42 \times 10^8 \times 1.52 \times 10^{-3} \times 3.77 \times 10^{-3} \; P_2 S^{0.1}}{P_2^{\;0.5}}$$

$$= 4825 \; S^{0.1} \; P^{0.5} \text{ watts}$$

These two loss curves are plotted on the graph Figure 44 for the two values of case clearance S of 1/2 inch and 1 inch.

Regime Transitions

Equations (14) above define the losses in Regines I and II, but the true transition point between the two curves is not at the intersection of these two curves. The transitions are controlled by the Reynolds Number and the clearance distance S. Assuming two clearance values of 1/2 inch and 1 inch allows the determination of the Reynolds Number at the transitions by picking the values from the curves Figure 42. The transition points for these conditions are shown in Table XX below.

Table XX

Regime Transition Points

S	S/R	N_{Re} REGIME I TO II	N_{Re} REGIME II TO III	N_{Re} REGIME III TO IV
0.5	0.0235	1×10^4	1×10^5	-
1.0	0.0470	2×10^3	1.5×10^5	1.5×10^5

The calculations of flywheel windage losses provide a bogey of .04 psia for the pressure in the capsule. This is no problem for a vacuum pump, and since the capsule is completely sealed, the pump capacity need not be large.

The heat transfer calculations indicate that most of the flywheel heat is transferred by radiation to the case. So, if the pressure is drawn down below .04 psia, the lower pressure decreases the conduction component, but also decreases the heat generated. It does not appear that there needs to be a closed loop control to run the vacuum pump. Very likely, pumping down once a shift will be adequate, or at most it may be necessary to run the vacuum pump during each spin-up.

4.5.4 Vacuum System

To the curves of Figure 44 is also added a plot of Equation 9 which relates Reynolds Number to case pressure. With this it is possible to enter the ordinate with the transition values of Reynolds Number from Table XV and project up to the loss curves to establish the transition points.

4.5.5 Cooling

There are three places that heat is generated in the flywheel capsule: (1) in the inductor motor stator, (2) in the inductor motor rotor, and (3) windage losses in the air gap between the flywheel and the case. To sum up the situation, the stator will be quite hot and will present a cooling problem itself. Windage heat generated by the flywheel must be taken out by cooling the case surrounding the flywheel. Since the motor stator will probably be hotter than the rotor, the only path for motor rotor heat must be out through the flywheel to the case.

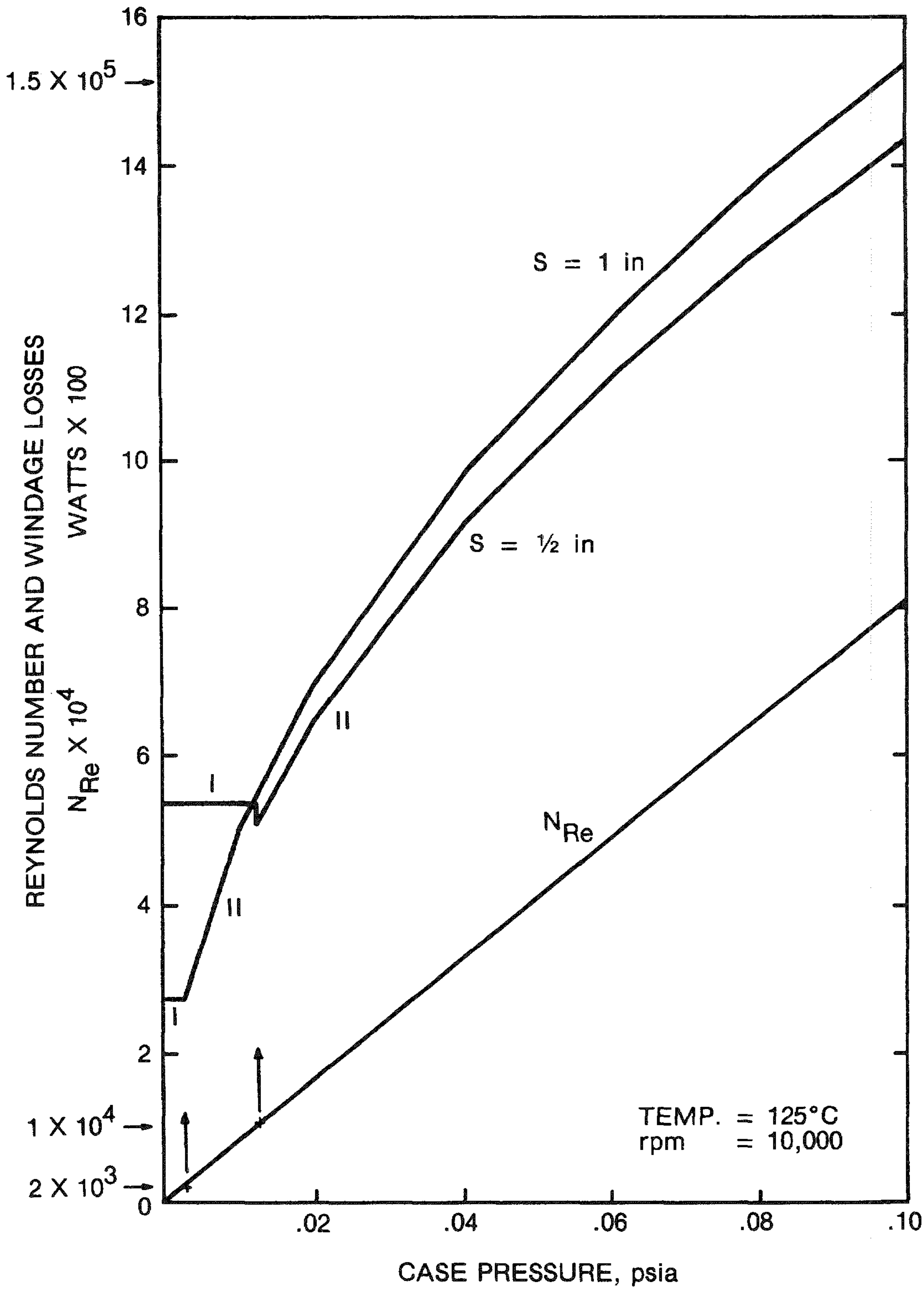


Figure 44. Reynolds Number and Windage Losses Versus Case Pressure

A serious problem may arise in improving the heat conduction between the rotor and the flywheel without affecting the localized stresses at the EB weld joint.

The solid state electronic industry is totally dependent upon silicone gel to enhance heat conduction and this may be resorted to in the cavity between the motor rotor stub shaft and the flywheel. An aluminum plug could be used to fill the space inside the transition piece, and silicone gel can be used to eliminate air gaps. The aluminum plug must be anchored firmly.

Figure 45 is an adaptation of Figure 21 to show coolant tubes to cool the stator and the flywheel.

4.5.6 Heat Transfer - Flywheel and Motor Rotor

Flywheel windage losses have been calculated (see Section 41) to range between 650 watts at 0.02 psia to 1500 watts at 0.1 psia. In addition, it may be assumed that some 10 percent of the motor/alternator losses will be generated in the rotor due to eddy current and hysteresis effects. The rotor losses will amount to something like 750 watts. The question arises as to what level of temperatures might be expected to develop in the radiation and convection of these energies from the motor rotor and flywheel to the containment.

To model this case it will be assumed that the motor rotor is a right cylinder 12.82 inches in diameter by 7.65 inches long (203 kW machine) and that the flywheel is a thin disc 42.52 inches in diameter. It will further be assumed that only the outer one-third of the flywheel is effective in convective heat transfer (see Figure 46) and that only the outer one-third of the flywheel facing the rotor is effective in radiant energy transfer while the entire opposite side is effective.

The basic equations for heat transfer are:

Radiant Energy Transfer

$$q_r = 1.47 \times 10^{-10}\ \varepsilon\ T_a^{\ 3}\ \Delta T \eta A_r, \text{ in watts}$$

where: ε = emissivity (assumed = 0.9)

T_a = average temperature of flywheel and containment in °K

ΔT = temperature differential in °C

η = effectiveness factor (assumed = 1)

A_r = radiation area in sq. in = 2209 in^2

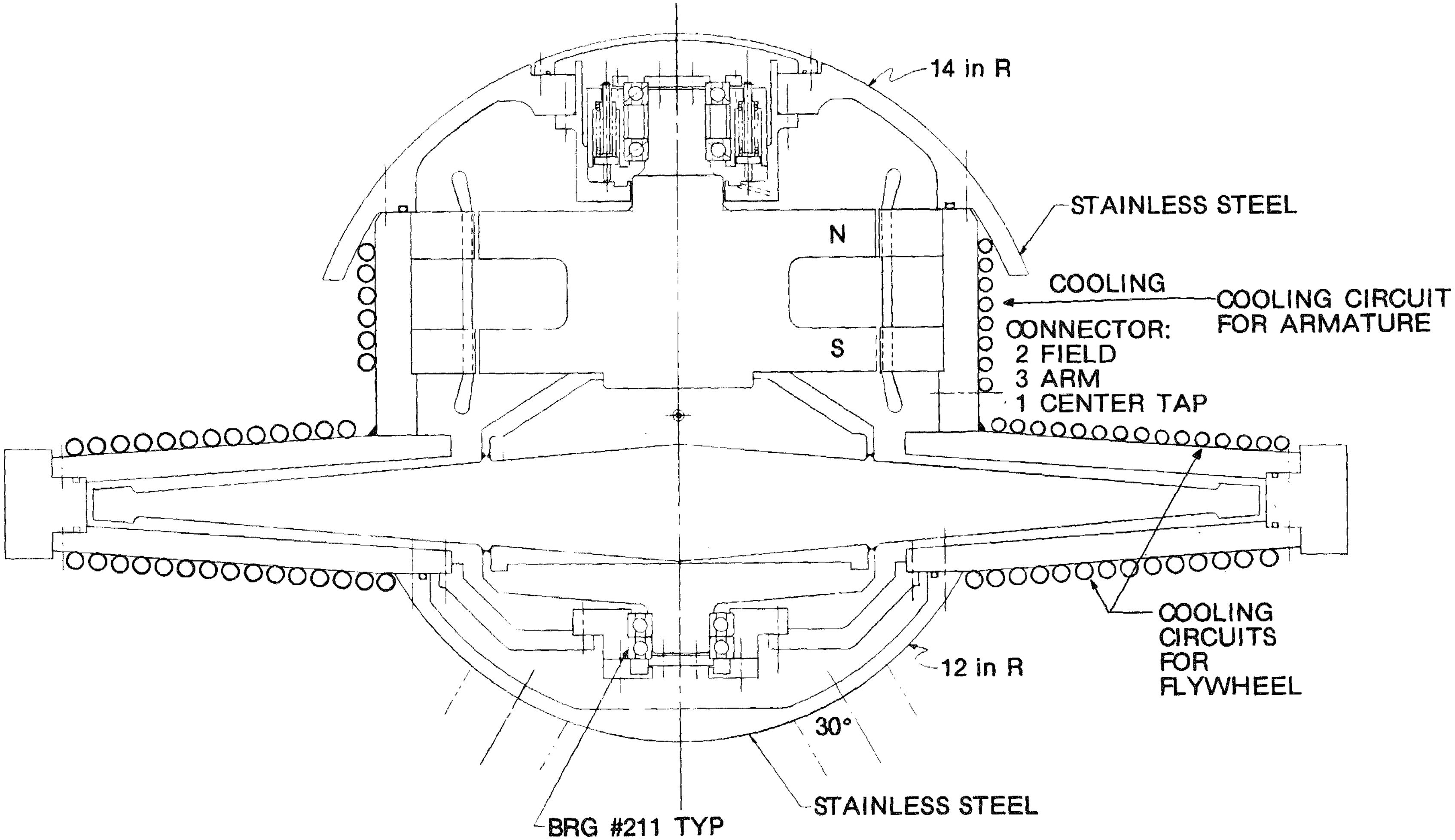
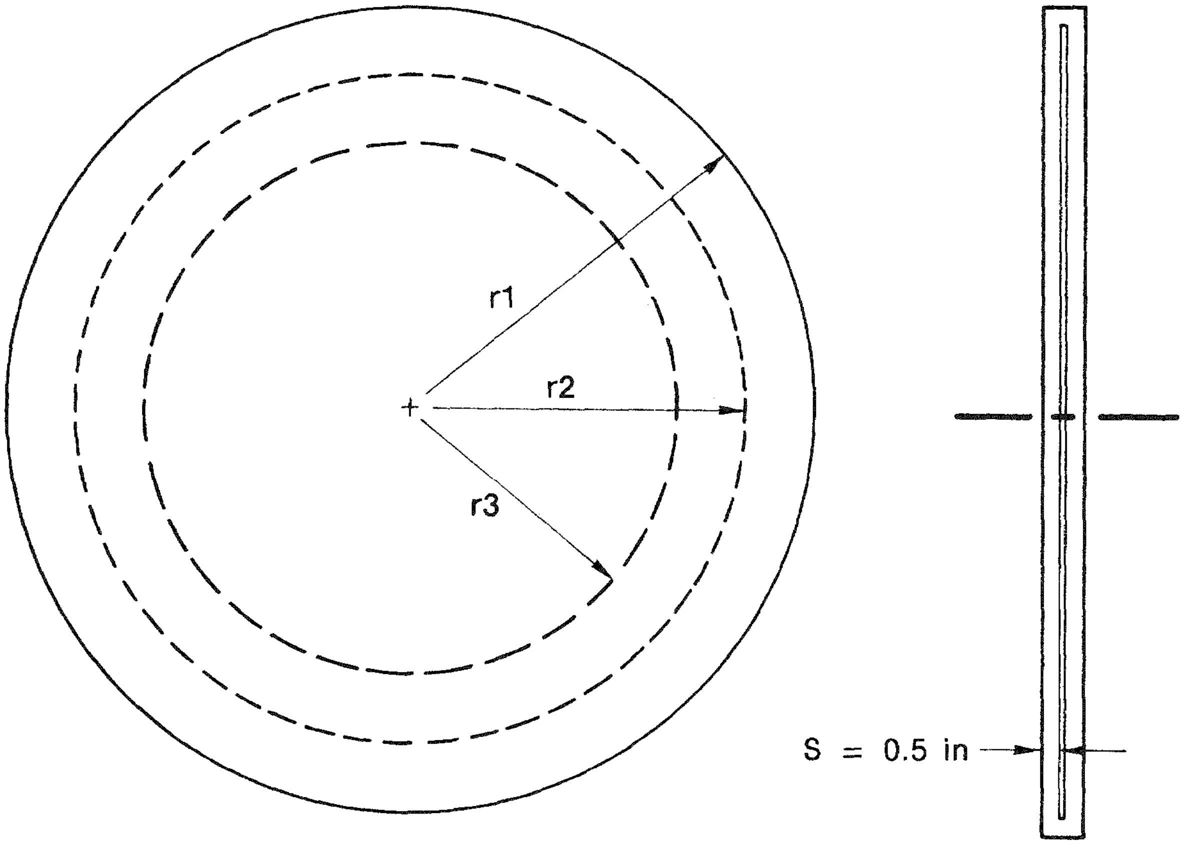


Figure 45. Coolant Tubes Brazed to Capsule



$r_1 = 21.26$ in, $r_3 = 2\,r_1/3 = 14.17$ in, $r_2 = (r_1 + r_3)/2 = 17.72$ in

RADIATION AREA $A_r = \pi r_1^2 + \pi(r_1^2 - r_3^2) = 2{,}209\ \text{in}^2$

CONVECTION AREA $A_c = \pi(r_1^2 - r_3^2) \times 2$ SIDES $= 1{,}578\ \text{in}^2$

VELOCITY $V = 2\pi 17.72 \times 10{,}000 \times 60/12 = 5.57 \times 10^6$ ft/hr

EQUIVALENT DUCT DIA. $D = \dfrac{\text{4 CROSS SECTION AREA}}{\text{CROSS SECTION PERIMETER}}$

$$= \frac{4\,(r_1 - r_3)S}{2\,(r_1 - r_3 + S)} = 0.93 \text{ in} = 0.078 \text{ ft}$$

$$\frac{L}{D} = \frac{2\pi r_2}{D} = \frac{2\pi 17.72}{0.93} = 120$$

Figure 46. Heat Transfer Model

Inserting known values -

$$q_r = \frac{1.47}{3} \times 10^{-10} \times 0.9 \times T_a^{\,3} \times \Delta T \times 1 \times 2209$$

$$= 2.92 \times 10^{-7}\ T_a^{\,3}\ \Delta T \text{ in watts}$$

<u>Convective Energy Transfer</u>

$$q_c = 3.66 \times 10^{-3}\ j\ Cp\ \rho V\ \frac{(Cp\mu)}{k}^{-2/3}\ \Delta T\eta\ A_c,$$

in watts

where: j = heat transfer factor from Figure 47

C_p = specific heat (0.242 for air)

ρ = density corrected for temperature and pressure

$$(\rho = 0.075\ \frac{\text{psia}}{14.7} \times \frac{293}{T_{ave}\ {}^\circ K})$$

V = velocity = 5.57×10^6 ft/hr

μ = viscosity (assumed constant = 0.055)

K = thermal conductivity = 0.018

ΔT = temperature difference in °C

η = effectiveness factor (assumed = 1)

A_c = convective area in sq. in = 1570 in^2

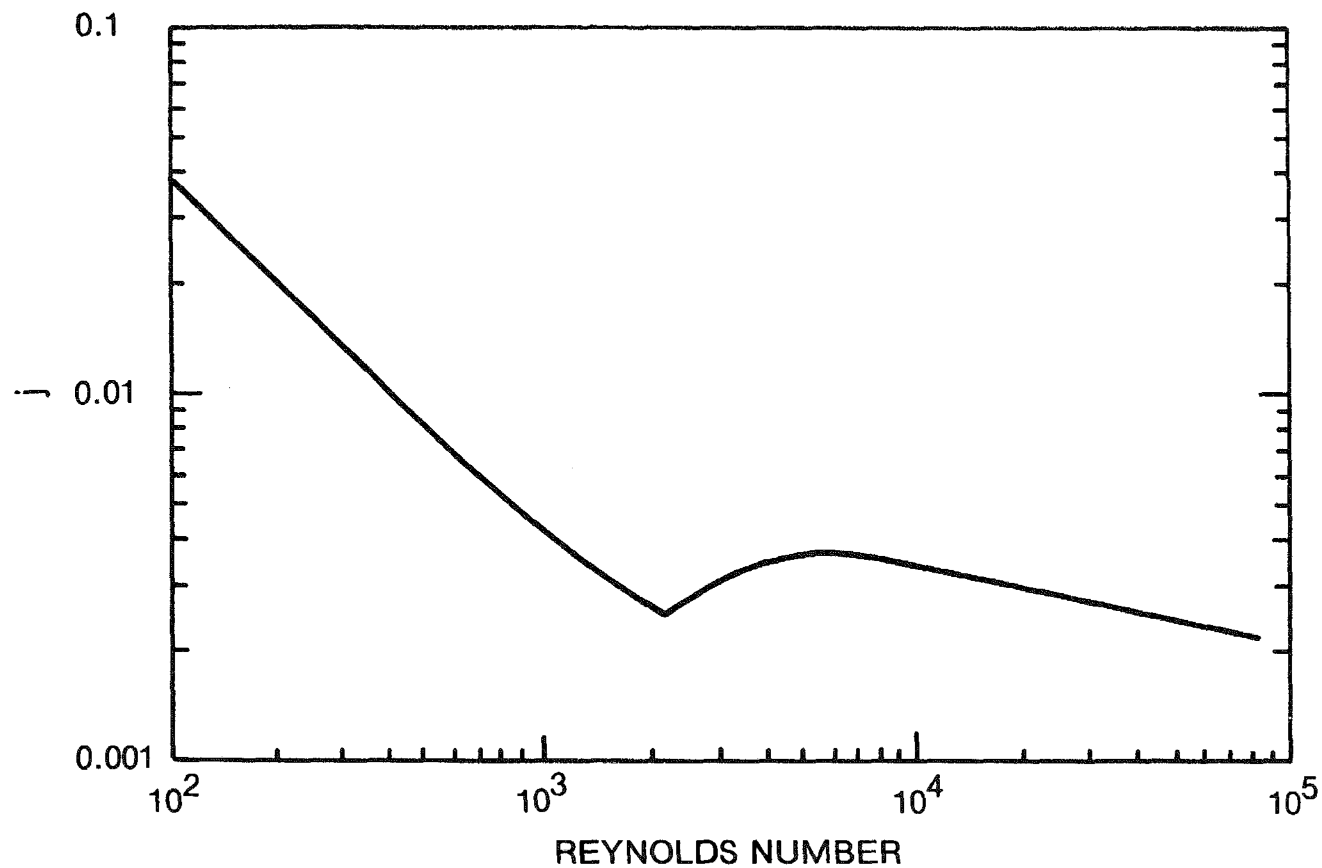
j is found from Figure 47 using Reynolds number

$$N_{Re} = \frac{\rho VD}{\mu} = 0.075\ \frac{\text{psia}}{14.7} \times \frac{293}{Ta} \times 5.57 \times 10^6$$

$$\times \frac{0.078}{0.055} = \frac{1.18 \times 10^7 Pa}{Ta}$$

D = equivalent duct diameter from Figure 46 = 0.078

(Note: This is not the same Reynolds number as that used for windage loss calculations.)



NOTES: 1. FOR SMOOTH STRAIGHT DUCTS
2. FOR GASES WHERE $c\mu/k < 1.5$
3. FOR L/D = 100
4. j IS DIMENSIONLESS

PLOT IS FROM GE INTERNAL DOCUMENTATION

Figure 47. Heat Transfer Factor Versus Reynolds Number

The results of energy transfer due to convection are shown in Figure 48 where energy in watts is plotted against pressure with the enclosure and flywheel temperatures held constant at 50°C and 125°C respectively. It can be seen that over the range of 0.01 to 0.06 psia there is very little change in convected energy transfer. This is because the heat transfer factor (j) decreases almost linearly as pressure increases in this region. The sharp break point at about 0.06 psia is due to the transition from laminar to turbulent flow conditions.

Figure 49 is perhaps of most interest. It shows the rate of energy transfer from the flywheel body (in kilowatts) as a function of its temperature rise above the temperature of the enclosure (50°C). So with something like 750 watts of electrical energy losses in the motor rotor and 500 watts wadage losses in the flywheel, the flywheel temperature might be expected to rise some 65°C above the enclosure ambient of 50°C. This assumes that all of the energy is transferred via the relatively large surface area of the flywheel.

The total motor rotor area is:

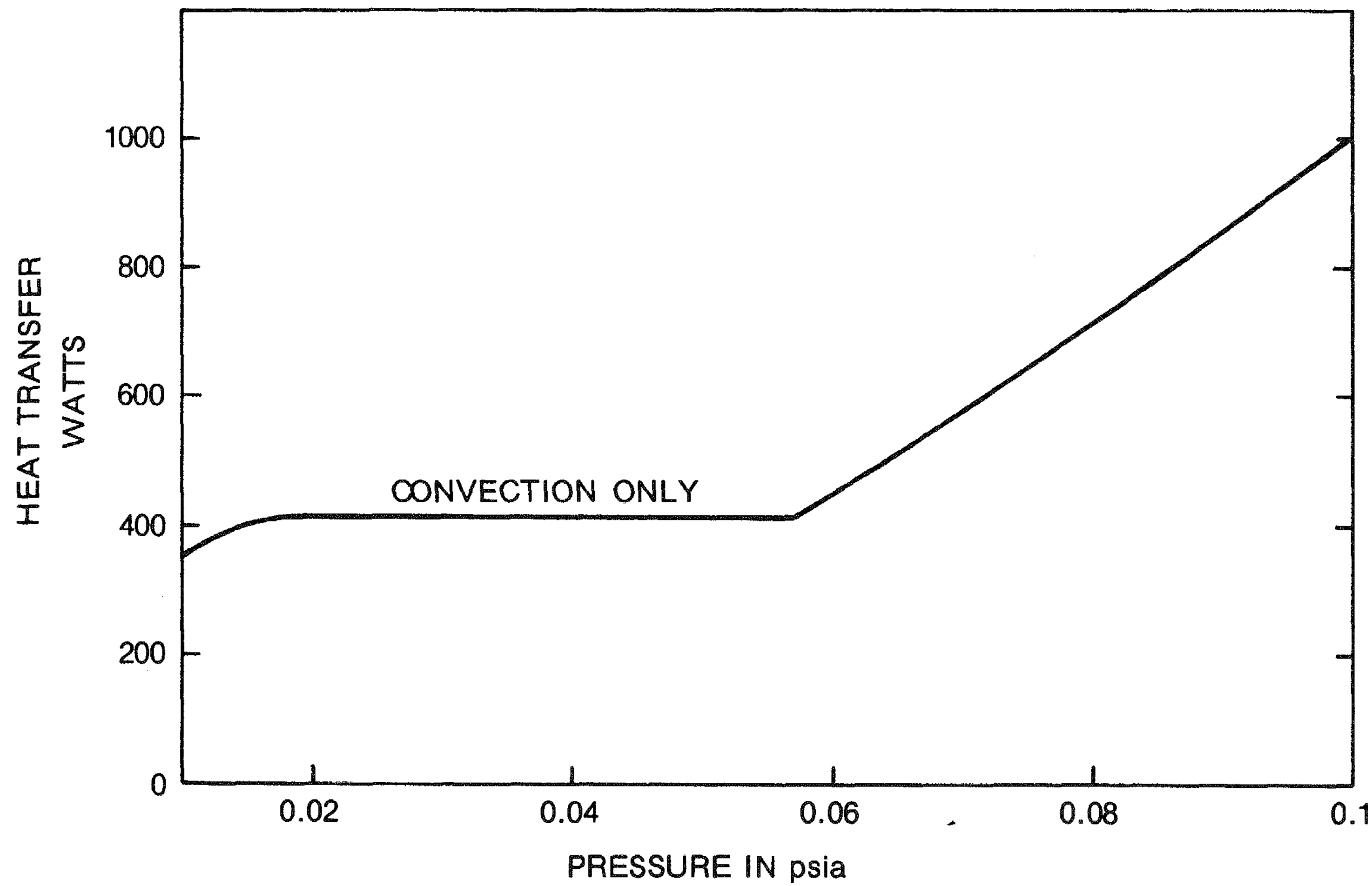
$$A_m = 2\pi r \ell + 2\pi r^2 = 2\pi\ 6.41 \times 7.65 + 2\pi(6.41)^2$$

$$= 1151 \text{ in}^2$$

Assuming that all of the motor rotor area was effective, it represents about 50 percent of the effective area of the flywheel. But due to the electrical losses in the motor stator, in all probability the stator will be operating at an equal or higher temperature than the rotor.

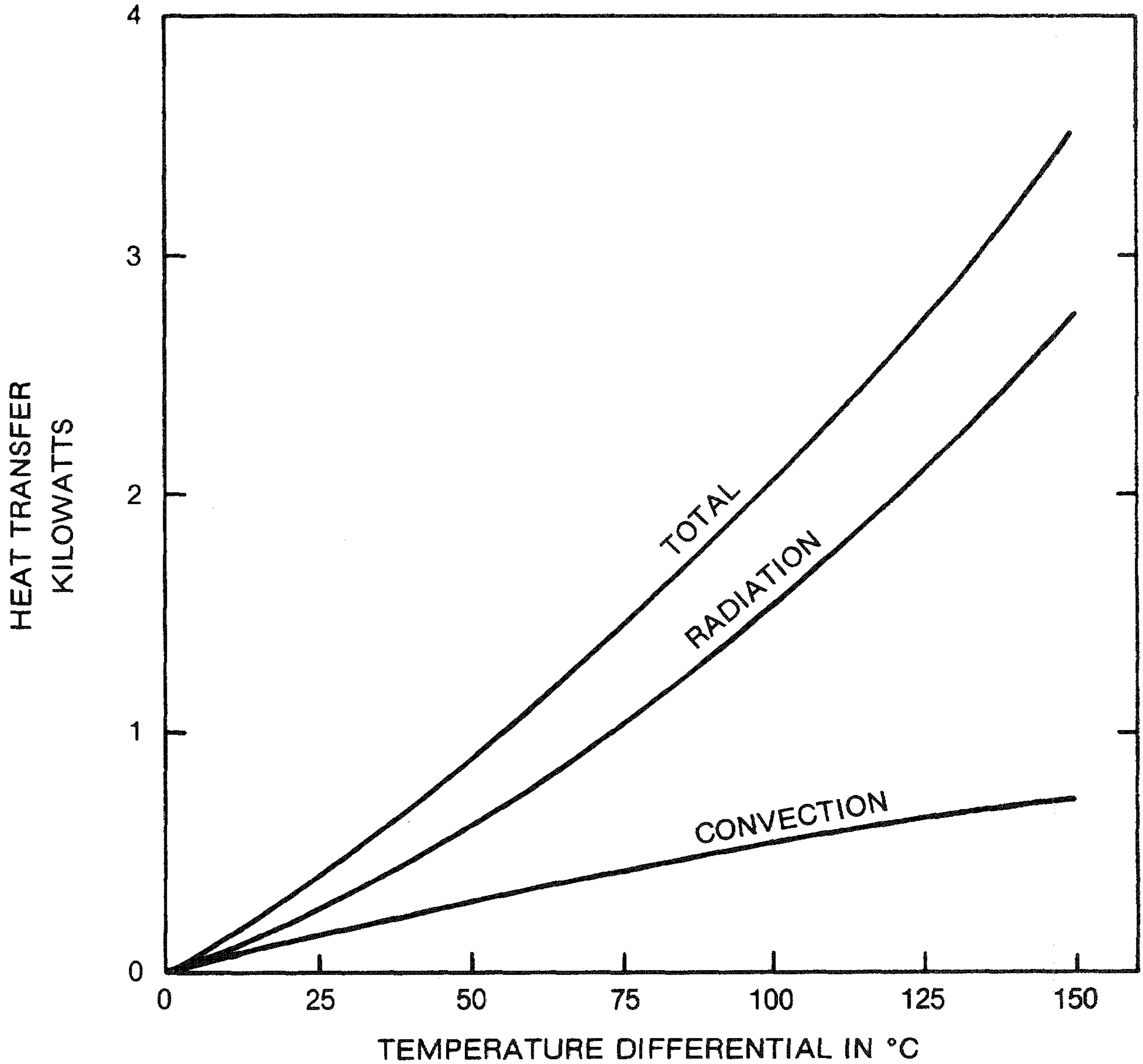
A 65°C rise, or an operating temperature of 115°C, is not considered to pose any serious problems. However, this is a temperature calculated on the basis of an average flow through the motor-alternator. In the worst case duty cycle the motor will be operating at the 200 kW level for 80 seconds followed by several minutes of fluctuating alternator duty at about 20-25 kW. In other words, the whole system (the motor rotor in particular) will be subjected to thermal transients. There is need to investigate the dynamic thermal behavior of the system, taking into account the thermal masses and resistances of the motor rotor, flywheel, and interconnecting shaft to assure that thermal transients can be accommodated without unduly high temperatures in any component.

This analysis has shown that for the worst case mission (4.5 kW hrs) the average temperature rise of the flywheel is within acceptable limits (65°C). However, in the design stage there is a need to investigate the dynamic thermal behavior of the system, particularly the motor rotor.



NOTES: 1. FOR DISC FLYWHEEL AS SHOWN IN FIGURE 20
2. FOR ENCLOSURE TEMPERATURE = 50°C, FLYWHEEL TEMPERATURE = 125°C

Figure 48. Heat Transfer Versus Pressure at Constant ΔT



NOTES: 1. FOR DISC FLYWHEEL AS SHOWN IN FIGURE 20
2. FOR ENCLOSURE TEMPERATURE = 50°C
3. FOR ENCLOSURE PRESSURE 0.01 TO 0.06 psia

Figure 49. Heat Transfer Versus ΔT at Constant Pressure

4.5.7 Design Areas for Further Work

The following design areas for the flywheel capsule warrant significant effort in the subsequent design stage.

Bearings

- Establish a design goal for minimum life
- Investigate feasibility of controlling load division between two bearings by controlling the spacing between races
- Confirm calculations with bearing manufacturer
- Develop an impact method for creating a "mist"
- Develop reliable oil metering method
- Determine bearing losses from bearing manufacturer
- Evaluate bearing heating and incorporate in loss calculations
- Evaluate test data from the Postal Jeep Program

Vacuum System

- Determine the vacuum pump capacity needed
- Assess gas generation from the oil and its impact on the pressure
- Identify the pump down demand and frequency of pumping
- Evaluate the consequences of pumping down too low
- Select vacuum instrumentation

Cooling

- Resolve the heat conduction from the flywheel into the flywheel
- A dynamic evaluation of cooling must be made to consider the momentary high heat input during spin-up
- Ascertain need for temperature instrumentation

4.6 ENCLOSURE HOUSING FLYWHEEL CAPSULE

The following discussion evaluates the design considerations in the enclosure due to the gyroscopic action of the flywheel. The analysis of radial loads on the spin axis bearings suggests that the enclosure be designed with spherical bearing supports that allow $\pm$ 20° tilt, an antirotation gimbal, and vertical alignment equipment.

4.6.1 Description

Recognizing that a high-energy flywheel is by definition also a high-energy gyro, it is obvious that the maneuvering of the shuttle car can generate serious problems for the flywheel. Since the car maneuvering is essentially in a horizontal plane, the primary gyroscopic problem is eliminated by orienting the spin axis of the flywheel to local vertical.

Next in order is the matter of mounting of the flywheel capsule to the chassis. If the chassis had a soft suspension like a bus, one could seriously consider direct mounting to the chassis, but the shuttle car has no suspension other than the 100 psi tires, and the mine road bed is much worse than ordinary roads. Therefore, a soft suspension for the capsule is mandatory.

Figure 21 illustrates the mounting approach. The flywheel capsule is enclosed in a dust-proof enclosure which is shock-mounted to the chassis of the vehicle, using conventional means such as rubber mounts in conjuction with shock absorbers. This alone would be adequate if the flywheel were not spinning; however, the gyroscopic action requires the equivalent of gimbals to permit angular excursions of the spin axis relative to the car chassis. A true gimbal to support a 2000# flywheel unit is a sizable structure. To improve on this, an alternate method is used as shown in Figure 21. Note the shperical bearing surface on the bottom of the flywheel capsule with spherically oriented bearing pads on the enclosure. Thus, the capsule has the tilting freedoms of a gimbal, yet the vertical load and shock forces are carried directly to the enclosure base. A capsule rotation restraint to oppose the spin-up motor torque (300 ft lbs) is also required, as covered in Section 4.6.3.

The consultant for gyroscopic action has been A.G. Robins of the General Electric Aerospace Controls and Electrical Systems Department, Binghamtom, N.Y.

4.6.2 Radial Loads on Spin Axis Bearings

The major concern in mounting the flywheel capsule is the effect of pitch and roll vehicular motions on the main spin axis bearings. These bearings operate at extreme speeds for their large size, so they have little excess capacity. The radial loads imposed on the spin axis bearings are directly proportional to the angular rate of pitch and roll motion of the flywheel capsule. The angular rate produces a torque:

$$(1) \quad T = I\,\omega\Omega$$

I = Spin element inertia, lb-in-sec^2

ω = Spin velocity, radians per sec = 1047

Ω = Precession rate, radians per sec

T = Torque, lb-in

And, the bearing loads are related to torque:

(2) PB = T

P = Radial Load on Bearings, lbs

B = Bearing Effective Separation, 18 in

P = T/18

The inertia for Equation (1) is calculated by Equation (21) from Figure 7:

(3) $$KE = \frac{1}{31,850} \left(\frac{I\omega^2}{2}\right)$$

KE = Watt-Hours Total Stored Energy, 6000 W hrs

$\therefore$ I = 349 lb-in-sec^2

Combining all three equations:

(4) $$P = T/B = \frac{I\omega\Omega}{B}$$

(5) $$P = \frac{349 \times 1047\Omega}{18 \text{ in}} = 20,300 \text{ lb per rad./sec}$$

Converting to degrees:

(6) P = 354 lb per degree/sec

To evaluate this relationship, earth's rotation is 15 degrees per hour and the forces are insignificant:

(7) $$Pe = 354 \times \frac{15°}{3,600 \text{ sec}} = 1.5^{\#}$$

If the flywheel capsule were to be tilted at a moderate rate of 10 degrees/sec, the bearing forces would be extremely high:

(8) $$P_{10} = 354 \text{ lb} \times 10°/\text{sec} = 3540^{\#}$$

It is true that, with the spherical bearing that mounts the capsule, the static friction of the bearing limits the torque that can be imposed on the spin axis bearings. This is evaluated as follows:

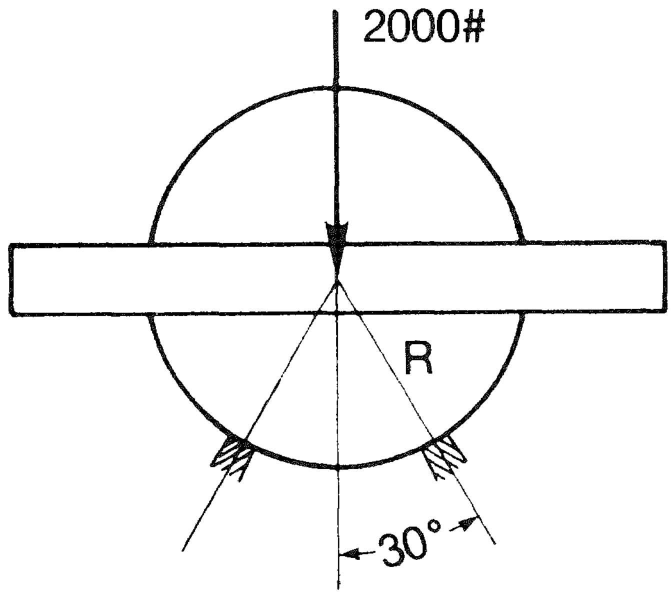


Figure 50. Force Diagram of Capsule

$$(9)\quad T_F = \mu NR = \mu \times \frac{\omega}{\cos 30} \times R = \frac{.1 \times 2{,}000}{.866} \times 12 \text{ in}$$

$$= 2770 \text{ in-lb}$$

$$(10)\quad P = 2770/18 \text{ in} = 150^{\#}$$

From this it can be seen that the coefficient of friction cannot be allowed to be very high. This is an application for Teflon-based bearing stock or an equivalent low friction material. Complete elimination of friction could be achieved by using hydrostatic pressure in the spherical bearing.

4.6.3 Gimbal to Provide Motor Torque Restraint

The spin axis motor-generator applies torque to the flywheel and this torque must have an equal and opposite reaction from the vehicle. This means the capsule must be anchored to the enclosure or the capsule will spin in the opposite direction of the flywheel. Ideally this restraint should be a pure torque so that it produces no sidewise forces on the spherical bearing. If a single tie rod were used to anchor the capsule to the enclosure, there would be a sidewise force on the spherical bearing that would be equal in magnitude to the tie rod force. The ideal method would be to use a gimbal since it will remove torque without generating any disturbing forces. With gimbal bearings located just beyond the outside diameter of the capsule, the motor torque will thus produce gimbal bearing loads of approximately 70 lbs. This is significantly lower load than the 2000# weight of the flywheel capsule which bears on the spherical bearing, so the admission of a gimbal in no way opens the subject of using a gimbal to support the whole unit as in a gyro.

It will be impossible to insure that the center of the spherical bearing coincides with the center of the axis of the gimbal; therefore, all four gimbal bearings should be sleeve bearings, with rubber support.

The gimbal ring may be given additional stiffness in the area of the attachment to the capsule, since the ± 20° motion does not sweep out that area. Full clearance must be provided inside the ring at the points of attachment to the enclosure.

Figure 21 shows a cross section of a gimbal for the purpose of determining the clearances required.

4.6.4 Precession Characteristics of the Flywheel Capsule

The total tilting angle of the flywheel capsule is made up of the roll and pitch angles of the vehicle plus the precession angle of the spin axis. Qualitatively, the spin axis tends to align itself in a normal manner to the plane of vehicular maneuvering. This rate of alignment is in proportion to the "error" from normal. Most importantly, there are two ways to align the spin axis with the normal: top side up, and top side down. The direction is determined by the direction of maneuvering, CCW producing the opposite direction of CW. This bi-directionality of self alignment is the sole reason that necessitates active alignment facilities on board the vehicle.

4.6.5 Alignment of the Spin Axis

All gyros are equipped with a caging system to hold the spin axis in correct orientation during spin-up. The problem is slightly different with the shuttle car; the spin axis must, on occasion, be brought back to vehicle vertical to avoid contacting the ± 20° limits to capsule tilt. Hydraulic cylinders seem ideal for applying a force to restore the capsule to vertical, and this force must be applied at 90° from the tilt angle because of the precessional characteristic of the flywheel. This is unlike the conventional gyro-caging system. It requires sensing the direction of tilt to be able to apply the force at 90° from the error. The simplest solution would be to use four hydraulic cylinders with appropriate control to provide alignment. On the basis that correction once per trip is adequate, the cylinders could be reduced to two. The vehicle in backing out of the charging station and returning must maneuver through at least 180° which provides at least two passes per trip at the correct position to apply the correction force (torque). Undoubtedly, there are several other combinations possible and these all need to be defined carefully prior to evaluation and finalizing an approach.

One complicating factor to alignment is the precession rate versus bearing load relationships of Equations (1) and (2) above. The faster the correction is achieved, the higher are the radial loads that are applied to the bearings. In fact, as a reference, Equation (6) indicates that for a correction rate of one degree per second the corresponding bearing load is 354 lbs. Thus one degree per second can readily become the limiting rate.

The tilt excursion limits of $\pm$ 20° were determined by the geometry of the enclosure and are not based on an analysis of what is needed. The $\pm$ 20° is the maximum allowable without increasing the height of the enclosure over that necessary to house the flywheel capsule in the vertical position.

4.6.6 Center of Gravity of Flywheel Capsule

A qualitative check indicates there is no advantage to giving the flywheel capsule any pendulous moment. Such a restoring moment tends toward precession in a circular fashion, so it is recommended that the center of gravity should be placed coincident with the center of radius of the spherical support bearing.

4.6.7 Design Areas for Further Work

Gyroscopic Action of Flywheel

- Calculate gyroscopic forces, under simulated dynamic conditions.
- Determine tilt limits permissible.
- Design the restore-to-vertical actuators, sensors, and control.
- Calculate spin axis bearing forces generated by gyroscopic action.
- Design the motor torque restraint gimbal.

Enclosure Mounting

- Determine the mine vehicle excursions.
- Design the rubber mount and shock absorbers, and clearances suitable for the vibration input.
- Design the rubber mount to resist the motor torques.

Enclosure as Adjunct to Containment

- Strengthen sidewalls to enhance flywheel containments.

- Add emergency rings around spherical bearings to keep the capsule captive in event of catastrophic failure of the flywheel.

4.7 ELECTRICAL SYSTEMS FOR FLYWHEEL POWERED SHUTTLE CAR

Two electrical systems have been studied (Figures 51 and 52). System concepts are based, in part, on General Electric Company Technical Quarterly Progress Reports 1, 2, and 3 entitled "Demonstration of an Inductor Motor/Alternator/Flywheel Energy Storage System" for the U.S. Energy and Research Administration. The first system is a rectifier-inductor machine system, wherein the flywheel inductor motor/alternator ac output is rectified with an uncontrolled diode rectifier to provide power to a dc motor-transmission-torque converter system. Charging of the flywheel to maximum speed is accomplished through a load commutated inverter (L.C.I.) at the recharge station.

The second system is an L.C.I. inductor machine system wherein the mechanical transmission is replaced by an electrical transmission consisting of a dc traction motor and an on-board L.C.I. Both systems require a 600 Volt dc power source when recharging the flywheel to maximum speed.

4.7.1 Rectifier-Inductor System (Figure 53)

Rectifier-Inductor Drive

The inductor motor/flywheel package initially operates from a wayside power supply consisting of a solid-state load commutated inverter (L.C.I.) and control designed to provide the necessary frequency and voltage control to the on-board inductor motor from a 600 volt dc source. During operation of the inductor motor/flywheel unit, the inductor motor field excitation is controlled by a phase controlled rectifier. Once the motor flywheel is charged to full speed, the car is disconnected from the recharge station, and the flywheel drives the inductor motor as an alternator. The resultant ac power is rectified with a three-phase diode rectifier to provide current for the dc motor. The dc motor operates at an essentially constant speed and is provided with constant voltage from the regulated inductor machine rectified output. The dc motor is non-reversing and has compound wound fields. A starting resistor is provided to start the motor. The power transmission is based on the drive train of the tractor-trailer car and is accomplished through an operator-controlled power shift transmission/torque converter unit and a forward/reverse gear.

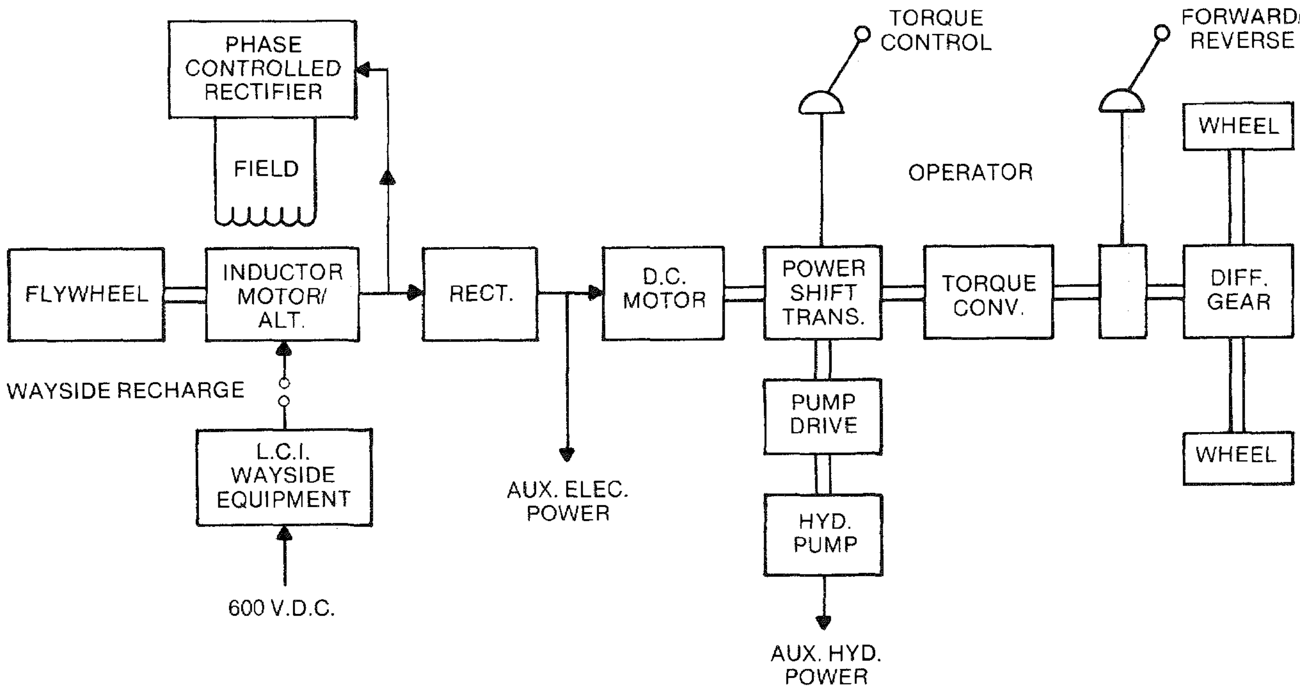


Figure 51. Rectifier-Inductor Control System

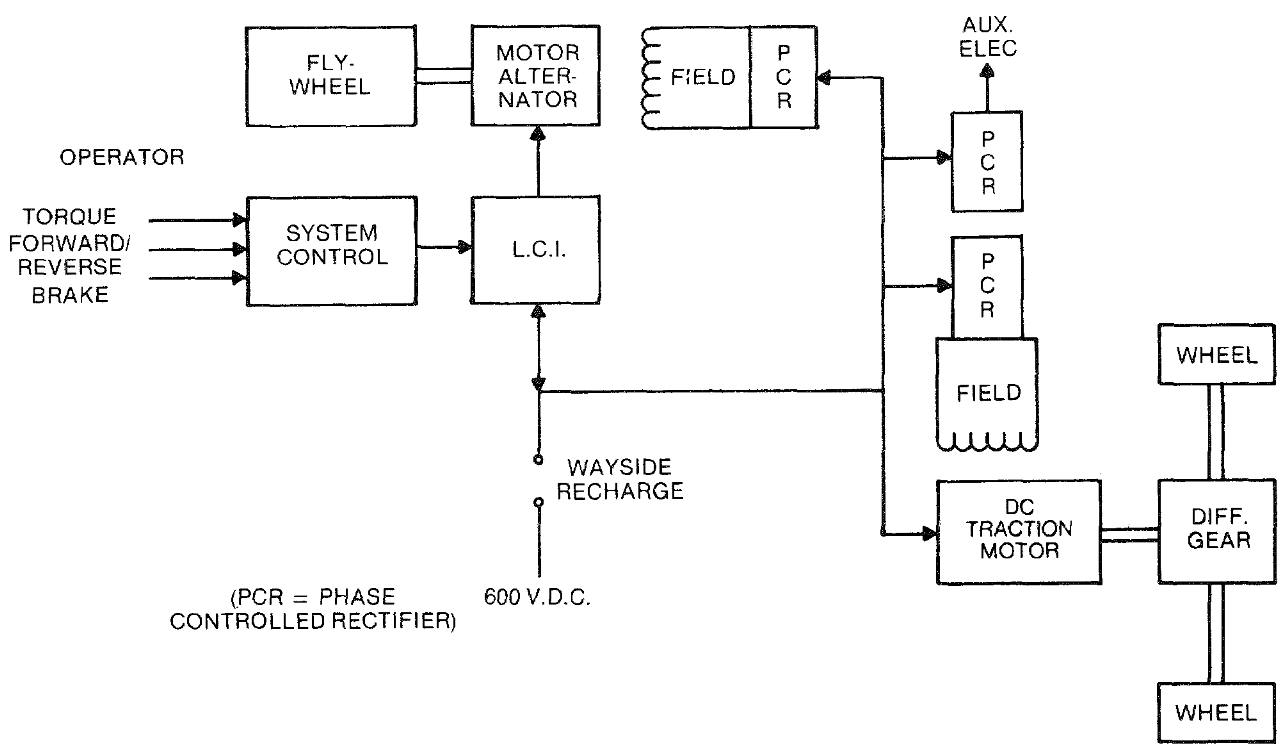


Figure 52. L.C.I. - Inductor Control System

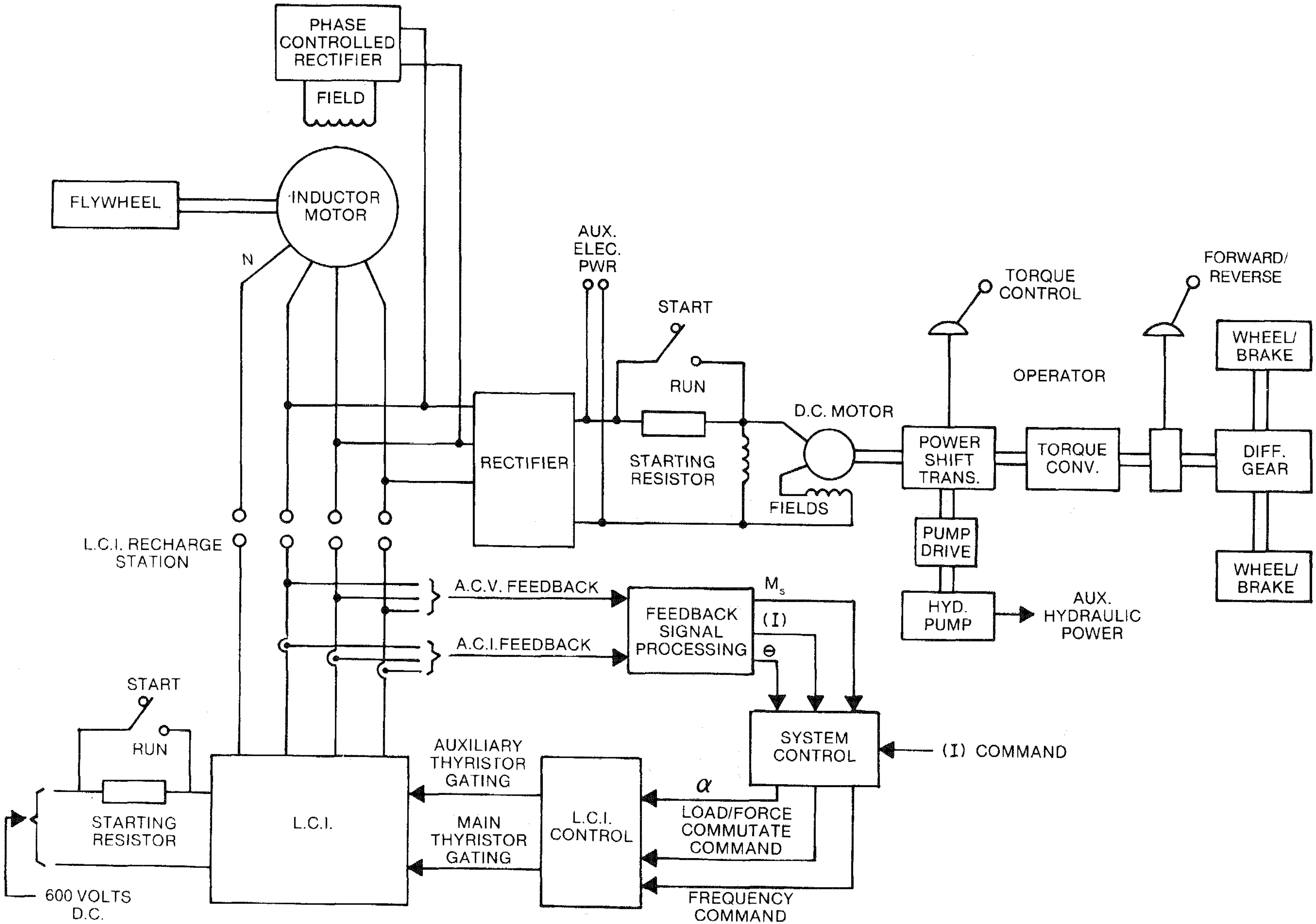


Figure 53. Rectifier-Inductor Control System

Auxiliary hydraulic power for operation of power steering and trailer unloading mechanisms is obtained from a hydraulic pump/pump drive system utilizing an output connection of the power shift transmission. Auxiliary electrical power for lights and any other peripheral electrical equipment is obtained from the rectified output of the inductor machine operating as an alternator.

There is no regeneration of flywheel power upon braking or slowing down of the vehicle.

Load Commutated Inverter

Figure 54 shows the motor/flywheel is initially coupled electrically to the L.C.I. at the recharge station to bring the flywheel up to speed. This configuration would be used in the block diagram of Figure 53. Note that access to the motor neutral is required. When charging, the L.C.I. is controlled by either auxiliary or main thyristor gating signals. The gating frequency of the inverter is matched to the speed of the inductor motor and proper gating delay angle is determined by L.C.I. control.

When initially starting up from zero rpm, this circuit employs two auxiliary thyristors (TN, TP) and a single commutating capacitor. The conducting main thyristor connected to the positive dc bus (T1, T2, and T3) is commutated off by firing auxiliary thyristor TP. The conducting thyristor connected to the negative dc bus (T4, T5,and T6) is commutated off by firing auxiliary thyristor (TN). The peak capacitor and thyristor voltages are determined by the capacitor size and by the inductor motor parameters. In this circuit the rate of rise of thyristor currents is inherently limited by the motor leakage inductances. The inductor LI is required for smoothing of the dc power source.

Resistor (R) and switch (S1) are used only when charging the flywheel up to about 15% speed. The resistor function is to limit the current when the motor back e.m.f. is low. Also required is an auxiliary dc field supply for the inductor motor for starting the system if residual voltage on the motor is too low. The source of this dc may be from the wayside station or vehicle batteries.

The normal operation during the charging cycle from 5,000 to 10,000 rpm involves the thyristors T1 - T6. The other two thyristors TN, TP and Capacitor C1 are components that are necessary when starting the system from zero speed and are commonly referred to as the forced commutation circuitry. These are necessary to provide forced

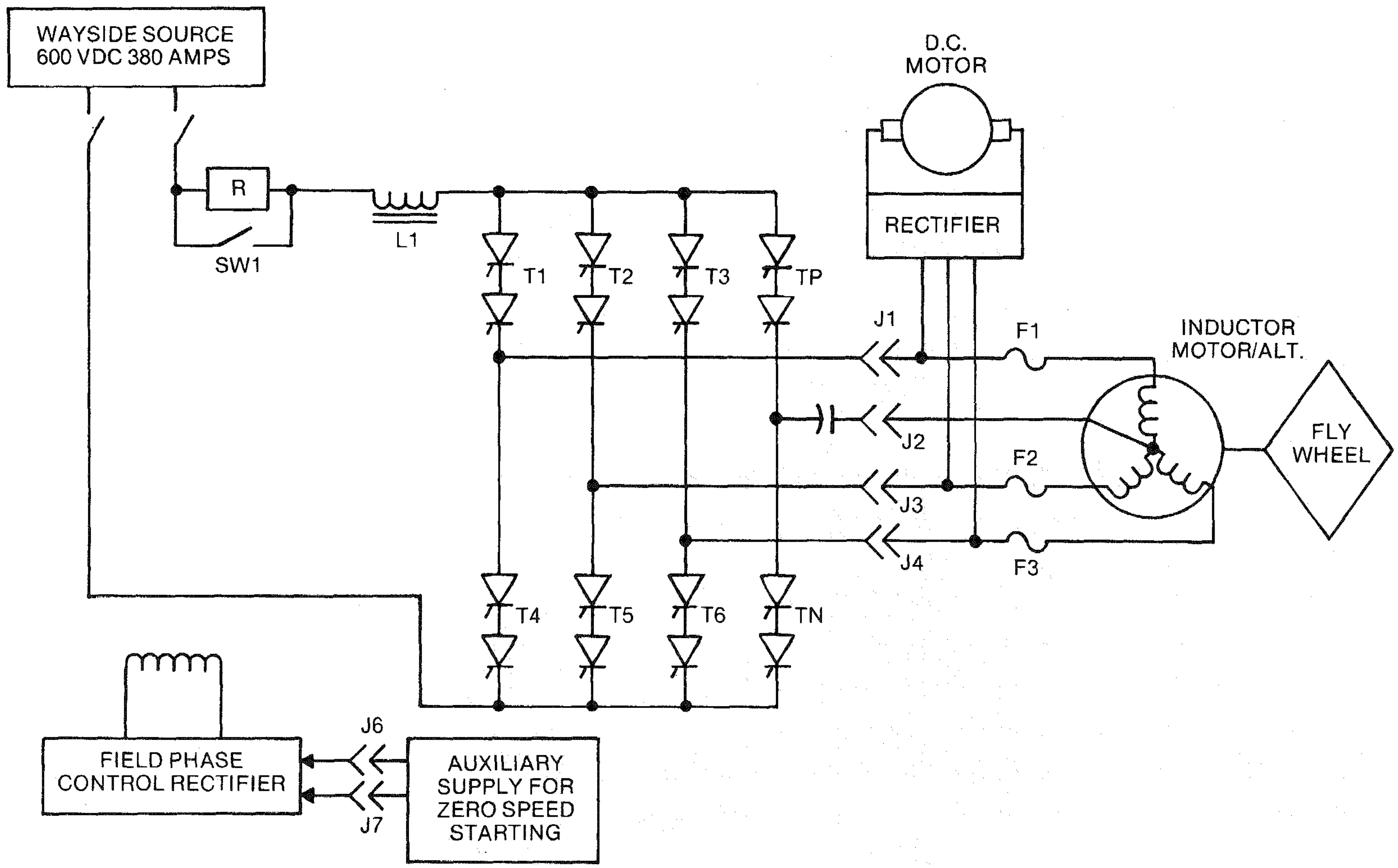


Figure 54. Wayside Load Commutated Inverter

commutation to thyristors T1 through T6. After sufficient speed is attained, gating signals to TN and TP are removed and the normal control for gating T1 through T6 will continue the operation up to the maximum speed.

Figure 54 also indicates the need for four power connections J1, J2, J3 and J4 to the inductor motor and connections J6 and J7 to the field phase control rectifier.

Preliminary work with the 4.5 kW hr flywheel motor/ alternator set indicates that commercially available inverter grade C612 thyristors can be used. Two C612 thyristors (2,000 V-PRV) must be connected in series to accommodate the motor back e.m.f. generated at top speed.

The selection is based upon:

- A 50% duty cycle, (80 seconds charge time 80 seconds off time)
- frequency variation from 417 to 834 hertz
- A dV/dt of 100 V/μs
- 100 A/μs leading and falling edge of the current waveform
- 600 volt 380 ampere supply
- Airflow of 450 SCFM at 2 inch water gauge pressure drop for cooling.

A preliminary estimate of the size of the inverter/ rectifier or wayside inverter would be approximately 30 inches wide, 30 inches long and 42 inches in depth. The weight would be approximately 1,000 pounds including the auxiliary field supply. The sizing is based upon the circuit involving the eight thyristors which is applicable to wayside or on-board systems. If ac power is available at the recharge station, the lossy starting resistor circuit can be replaced by a more efficient phase controlled rectifier.

Load Commutated Inverter Control

The L.C.I. control provides the thyristor gating signals synchronized to the inductor motor speed by the frequency command from system control. When necessary, the L.C.I. control also drives the forced commutation circuitry of the L.C.I. as required for initial starting. The gate firing delay angle α is used to control the current to the inductor motor, and it limits the dc link current flowing from the 600 volt dc source through the L.C.I.

System Control

The system control circuit receives the preset inductor motor stator current $|I|$ command signal in addition to feedback signals such as inductor motor speed

M_S, actual inductor motor stator current $|I|$, and the ac voltage-current phase angle θ. System control then generates output signals described in the preceding paragraph which initiate proper response of the L.C.I.

Feedback Signal Processing

Motor terminal voltage and stator current are sensed by the feedback signal processing unit to determine motor internal conditions such as current magnitude $|I|$, speed M_S which synchronizes L.C.I. firing, and the feedback signal θ which represents the stator voltage-current phase angle and determines the gate firing delay α of the L.C.I.

Control Strategy

During charging, the system operates by maintaining essentially constant flux in the inductor machine as its speed is varied from half speed to full speed. This is accomplished by maintaining the field excitation relatively constant. The resultant inductor motor internal voltage increases in proportion to speed. The machine terminal voltage also increases but at a lesser ratio because of the nonsinusoidal terminal voltage.

Since the machine reactance increases with frequency, the ac voltage increasing with frequency causes the commutation time (the time it takes to commutate current from one machine phase to the next) to remain a fixed percentage of the cycle (with constant current).

In addition, the percentage of the time available for turn-off time tends to remain constant with increasing voltage and with increasing frequency. Thus, this strategy of allowing ac voltage to vary with frequency is favorable in terms of operation of the inverter. The motor losses, however, are higher in this mode of operation than if the voltage were held constant with increasing frequency. The power provided to the flywheel is constant during the charge time. The dc link current is regulated constant and the dc input voltage is constant. Thus, input power to the inverter is constant.

4.7.2 L.C.I. - Inductor System (Figure 55)

The inductor motor/flywheel package initially operates from an on-board solid-state load commutated inverter (L.C.I.) designed to provide the necessary frequency and voltage control of ac power from a 600 volt dc source.

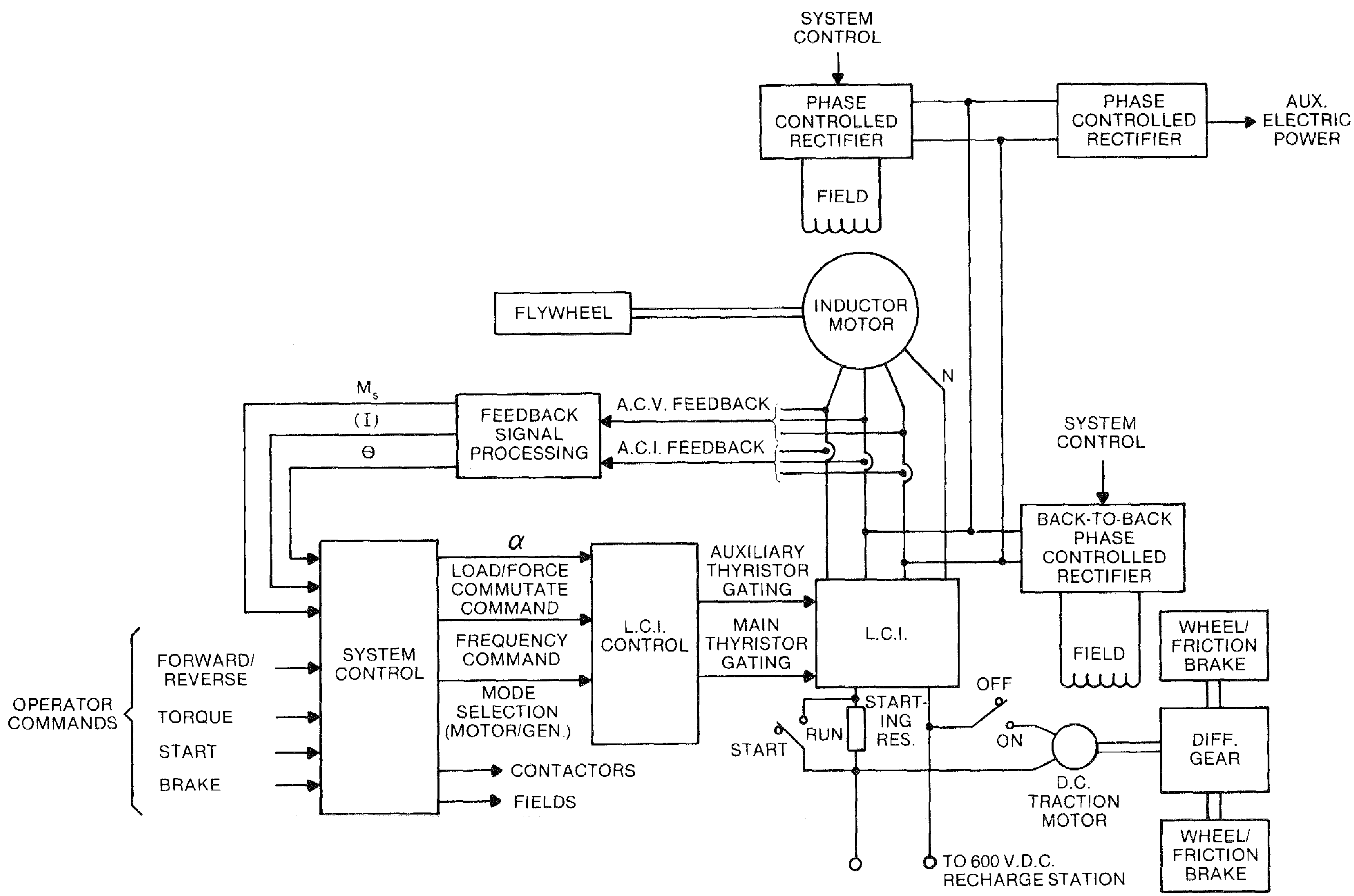


Figure 55. L.C.I. - Inductor System

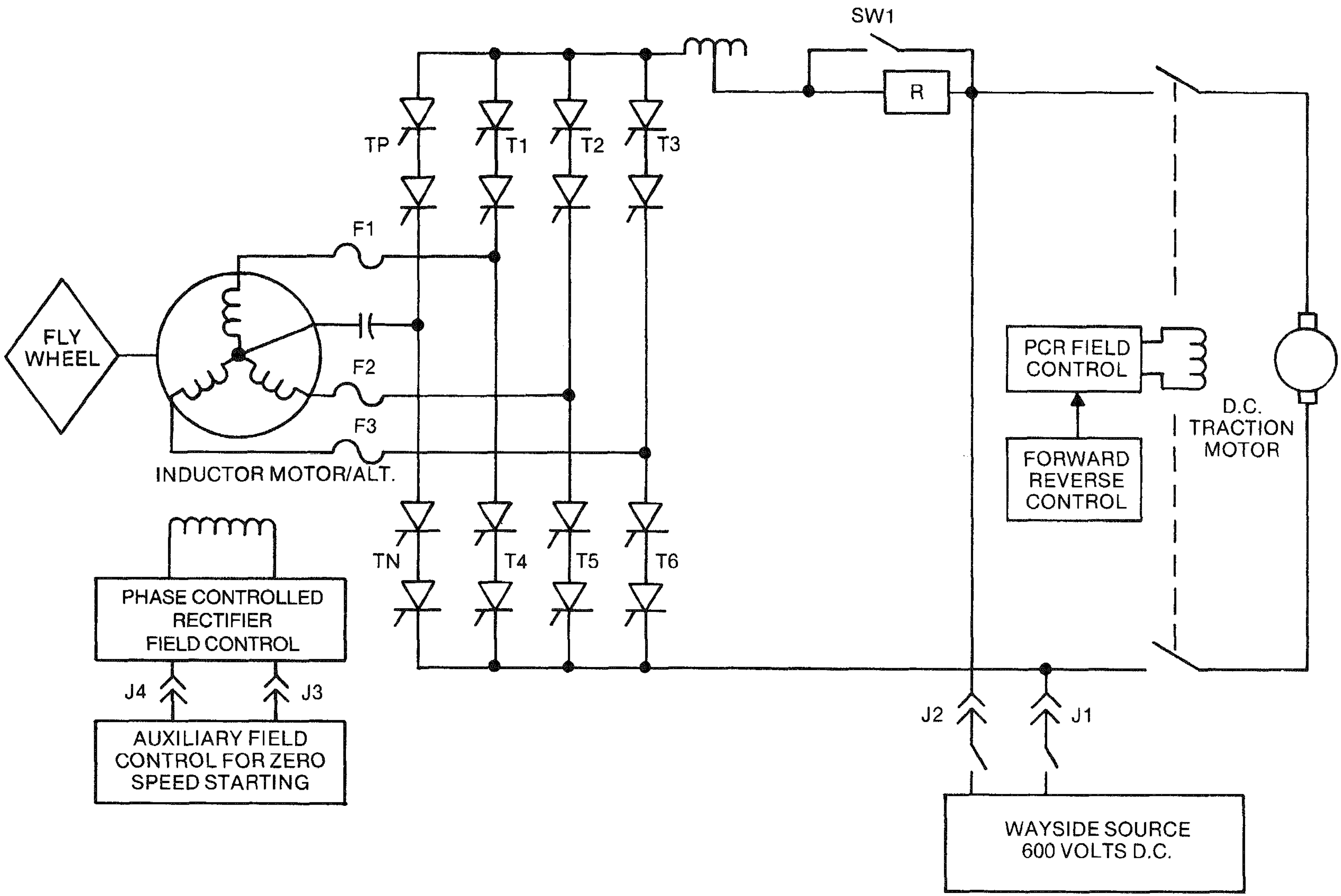


Figure 56. Load Commutated Inverter for Regeneration

System control maintains the inductor motor field at a near-constant value while charging. Once the motor/flywheel is charged to full speed, the dc power can be disconnected and the flywheel will drive the inductor motor as an alternator. The ac power generated is then passed through the L.C.I. operating in the phase controlled rectifying mode, producing adjustable dc power for the traction power through the differential gears. The system control circuits receive operator commands, such as desired torque and brake signals, and provide appropriate commands to the L.C.I. and motor fields for proper vehicle operation. When the vehicle slows down or brakes, the dc traction motor field current is reversed. The traction motor then operates as a generator and transfers a portion of the vehicle kinetic energy through the L.C.I. to the inductor/flywheel to increase the flywheel speed.

The dc traction motor field is excited by a back-to-back thyristor configuration which provides the reversing action needed when the operator commands a change in drive direction (Forward/Reverse) or when the system changes from the regenerative to the motoring mode.

In the event the braking action of the electrical drive system in the regenerative mode is not adequate or fails, electrically or hydraulically actuated friction brakes are used. Auxiliary power for electrical accessories and electrically operated hydraulic drives are obtained from the regulated output of the inductor motor/alternator through a phase controlled rectifier.

L.C.I.

The on-board load commutated inverter (L.C.I.) is initially coupled electrically to a 600 volt dc source at the recharge station to bring the flywheel up to speed (Figure 56). The L.C.I. is bi-directional in that dc to ac inversion and ac to dc rectification can be accomplished through the same unit. This bi-directionality is the result of the main thyristors within the L.C.I. being gated by control signals appropriate for the two modes used.

The inverter used in the L.C.I. - Inductor System is identical to the inverter used in the Rectifier-Inductor System. For regeneration, control of the dc traction motor field polarity is required. This allows the motoring and braking mode of operation to be accomplished using the six thyristors (T1 - T6) in the same manner as the Rectifier-Inductor system inverter previously discussed. The diagram of Figure 56 also shows the connections for wayside charging. As the figures indicate, the number of connections has been decreased by two for the inverter/rectifier on-board than for the inverter located on the wayside. This is a result of two-wire dc power required by the on-board L.C.I. when at the way-station rather than the four-wire ac required by the inductor machine.

L.C.I. Control

Bi-directional power control capability is required of the L.C.I., as it must provide power to the inductor motor during vehicle braking or coasting and extract power when cruising or accelerating. The load commutated inverter control provides the thyristor gating signals for the motoring and generating modes of the L.C.I. The L.C.I. control also drives the forced commutation circuitry of the L.C.I. during initial starting.

The load commutated inverter cannot operate in the motoring mode with the inductor motor at a standstill. Hence, the forced commutation circuitry is activated by providing gating signals to the auxiliary thyristors and turning off the main thyristors. Once a net positive torque is produced the motor/flywheel accelerates. At 10 to 20 percent of rated motor speed, system control provides a load commutate command which stops the forced commutation circuit from operating and enables normal motoring using the main thyristors.

The desired L.C.I. gating frequency is supplied by system control and is a function of motor speed. The gate firing delay angle α is used to control the current to the inductor motor or dc motor, depending on the mode, and is limited by the dc link current flowing from the dc side of the L.C.I.

System Control

Composed of both digital and analog control elements, system control computes desired responses of vehicle components given a set of operator commands and operating conditions.

System control is responsible for mode selection of the L.C.I. The L.C.I. must be capable of extracting energy from the inductor motor/flywheel during one mode of operation (generating) and supplying energy to the motor/flywheel during the second mode of operation (motoring). The L.C.I. must also be capable of starting the motor/flywheel from rest and stopping the flywheel during the final coastdown to a standstill. The system control is designed to monitor the status of the system to ensure that the change in modes, i.e., motoring to generating, forced commutation to motor commutation, etc., are accomplished at the correct time so as to minimize disturbances to the system.

The system control circuit receives operator commands, such as torque and brake signals in addition to feedback signals, such as the magnitude of stator current $|I|$, inductor motor speed M_S, and the ac voltage-current phase angle θ. These input signals are compared to the motor operating conditions, and appropriate commands are sent to the motor field exciters, the L.C.I., and contactors switching major power flow. The desired L.C.I. frequency is determined by system control. The field of the inductor motor operating in conjunction with the field excitation of the dc motor, under system control, adjusts the dc link current from the L.C.I. and, hence, the developed torque of the traction motor. Contactors are actuated by system control to reduce voltages applied to the motors during initial start-up - and for traction motor stopping or reversing functions.

Feedback Signal Processing

Motor terminal voltage and stator currents are sensed by the feedback signal processing unit to determine the motor internal operating conditions such as current magnitude $|I|$, speed M_S which synchronizes L.C.I. firing, and the feedback signal θ which represents the stator voltage-current phase angle and determines the gate firing delay α of the L.C.I.

4.7.3 Comments on the Electrical Systems

The Rectifier-Inductor Machine System shown in Figures 51 and 52 involves an on-board 3-phase diode rectifier for main power. The on-board electrical control is comparatively simple since the sole purpose of the electrical system is to provide relatively constant dc voltage to the existing drive system of a conventional tractor-trailer car, thus facilitating demonstration of the flywheel concept. All operator commands are mechanically actuated in the system. L.C.I. equipment is at the wayside and can be shared with several shuttle cars.

The L.C.I. - Inductor/Machine System (Figures 52 and 55), on the other hand, requires an L.C.I. similar to that used on the Rectifier-Inductor Machine System recharge stations on-board each vehicle. The associated control equipment is relatively complex because of the bidirectional power flow requirement. Motor fields must be under system control, and all operator commands must be electrically actuated.

Despite the added complexity of the L.C.I. - Inductor Machine System, it is capable of regenerating flywheel energy during vehicle coasting or braking, whereas the rectified system cannot. An analysis of the amount of energy available for regeneration (see Section 5.4) indicates that it is insufficient to justify the added complexity and cost of an L.C.I. on each car. Therefore, the simpler Rectifier-Inductor Machine System is used for the conceptual design.

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Section 5

FACE HAULAGE ECONOMIC ANALYSIS AND SYSTEM TRADE-OFFS

The Economic Analysis and Systems Trade-Off Section encompasses a discussion of:

- Annual operating costs of shuttle cars powered by battery, diesel, flywheel, steam as well as conventional tether

- Cost/ton analysis of tractor-trailer and conventional shuttle flywheel powered cars in two-car and three-car systems for several spin-up times and at several load capacities as compared to 2 conventional tethered shuttle cars

- System trade-off considerations including flywheel size, number and type of shuttle cars, effect of spin-up time on productivity, spin-up time versus motor size, spin-up time versus wayside equipment size, feasibility of extended tram distances, and energy recovery and regeneration

5.1 SUMMARY AND CONCLUSIONS

System Trade-Off Conclusions

- The 4.5 kW hr usable energy flywheel size is consistent with requirements for 90% of the face haulage applications and represents a conservative rating. To accommodate its size, this flywheel requires using a tractor-trailer shuttle car.

- A 3.0 kW hr flywheel size may be adequate to meet 80% of the grade and bottom conditions according to findings from a study performed by C.B. Manula. This size flywheel may be fitted in a conventional shuttle car and would provide cost per ton improvements in either two or three car systems.

- Higher car capacity, within limits, yields higher productivity.

- A conventional shuttle car has higher productivity than a tractor-trailer car of equal capacity due to turn around time, making a three tractor-trailer car system necessary for productivity benefits.

- Spin-up time required is inversely proportional to car weight and the severity of bottom conditions. Thus, the productivity gains stated for the 300#/ton bottom conditions are conservative since average bottom conditions are less severe.

- A charging motor and wayside inverter which can provide 4.5 kW hrs charge in 80 seconds is an adequate design since average spin-up time will be 30 seconds with average bottom conditions.

- A 750 kVA capacity mine power center can accommodate the charging motor load without significant effect on other mine equipment.

- Shuttle car tram distances greater than 1000 feet (round trip) are feasible with flywheel-powered shuttle cars but with some loss in productivity for three-car systems.

  A four-car system will permit about 1000 additional feet of tramming with little or no loss in productivity over a three-car system, but with some increase in cost per ton due to an additional operator, extra car, etc.

  About 1000 additional feet of tramming are feasible with a 4.5 kW hr flywheel and average bottom conditions. Poor bottom conditions would require a 7.5 kW hr flywheel.

- Most of the cars' kinetic and potential energy is consumed in overcoming tire-rolling resistance. The small amount of energy remaining for regeneration coupled with its low frequency of occurrence does not warrant any sacrafice in system cost performance for recovery.

Economic Conclusions

- Annual operating costs, exclusive of production labor, of flywheel systems compared to diesel, battery, steam, and conventional shuttle cars show that the flywheel system is equivalent or advantageous to all systems.

- The cost/ton and rate of return for the flywheel system on a tractor-trailer car are superior to the base case two conventional tethered shuttle car system when used as follows:

  With equal capacity cars in a three-shuttle car system with charge times up to 90 seconds

  With 14% greater capacity in a two-shuttle car system with a 30 second or less average charge time

- The cost/ton for a flywheel system on a conventional shuttle car (if it were possible to fit in the required size flywheel) is superior to the base case of two conventional tethered cars when used as follows:

  With equal capacities in a two-car system with average charge times of 60 seconds or less

  With equal capacities in a three-car system with charge times of 90 seconds or less

5.2 SHUTTLE CAR ANNUAL OPERATING COSTS

The annual operating costs shown in Table XXI do not show significant economic advantage or disadvantage among the various types of shuttle cars considered in this study when it is noted that the diesel car has about twice the payload of the other cars. As a result, the principal economic benefits will be largely dependent on performance as expressed in tons per shift and ultimately in the cost of coal mined in dollars per ton. The figures developed in this portion of the study are essential to trade-off considerations, operating cost item comparisons and the ultimate cost per ton comparisons.

Table XXI tabulates the most significant incremental annual operating costs for the various shuttle cars considered in this study:

- The Joy 18SC13DC tethered car used in the "base case" simulations

- The Jeffrey 404L battery-powered Ramcar

- The Jeffrey 410H diesel powered Ramcar

- A flywheel-powered car based on modifying a Jeffrey 404L car (tractor-trailer car)

- The Jeffrey developmental steam car

Table XXI

Shuttle Car Annual Operating Costs

		TYPE OF CAR				
COST ITEM/DOLLARS/YEAR	(NOTES)	TETHER	BATTERY	DIESEL	FLYWHEEL	STEAM
Basic Car	(1)	$8,000	$8,400	$11,100	$10,000	13,000
Inverter	(2)	-	-	-	530	-
Power Center	(3)	220	70	-	920	-
Fuel Store & Handl. Equipt.	(4)	-	-	500	-	500
Basic Car Maint.	(5)	4,030	4,030	4,030	4,030	4,030
Cable Maint. & Replace	(6)	5,800	-	-	-	-
Battery Maint. & Replace	(7)	-	6,000	-	-	-
Engine Maint.	(8)	-	-	12,800	-	1,500
Flywheel Maint.	(9)	-	-	-	1,400	-
Electric Power Cost	(10)	1,170	1,780	-	1,540	-
Diesel Fuel Cost	(10)	-	-	1,400	-	1,550
Total		$19,220	$20,280	$29,830	$18,420	$20,580

Notes:
1. See Shuttle Car Specifications, Appendix A.1
   Battery car includes two battery sets and charger.
2. Based on 203 kW ("80 sec") spin-up motor, see Inverter Costs, Appendix A.2.
3. Based on 203 kW spin-up time, see Power Center Costs, Appendix A.3.
4. Estimated.
5. Includes supplies and labor, from data provided by PSU, Appendix B.
6. Cable Maintenance & Repair, Appendix A.6.
7. Battery Maintenance and Replacement, Appendix A.7.
8. Engine Maintenance, Appendix A.8.
9. Flywheel Maintenance, Appendix A.9.
10. Energy Costs, Appendix A.10.

The detailed cost data used in developing these numbers is developed in Appendix A. Considering that the 404H diesel car has roughly twice the payload capacity of the base case, the ± 5% spread in costs of the other cars shown is probably within the accuracy of the estimates, especially for the developmental steam car and the conceptual flywheel car. The relatively narrow spread of these costs results from the added complexity and hence cost of the flywheel and steam powered cars balanced by cable repair and replacement for the tethered car, battery set replacement for the battery-powered car, and engine maintenance and overhaul for the diesel-powered car.

Other than the initial purchase price of the cars and the outstanding replacement items just mentioned, the next most significant cost item is basic car maintenance. This is assumed to be equal for all cars since it includes such items as: tire replacement, lubrication, maintenance and repair of drive train components (transmission, differential, wheels, bearings), steering, and unloading mechanisms.

Finally, although the operating cost of energy to power the shuttle cars is not very significant on a dollars per ton basis, it is seen to represent some 5 to 8% of the annual operating cost of the shuttle cars and may be expected to increase in the future.

5.3 PRODUCTIVITY OR COST EFFECTIVENESS TRADE-OFFS

The cost/ton for the cases considered are shown in Table XXII. To evaluate the impacts of spin-up motor size, wayside power capability, car load capacity, and to provide cost comparisons of other types of cars such as diesels, battery and steam, General Electric conducted parametric cost per ton evaluations. C.B. Manula made a mine economic analysis which in part displayed cost/ton effectiveness. There is remarkably good correlation between the cost effectiveness estimated by C.B. Manula and associates and those estimated by General Electric despite the use of somewhat different assumptions regarding base costs and depreciation periods. While the cost/ton differences seem small, $.29/ton improvement yields $290,000/yr for a 1,000,000 ton/yr mine.

Some initial conclusions are immediately evident from the summary table:

- With two- car face haulage systems, the average spin-up time must be kept as low as possible, preferably below the 30 second unload time, to achieve a recognizable cost improvement.

- Three-car face haulage systems are economically viable and permit greater freedom for longer spin-up times without severe economic penalties.

Table XXII

Cost Effectiveness Summary

	GENERAL ELECTRIC		C.B. MANULA	
	Cost/Ton	Improvement	Cost/Ton	Improvement
Base Case*, (Tethered)	$16.48		$16.92	
Tractor-Trailer Cars (Flywheel Powered)				
2 Cars, 30 sec	16.53	(.05)	17.04	(.12)
2 Cars, 60 sec	16.75	(.27)	17.28	(.36)
2 Cars, 90 sec	17.13	(.65)		
3 Cars, 30 sec	16.35	.13	16.78	.14
3 Cars, 60 sec	16.38	.10	16.82	.10
3 Cars, 90 sec	16.41	.07		
Conventional Cars* (Flywheel Powered)				
2 Cars, 30 sec	15.84	.64	16.40	.52
2 Cars, 60 sec	16.20	.28	16.78	.14
2 Cars, 90 sec	16.55	(.07)		
3 Cars, 30 sec	15.87	.61	16.28	.64
3 Cars, 60 sec	15.87	.61	16.28	.64
3 Cars, 90 sec	16.00	.48		
Load Capacity Effects** (Flywheel Powered)				
2 Tractor-trailer cars, 200 ft^3	17.01	(.53)		
2 Tractor-trailer cars, 236 ft^3	16.53	(.05)		
2 Tractor-trailer cars, 270 ft^3	16.19	.29		
2 Conventional Cars, 200 ft^3	16.36	.12		
2 Conventional Cars, 236 ft^3	15.93	.55		
2 Conventional Cars, 270 ft^3	15.68	.80		

* All at 236 ft^3 capacity.
**All at 30 second spin-up time.
() Brackets denote negative numbers or loss from base case.

- Conventional shuttle cars are more economically attractive than tractor-trailer cars, if an adequately sized flywheel could be fitted.

- The greater the load carrying capacity, within reasonable limits, the greater the economic benefits. If this notion were carried to extremes, changes in car weight, traction energy requirements, and spin-up times would have to be factored into the calculations.

5.3.1 Alternate Primary Power Cars

Table XXIII is a summary of the cost-per-ton performance for the various types of primary motive power cars considered in

Table XXIII

Cost Effectiveness Alternate Power Systems

	COST/ TON	IMPROVEMENT ($/TON)
Tethered Shuttle Car, Base Case	$16.48	-
Battery-Powered Ramcar	16.17	.31
Diesel-Powered Ramcar	15.55	.93
Steam-Powered Ramcar	16.66	(.18)
Flywheel-Powered Tractor-Trailer Car	16.19	.29

this study. All are for two-car face haulage systems; all assume a 30 second unloading time during which the flywheel car is recharged. The base case tethered car includes a one minute delay at 40% frequency for cable repairs. All except the base case are tractor-trailer cars and include 2 x 0.25 minute delays per trip for turn around.

Even though the numbers in Table XXIII look like significant differences the temptation to draw any sweeping general conclusions from them should be resisted because:

- The battery-powered Ramcar appears to be an economically viable alternative. The unknown reservation here is how many months of operation before battery performance deteriorates to the point where it cannot get through one shift without recharge.

- Diesel-powered Ramcars look attractive because of their very high capacity. However, the requirement for increased ventilation and Organized Labor's opposition to their use must also be recognized.

- The slight economic disadvantage of the steam car could probably be reversed with a slightly higher payload capacity and if the car were unloaded in 15 seconds at its maximum discharge rate.

- The flywheel-powered tractor-trailer car is economically attractive. A flywheel-powered conventional shuttle car would be even more attractive because of the elimination of turn around time.

5.4 SYSTEM TRADE-OFF DISCUSSION

5.4.1 Flywheel Size

As long as the flywheel has adequate capacity to meet the energy requirements of its "worst case" mission profile, the actual size of the flywheel and hence its cost have an insignificant effect on economic performance. This is because incremental changes in flywheel cost (as much as 2 to 1) will be divided by 10 years expected life times 440 shifts per year times 327 (plus or minus) tons per shift, or 1.44×10^6. So a change in flywheel cost of $10,000 times two cars will only amount to a little over a penny a ton. Previous studies, covered in Section 2 - Mission Analysis and Energy Storage Requirements, led to the conclusion that a flywheel sized to provide 4.5 kWhrs of usable energy would meet 80% to 90% of actual mine conditions. This conclusion was based on bottom conditions of 5% grade and 300 pounds per ton rolling resistance. Since then an independent study was conducted for the U.S. Bureau of Mines by C.B. Manula and associates. The study surveyed bottom conditions in 600 sections of underground coal mines in Appalachia. The average of the bottom conditions reported was 2.07% grade and 165 pounds per ton rolling resistance. These results lend credibility to the 4.5 kWhr flywheel size conclusion.

However, flywheel size does have a definite influence on the selection of the type of car and this will be discussed next.

5.4.2 Car Options

Two conclusions regarding the selection of a car for optimum productivity emerge from a detailed consideration of productivity covered in Section 5.5.4. First, within limits the greater the payload capacity of the car, the better its productivity. This is principally due to reduced miner wait time since the higher capacity results in fewer shuttle car trips and therefore fewer waits.

Second, the conventional shuttle car has higher productivity than a tractor-trailer car. This is because the tractor-trailer must turn around at each end of its tram which adds to miner wait time in two-car face haulage systems.

Unfortunately, conventional shuttle cars have evolved to a state of very compact, space-efficient designs. There is no way to fit a 4.5 kWhr flywheel into one of these cars without a major redesign of the car which is not the purpose of this study. An alternative would be to elect to cover a lesser range of bad bottom conditions and opt for a 3 kWhr useful energy flywheel. It is believed that this size flywheel may be fitted into the cable reel compartment of many present tethered shuttle car designs. There is powerful argument for electing this option since it would permit retrofitting many existing operational tethered shuttle cars with higher productivity flywheel power.

A 4.5 kWhr flywheel will fit in the battery, or engine, compartment of most present tractor-trailer car designs. In this case it is necessary to go to a three-car face haulage system to achieve productivity improvements. Three car systems are, of course, quite practical without the hindrance of the tether cable and are used in a few mines equipped with diesel cars. Based on the information available, the Jeffrey 404L battery Ramcar is recommended for consideration for the installation of a 4.5 kWhr flywheel.

Conventional shuttle cars may also be operated in a three-car configuration. In this case, productivity gains would be even greater than three tractor-trailers or two conventional cars with the longer spin-up times.

5.4.3 Productivity Versus Spin-Up Time

Productivity begins to decrease as soon as the spin-up time exceeds the car unload time for two car systems. This is because the added spin-up time tends to add, in direct proportion, to the miner wait time thereby reducing its productivity. Spin-up time is directly proportional to car weight and the severity of the bottom conditions. The productivity and cost analysis shown in Section 5.3 are predicated on rather severe bottom conditions of 5% grade and 300 pounds per ton rolling resistance. A recent study cited earlier in this section indicates that average bottom conditions are only about half as severe as those used in the simulation. This means that the calculated spin-up times are higher than the actual average and hence the productivity gains as stated are conservative.

Spin-up time is also determined by the size of the spin-up motor, and this subject will be discussed next.

5.4.4 Motor Size Versus Spin-Up Time

Obviously the greater the capacity of the spin-up motor, the faster it can replenish a given amount of energy to the flywheel. The size of the motor has been selected at 203 kW so that, acting as an alternator, it provides the peak traction load requirements of the car in accelerating, jogging, and tramming. A 203 kW motor will require 80 seconds to replenish 4.5 kWhrs of energy in the flywheel. However, even with severe bottom conditions the average energy consumption per trip will only require 63 seconds for recharge. If average bottom conditions are assumed (165 pounds/ton and 2.07% grade), the average spin-up time will be less than 30 seconds, as shown in Section 5.5 - still with the 203 kW motor. In view of these results there seems to be little incentive to increase charging motor capacity. If it were desirabl to increase charging motor capacity to reduce maximum spin-up times with bad bottom conditions, the increased weight of a few hundred pounds would be trivial compared with the 27,000 or so pounds of basic car weight. As shown under Flywheel Size, Section 5.4.1, increased cost would have a negligible effect on cost per ton. Increased physical size and increased rotor stresses would increase design problems. Increased motor capacity would yield increased productivity. This has not been quantified since the present computer simulator program is not equipped to generate wait times which are proportional to energy consumption on the prior trip.

5.4.5 Wayside Equipment Cost Versus Spin-Up Time

A standard size 750 kVA capacity mine power center can accommodate a 203 kW surge load for the spin-up motor without any noticeable effect on other mine equipment operating from the same power center. The same statement is true up to about 250 kW of peak load. At 300 kW and higher it is necessary to think in terms of a separate power center dedicated exclusively to shuttle car power requirements. In round numbers this would mean about $60,000 for the power center, lead-in cable, and added inverter cost or, dividing by 1.44×10^6 tons in 10 years, about 4 cents per ton. Three hundred kW of spin-up capacity would reduce the 63 second average spin-up time (with bad bottom conditions) to 43 seconds. The 20 second reduction reflected in less miner wait time and added productivity will more than cover the 4 cents added cost. The major trade-off will be in the added design difficulty of physically accommodating the larger spin-up motor.

5.4.6 Extended Tram Distances

There is a significant benefit to the operation of untethered shuttle cars which has not been addressed. Relieved of the restriction of a finite length tether cable and assuming adequate on-board energy storage, tram distances greater than the nominal 500 foot tether are realizable. Extended tram distances could provide added flexibility in mine operations as follows.

It would be possible to extend a cut block by many cuts and thereby delay extending a secondary haulage conveyor belt until a convenient time such as a weekend. It is understood that a conveyor belt move requires about one shift. Therefore, productivity could be enhanced by one shift's coal production per cut block.

In retreat mining it is believed that it is common practice to keep the conveyor belt as far from the mining area as possible and to move it as seldom as possible. In some mining operations two tethered cars are used in a piggyback mode to achieve these goals. Obviously retreat mining productivity could be improved (under these conditions) if extended shuttle car tram distances were feasible.

To a certain extent the number of entries in a cut block is limited by the shuttle car's tether length. In some mining operations, particularly those with poor roof conditions requiring a lot of roof bolting, it might be advantageous to drive more than 6-8 entries simultaneously. Again extended shuttle car tram distances would be helpful. While it is not the intent of this study to delve very deeply into the field of mining engineering, the flexibility of route selection and extended tram distances offered by internally powered shuttle cars may increase options in mining practice.

Recognizing these possible advantages, a study has been made to determine the feasibility of operating untethered shuttle cars over tram distances exceeding the normal tethered car capabilities of approximately 500 feet.

For this investigation the six-entry cut plan described in Section 2.3 has been extended as shown in Figure 57. Tram paths to cuts A and B represent the shortest and longest tran distances employed by the PSU/USBM simulations in determining the shuttle car mission profile, energy storage requirements, and other performance factors. The tram distance to cut C represents an extension of about two times the distance to cut B (2200 feet round trip). The exact configuration of the cut plan, e.g. the number of entries, and the tram paths are unimportant to this discussion; tram distance is the significant factor. Tram distances are measured from the discharge point to the cut. It is assumed that the physical length of the miner and shuttle car are about equal to the distance required to turn a tractor-trailer car around at each end of the run.

For the purposes of this discussion the following assumptions have been made:

Unloaded car weight = 26,000 lbs
Loaded car weight = 40,000 lbs
Average bottom = 165 lbs/ton and 2% grade
Poor bottom = 300 lbs/ton and 5% grade
Overall efficiency = 60%
Standby losses = 250 W hrs
Spin-up motor = 200 kW
Tram speed unloaded, good bottom = 420 ft/min (4.8 mph)
Tram speed loaded, good bottom = 380 ft/min (4.3 mph)
Tram speed unloaded, bad bottom = 340 ft/min (3.9 mph)
Tram speed loaded, bad bottom = 300 ft/min (3.4 mph)

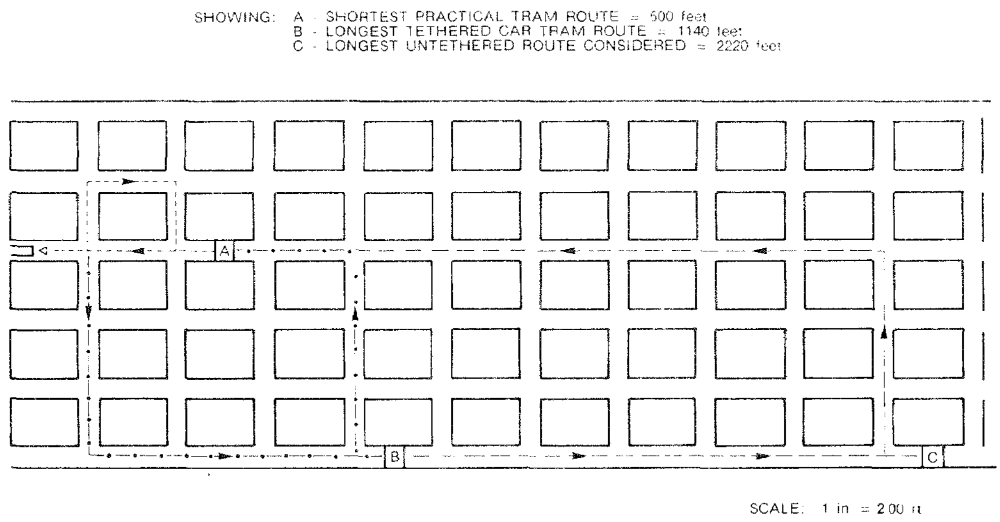


Figure 57. Extended Six-Entry Cut Plan

The assumptions on car performance characteristics are derived from the most recent information available from Jeffrey Mining Machinery Division of Dresser Industries. The detailed calculations of energy storage and spin-up time requirements are shown in Table XXIV.

The results of these calculations, spin-up time and energy storage required versus distance are plotted in Figures 58 and 59. The detailed calculations of car mission cycle times are shown in Table 58. The total tram time (only) versus the round trip tram distance is plotted in Figure 60.

In order to determine the impact of these extended tram and spin-up times on mine productivity, shuttle car mission cycles are staggered and plotted against elapsed time. The cycles are shown for three cases: three cars operating in good bottom conditions (Figure 61), three cars in bad bottom conditions (Figure 62), and four cars in good bottom conditions (Figure 63). While this is an idealized picture, the results of miner wait times show good correlation with the PSU/USBM Simulator outputs at those points where comparisons may be made. It should be noted that shuttle car total cycle times could be improved by increasing the capacity of the spin-up motor and wayside equipment. However,

Table XXIV

Energy Storage and Spin-Up Time Requirements

Empty drag, good bottom = 26,000 lbs x 165/2000 = 2145 lbs

Empty drag, bad bottom = 26,000 lbs x 300/2000 = 3900 lbs

Loaded drag, good bottom = 40,000 lbs x $(\frac{165}{2000} + 0.02)$ = 4100 lbs

Loaded drag, bad bottom = 40,000 lbs x $(\frac{300}{2000} + 0.05)$ = 8000 lbs

GOOD BOTTOM	500 Feet		1140 Feet		2220 Feet	
	10^6 ft lbs	kW*	10^6 ft lbs	kW*	10^6 ft lbs	kW*
Tram Empty	0.536	0.20	1.22	0.46	2.38	0.90
Tram Full	1.030	0.39	2.34	0.88	4.56	1.72
Total	1.57	0.59	3.56	1.34	6.94	2.62
Total/η = 0.6		0.98		2.23		4.36
Waiting losses		0.25		0.25		0.25
Total		1.23		2.48		4.61
Spin-up Time (secs)		22		47		82
BAD BOTTOM						
Tram Empty	0.98	0.37	2.22	0.84	4.33	1.63
Tram Full	2.00	0.75	4.56	1.72	8.88	3.34
Total	2.98	1.12	6.78	2.56	13.21	4.97
Total/η = 0.6		1.87		4.27		8.28
Waiting losses		0.25		0.25		0.25
Total		2.12		4.52		8.53
Spin-up Time (secs)		38		81		154

*2.656 x 10^6 ft lbs = 1 kW hr

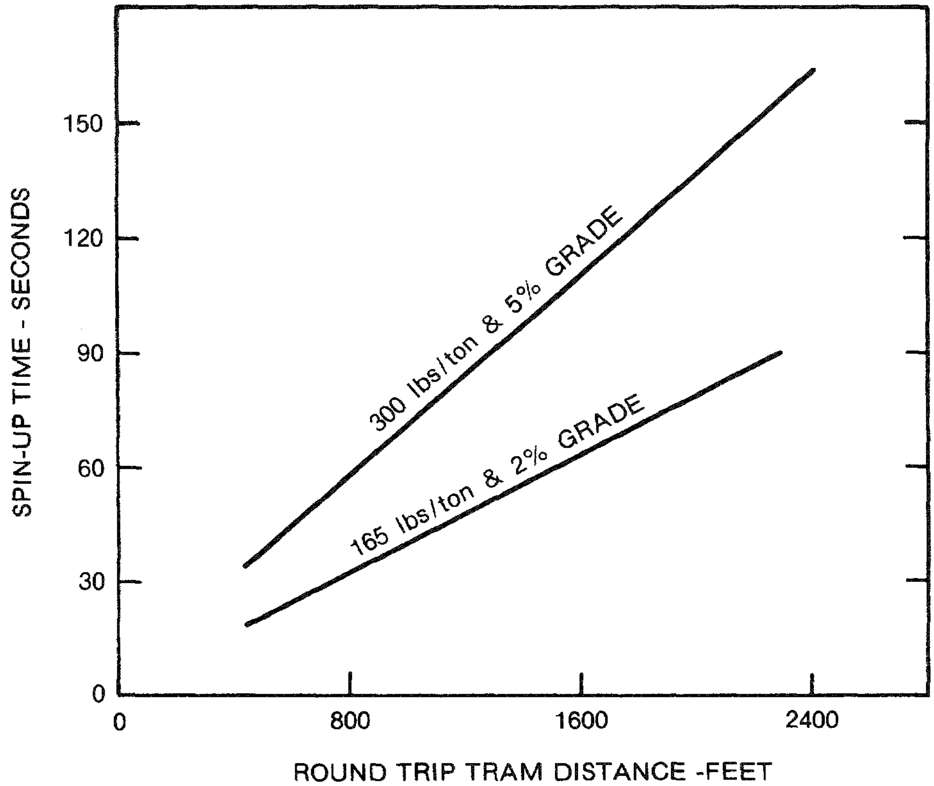


Figure 58. Spin-Up Time Versus Total Tram Distance

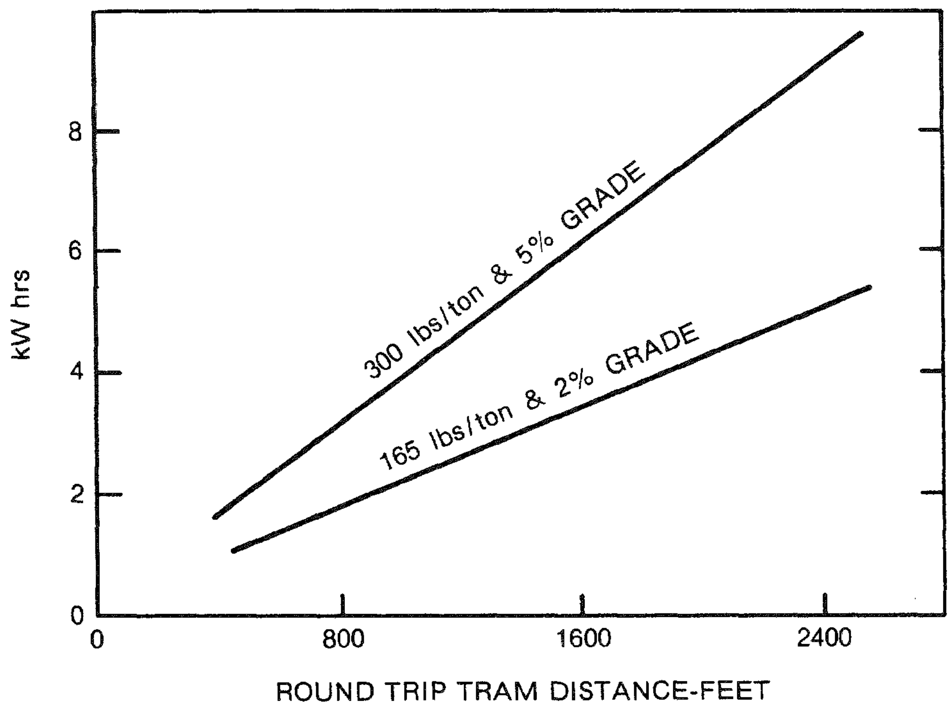


Figure 59. kW hrs Required Versus Total Tram Distance

Table XXV

Car Mission Cycle Times

Good Bottom

500 ft Tram	Load	= 1.00 min
	Tram loaded 180 ft @ 380 ft/ min + 0.17 turn around	= 0.64 min
	Unload and Spin-up, 22 secs	= 0.37 min
	Tram unloaded 320 ft @ 420 ft/min + 0.17 t.a.	= 0.93 min
1140 ft Tram	Load	= 1.00 min
	Tram loaded 570 ft @ 380 ft/min + 0.17 turn around	= 1.67 min
	Unload and Spin-up, 47 secs	= 0.78 min
	Tram unloaded 570 ft @ 420 ft/min + 0.17 t.a.	= 1.53 min
2220 ft Tram	Load	= 1.00 min
	Tram loaded 1110 ft @ 380 ft/min + 0.17 turn around	= 3.09 min
	Unload and Spin-up, 82 secs	= 1.48 min
	Tram unloaded 1110 ft @ 420 ft/min + 0.17 t.a.	= 2.81 min

Bad Bottom

500 ft Tram	Load	= 1.00 min
	Tram loaded 180 ft @ 300 ft/min + 0.17 turn around	= 0.77 min
	Unload and spin-up, 38 secs	= 0.63 min
	Tram unloaded 320 ft @ 340 ft/min + 0.17 t.a.	= 1.11 min
1140 ft Tram	Load	= 1.00 min
	Tram loaded 570 ft @ 300 ft/min + 0.17 t.a.	= 2.07 min
	Unload and spin-up, 81 secs	= 1.35 min
	Tram unloaded 570 ft @ 340 ft/min + 9.17 t.a.	= 1.85 min
2220 ft Tram	Load	= 1.00 min
	Tram loaded 1110 @ 300 ft/min + 0.17 t.a.	= 3.87 min
	Unload and spin-up 154 secs	= 2.57 min
	Tram unloaded 1110 @ 340 ft/min + 0.17	= 3.43 min

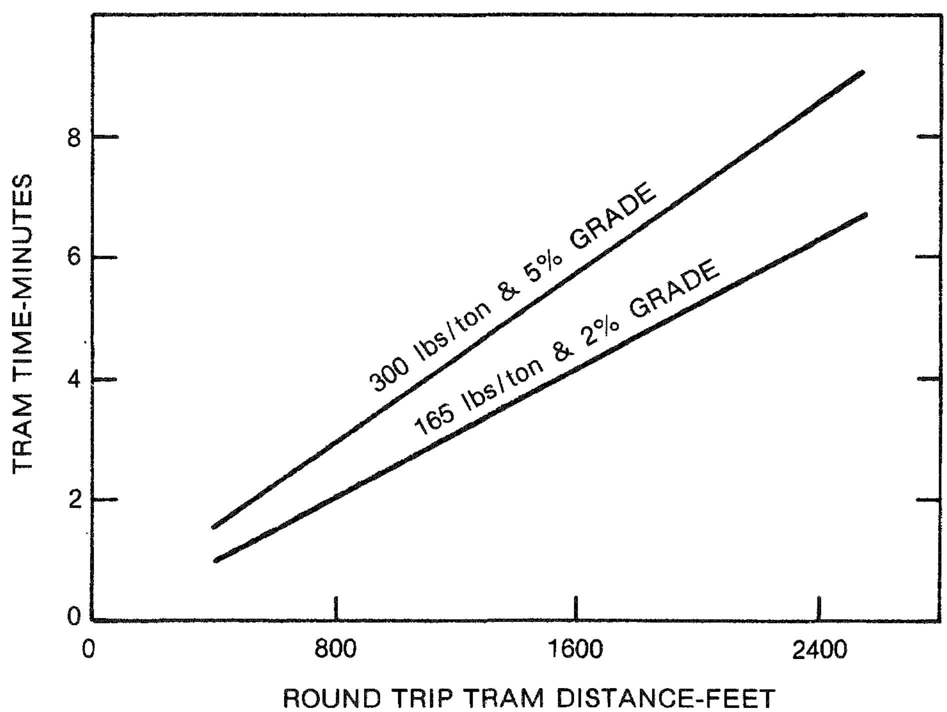


Figure 60. Total Tram Time Versus Total Tram Distance

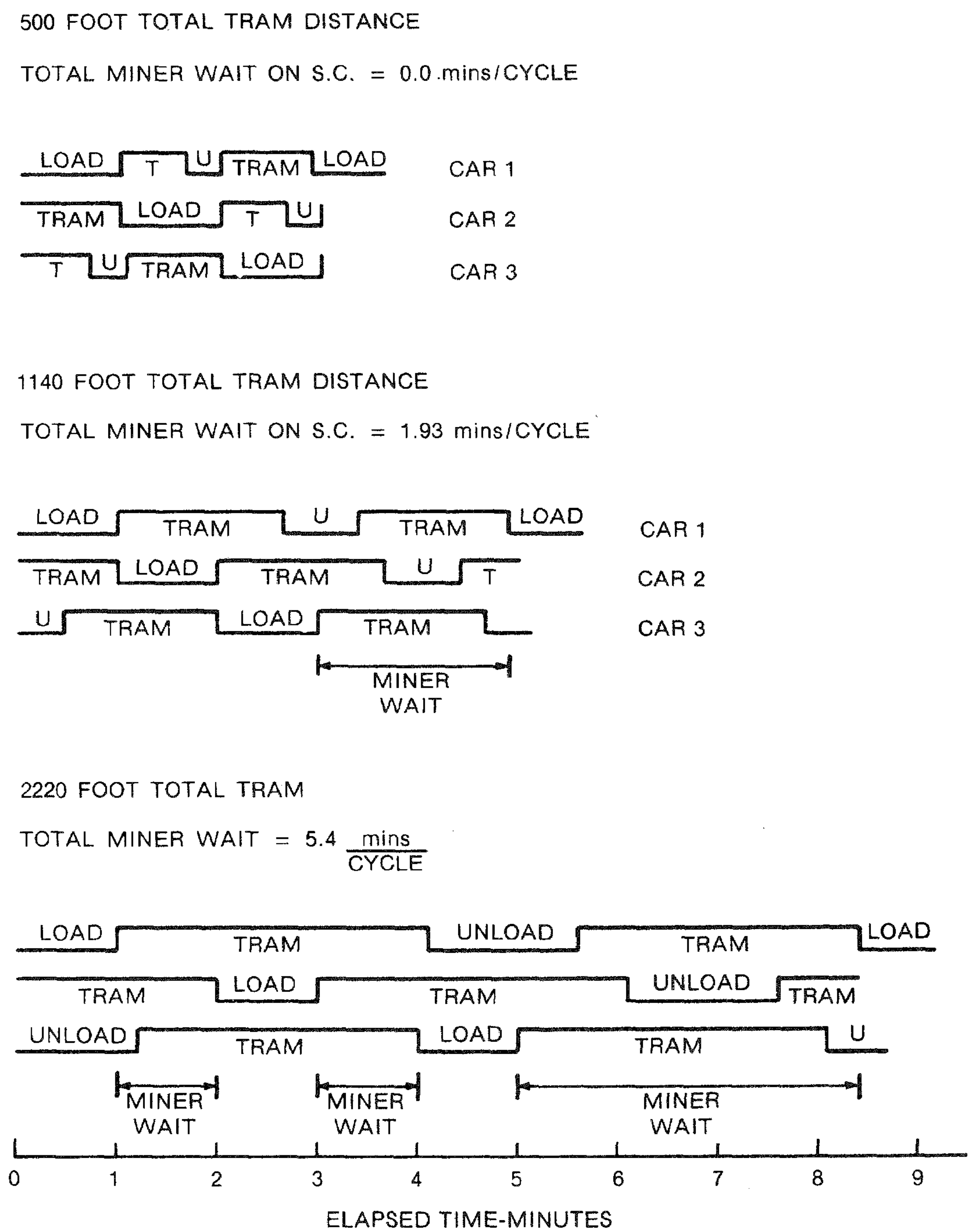


Figure 61. Shuttle Car Cycle Times for 3 Cars on Average Bottom

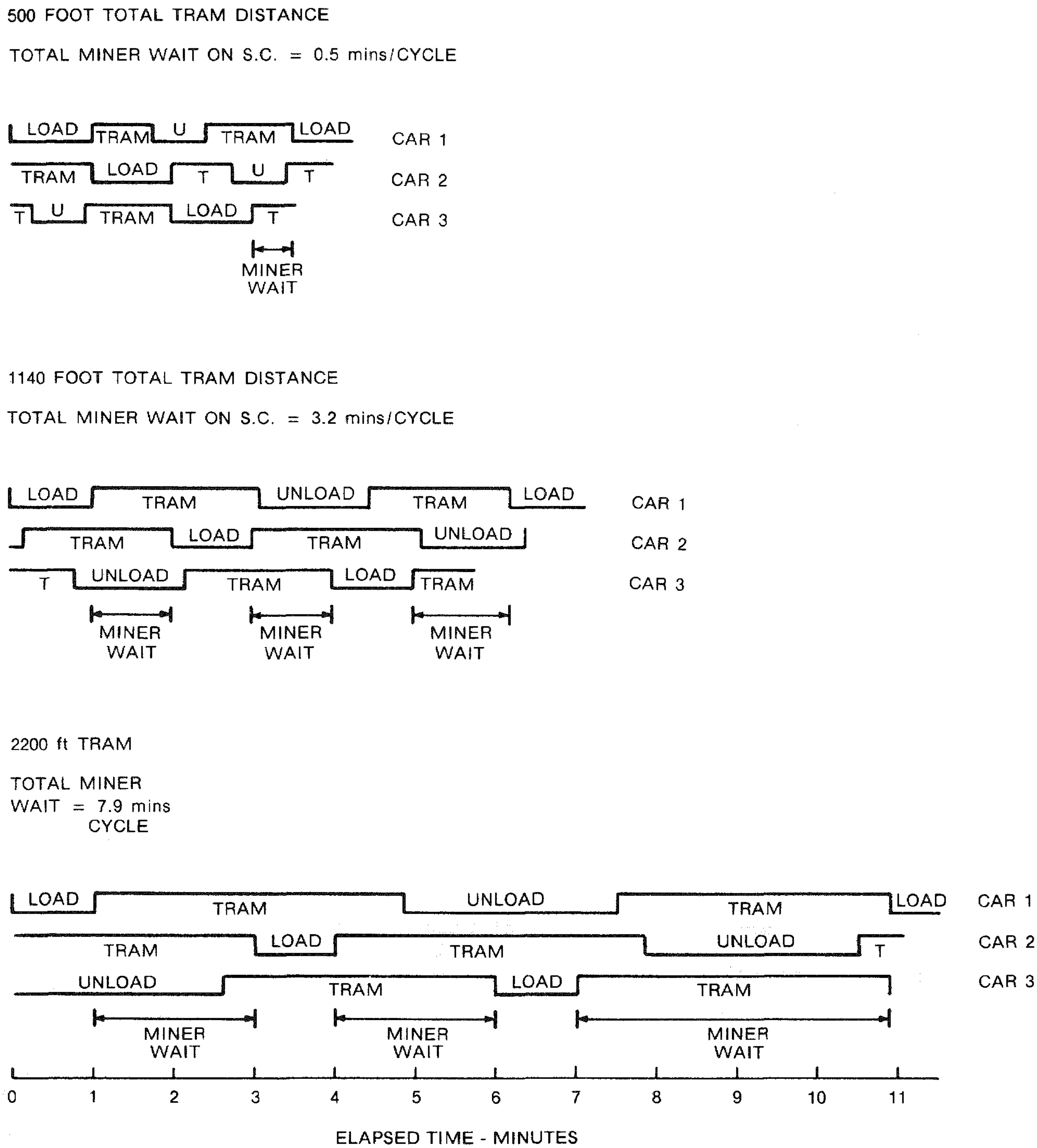


Figure 62. Shuttle Car Cycle Times for 3 Cars on Bad Bottom

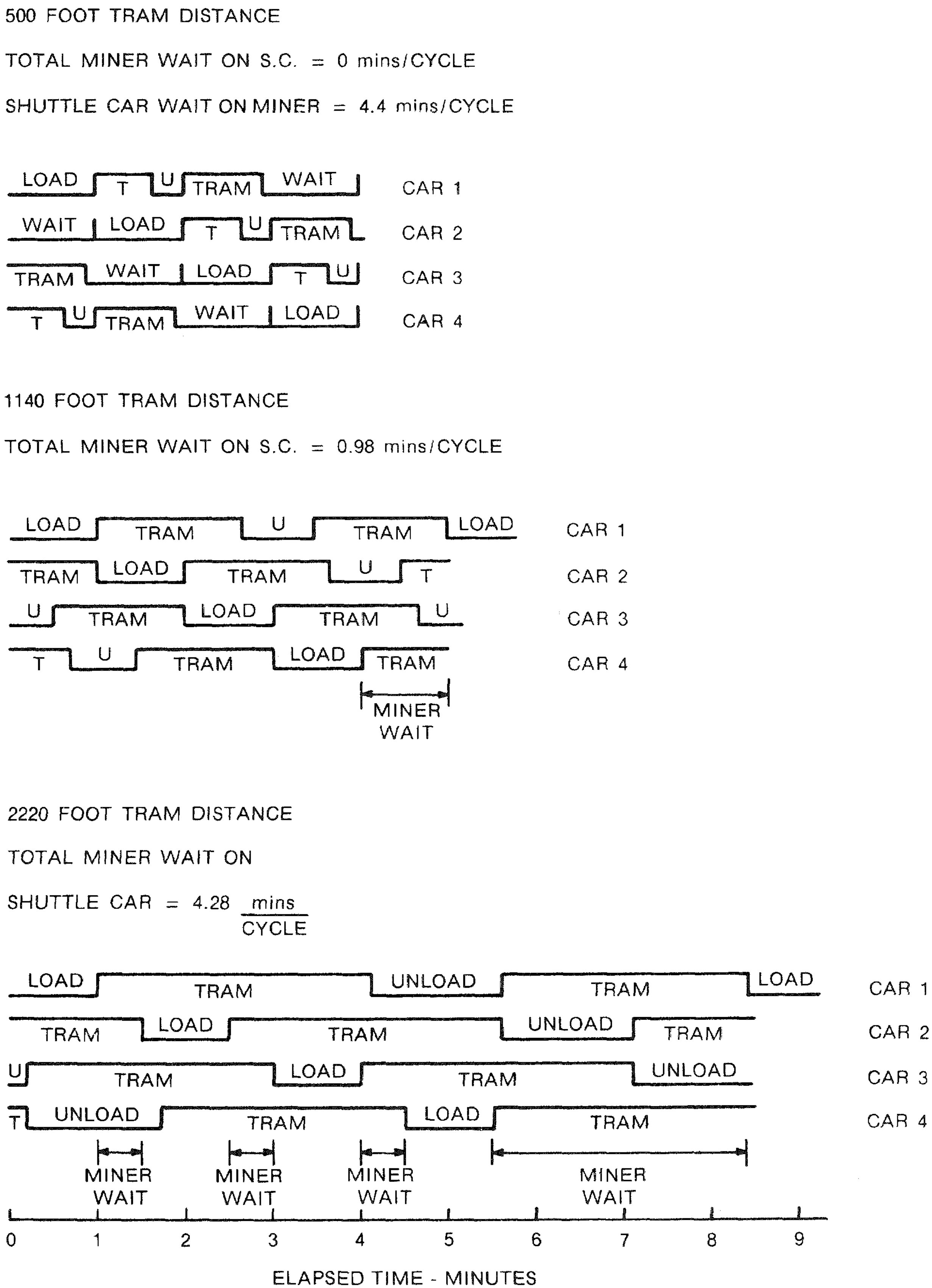


Figure 63. Shuttle Car Cycle Times for 4 Cars on Bad Bottom

the numbers of Table XXV show that the spin-up time only represents some 15 to 20% of the total cycle time so doubling the spin-up motor and wayside capacities would only improve the total cycle time some 10% at best.

For tram distances of 500 to 1000 feet, round trip, a two-car system will complete about 30 cycles per shift, a three-car system will run about 20 cycles per shift and it can be assumed that a four-car system will run about 15 cycles per shift. At tram distances greater than 1000 feet the number of cycles are unknown so for the purposes of this discussion the 20 and 15 cycle numbers will be used. Production will decrease whether the reduction is attributed to increased miner wait time, fewer shuttle car cycles, or some combination.

Figure 64 shows plots of increased miner wait time, or loss of production, as a function of extended tramming distances for the three cases studied.

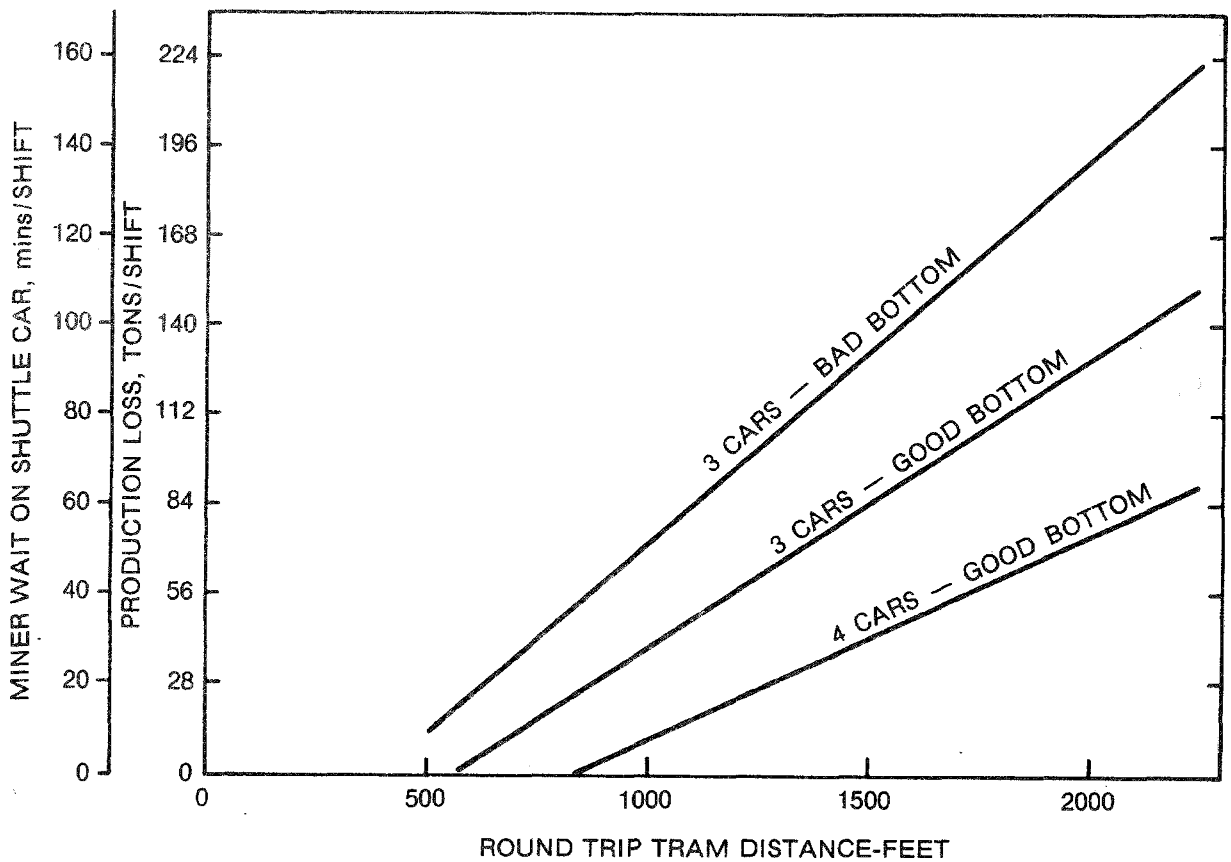


Figure 64. Miner Wait and Production Loss Versus Tram Distance

The results of this analysis show that shuttle car tram distances in excess of 1100 feet are feasible with some loss in productivity for three-car systems.

A four-car system will permit about 1000 additional feet of tramming with no significant loss in productivity over a three-car system. However, the cost per ton will be increased due to the extra shuttle car operator and cost of the car. It is important to note that about 800 to 1000 additional feet of tramming are feasible with a 4.5 kW hr flywheel and average bottom conditions of 165 lbs/ton rolling resistance and 2% grade or less. With bad bottom conditions, 300 pounds per ton rolling resistance and 5% grade, extending tramming distances are only feasible with a flywheel of about 7.5 kW hrs to accomodate 1000 extra feet of tramming.

5.4.7 Energy Recovery and Regeneration

An investigation was undertaken to determine the practicality of energy recovery and flywheel regeneration from deceleration and downhill braking. Any regenerative scheme, of course, implies a bi-directional drive train. While these are quite common, the bi-directionality does impose some design limitations particularly with fluid clutches and transmissions.

For the purposes of this discussion a gross vehicle weight of 40,000 lbs which is roughly the size of the cars under consideration is assumed. The kinetic energy of motion is:

$$K.E. = 1/2\ mv^2$$

where: $m = w/g - 40{,}000/32$, in $\frac{\text{lb sec}^2}{\text{ft}}$

$$v = 6.67 \text{ ft/sec } (400 \text{ ft/min, } 4.5 \text{ mph})$$

$$K.E. = 27{,}800 \text{ ft lbs or } \underline{0.01 \text{ kW hr}}$$

Taken by itself this is not enough energy to be worthy of any serious efforts at recovery.

The potential energy available from a downhill grade is:

$$P.E. = \text{weight} \times \text{distance} \times \text{grade}$$

With the average grade of 2.07% (Section 8.4.1) over an entire 500-foot tramming path, the following equation results:

$$P.E. = 40{,}000 \times 500 \times 0.0207 = 4.14 \times 10^5 \text{ ft lbs} = 0.16 \text{ kW hr}$$

Results of a study of mine bottom conditions conducted by C.B. Manula indicate that in approximately 8% of the mine sections a 6% grade might be encountered, in which case:

$$P.E. = 40{,}000 \times 500 \times 0.06 = 1.2 \times 10^6 \text{ft lbs} = 0.45 \text{ kW hr}$$

To recover this energy, it is first necessary to overcome the vehicle losses. The first and most significant of these is rolling resistance. It may be recalled that when estimating flywheel energy requirements it was assumed that a tire-rolling resistance of 300 pounds per ton represented the worst-case bottom condition, 165 pounds per ton was determined to be the average, and approximately 50 pounds per ton the best of rolling resistances to be found. The energy lost in tire rolling resistance under average conditions is:

$$\text{Rolling Loss} = 165 \text{ lbs/ton} \times 20 \text{ tons} \times 500 \text{ ft} = 1.65 \times 10^6 \text{ ft lbs}$$

$$= 0.62 \text{ kW hr}$$

Hence, under average bottom conditions more than the potential energy available from a downhill grade is consumed in overcoming rolling resistance with none left over to regeneratively charge the flywheel. It should be noted that this potential energy is not lost; it represents a saving in energy required from the flywheel, even though there is no energy available for regeneration.

Assuming the best bottom conditions which only occur about 14% of the time according to the same study of mine bottom conditions conducted by C.B. Manula, the losses due to rolling resistance are:

$$\text{Rolling Loss} = 50 \text{ lbs/ton} \times 20 \text{ tons} \times 500 \text{ ft} = 500{,}000 \text{ ft lbs}$$

$$= 0.19 \text{ kW hr}$$

Deducting this from the best potential energy estimate (0.45 kW hr) leaves 360 Watthours available. This must be taken back through the 70% efficiency of the drive train for a potential regeneration energy of 252 Watthours.

If the flywheel is directly coupled mechanically to the drive train and the drive train may be made bi-directional at no extra cost, then certainly whatever residual energy is available from downhill travel could be recaptured. If the flywheel is electrically coupled through a motor-alternator, then to achieve bi-directionality each car must be equipped with a load commutating inverter. Furthermore, the losses of the L.C.I. and spin-up motor must be taken into account which further reduces the energy available for regeneration to about 204 Watthours.

All the most favorable conditions were assumed for the preceding discussion and at best only 7% of the average trip energy (3 kW hr) might be recovered in something less than 8% of the sections.

The foregoing analysis indicates that the amount of kinetic energy available from car velocity, 10 Watthours from a 4.5 mph speed, is insignificant.

The amount of potential energy available from down grade travel is quite small, 200-250 Watthours per trip, when all losses are accounted for. Even this much energy is only available in less than 8% of the applications.

If the vehicle drive train can be made bi-directional at no sacrifice in cost or performance, then regeneration should be employed.

The maximum amount of energy available for regeneration coupled with its frequency of occurrence do not warrant any sacrifice in system cost or performance for energy recovery.

5.5 FLYWHEEL SPIN-UP TIME REQUIREMENTS

5.5.1 Requirements for Conservative Design

Section 2 discusses a number of different mine operating conditions including differing seam heights and grades which were modeled in the PSU/USBM Underground Mine computer simulator. The purpose of these computer simulations was to establish a "base case" mission profile and flywheel energy storage requirements. The results of this work led to the selection of a "base case" consisting of a cut-plan in a six-foot seam with poor bottom conditions (300 pounds per ton rolling resistance) and a 5% uphill grade for all loaded shuttle car tramming routes. In addition to the tramming energy requirements extracted from the computer simulation printout, a constant auxiliary load (for hydraulic steering, lights, etc.) of some 5 to 10 horsepower during all shuttle car waiting periods was also included. A discharge (or unload) energy of 220 Watthours and flywheel windage losses of 70 to 90 Watthours were also added. The sums of all these energy requirements and the times to recharge (or spin-up) the flywheel, based on a 203 kW (or "80 second") motor, are shown for each shift in Table XXVI. The 203 kW motor/alternator was sized to meet maximum alternator requirements for power supplied to the shuttle car during acceleration, jogging, and tramming. The "80 second" nomenclature is derived from the time required by this motor to replenish 4.5 kW hrs of energy to the flywheel. The numbers from the simulator in Table XXVI clearly show that under these conditions in Shift 17 the total energy used is 4.47 kW hrs, a recharge time of 79 seconds is required. However, it may also be noted that spin-up times range from 46 to 79 seconds with an average of 63 seconds. Since tram distances are not shown on the computer printout, the energy used to overcome the 5% grade was calculated as a mean tramming distance between 200 and 500 feet times the grade times the laden weight of the car, divided by the assumed 70% drive efficiency and converted to kW hrs (=0.36). This grade climbing energy is held constant throughout all the successive calculations.

All the initial mission profile simulations were made using tethered shuttle cars. The presence of the tether cable has little effect on the energy requirements of the car, but tons-per-shift

Table XXVI

Energy Usage (in kW hrs) from Base Case Simulation
(Joy 18SCI3DC in 6-foot seam)

SHIFT	300#/T ROLLING RESISTANCE	5% GRADE	TOTAL TRAM ENERGY	WAITING LOSSES	TOTAL CAR ENERGY	TOTAL FLYWHEEL ENERGY*	SECS. CHARGE TIME**
1	1.27	.36	1.63	0.98	2.61	2.89	51
2	1.42	.36	1.78	1.10	2.88	3.19	57
3	0.87	.36	1.23	1.10	2.33	2.58	46
4	1.16	.36	1.52	1.09	2.61	2.89	51
5	1.31	.36	1.67	1.08	2.75	3.04	54
6	2.16	.36	2.52	1.07	3.59	3.97	70
7	1.94	.36	2.30	1.06	3.36	3.72	66
8	2.11	.36	2.47	1.25	3.72	4.12	73
9	1.64	.36	2.00	1.00	3.00	3.32	59
10	1.89	.36	2.25	1.16	3.41	3.77	67
11	1.67	.36	2.03	1.08	3.11	3.44	61
12	1.54	.36	1.90	1.18	3.08	3.41	60
13	1.71	.36	2.07	1.05	3.12	3.45	61
14	1.94	.36	2.30	1.11	3.41	3.77	67
15	2.62	.36	2.98	1.06	4.04	4.47	79
16	2.36	.36	2.72	1.08	3.80	4.22	75
17	2.62	.36	2.98	1.00	3.98	4.41	78
					Average	3.57	63

*Assumes 90% alternator efficiency.
**With 203 kW (80 sec) charging motor
Potential Energy due to grade = $\frac{200 \text{ ft} + 500 \text{ ft}}{2}$ x .05 x (26700 + 11800) = 673,750 @ .70 drive efficiency = 962,500 ÷ 2,656,000 = 0.36 kW hr average

productivity is reduced due to the delays caused by cable breakage. To simulate flywheel shuttle cars the computer simulation program was changed to eliminate cable delays, to add spin-up time delays, and to permit the use of three shuttle cars per mine section. The addition of a third car was predicated on the belief that three cars would reduce miner wait-time leading to increased productivity and reduced cost per ton. This assumption subsequently proved to be valid. Twelve additional computer simulation runs were made with the conditions tabulated in Table XXVIII.

Table XXVII

Untethered Shuttle Car Simulators

Untethered Shuttle Car Simulations

CASE	NO. OF CARS	TYPE	SPIN-UP DELAY (SEC)
1	2	Tractor-Trailer	30
2	2	Tractor-Trailer	60
3	2	Tractor-Trailer	90
4	3	Tractor-Trailer	30
5	3	Tractor-Trailer	60
6	3	Tractor-Trailer	90
7	2	Conventional	30
8	2	Conventional	60
9	2	Conventional	90
10	3	Conventional	30
11	3	Conventional	60
12	3	Conventional	90

The results of these simulations were analyzed to determine miner wait on shuttle car times, number of shuttle car trips and tons of coal mined, all on a per shift basis. An example of one of these analyses is shown in Table XXVIII.

The results of these analyses are most easily appreciated by a discussion of the plots of Miner Wait on Shuttle Car versus Spin-Up Time shown in Figure 65. The first point of significance is an improvement in miner wait time of approximately 10 minutes per shift of conventional shuttle cars compared to tractor-trailer cars. This is due to the additional 15 seconds required at each end of the tram to turn the tractor-trailer cars around. The second point of interest is the improvement of three shuttle car operations over two cars, especially at the longer spin-up times. It is possible to operate 3 untethered cars effectively due to the absence of cable interference and hence the cars may tram on a circular route. The final point is the relative flatness, or little change in miner wait time, of the three-car systems regardless of spin-up time (at least up to 90 seconds). This is due to the fact that at least one of the cars is usually empty and ready to service the miner. It is anticipated that if the spin-up delay were extended much beyond 90 seconds, say to 105 seconds, the miner wait time would start to increase more rapidly.

The data generated is also utilized to develop Tons Produced Versus Miner Wait Time shown in Figure 66. With minor perturbations, all 12 cases examined fall on a straight line indicating a miner operating rate of 1.4 tons per minute or 84 tons per hour. The graph also highlights the importance of miner waits imposed by any and all shuttle car delays.

Table XXVIII

Simulation Results
Two Untethered Shuttle Cars, 30 Second Spin-Up Delay

SHIFT	MINER WAIT ON S.C. TIME IN MINS	NO. OF S.C. TRIPS	TONS PRODUCED
1	47.4	65	362
2	49.3	56	310
3	40.9	67	371
4	45.6	60	334
5	50.6	55	307
6	55.3	62	346
7	52.7	56	309
8	50.4	55	306
9	56.9	56	310
10	54.5	61	326
11	65.2	58	326
12	57.6	55	307
13	55.7	63	346
14	55.1	55	307
15	61.5	62	346
16	58.0	59	326
17	60.5	57	306
18	62.1	55	302
19	55.7	44	243
Average	54.2	58.7	324.8

5.5.2 Requirements for Average Operating Conditions

While the numbers in Table XXVI are useful for assessing the worst case of conservative energy storage requirements of the flywheel, for economic comparisons, they should be tempered to more accurately reflect actual average operating conditions. It has been shown that productivity in tons per shift and hence economic advantages in dollars per ton are highly dependent on flywheel spin-up time. Thus, it is important that the flywheel be spun-up while it is unloading, hence the 220 Watthours of unloading energy will be provided directly from the wayside power equipment and not from the flywheel. Secondly, it has been proposed that waiting time losses due to auxiliary loads (primarily hydraulic pumps for steering, conveyor elevator or ram, etc.) be reduced by automatically disengaging them after say a 20 second delay. Such auxiliary loads would automatically be re-engaged as soon as any

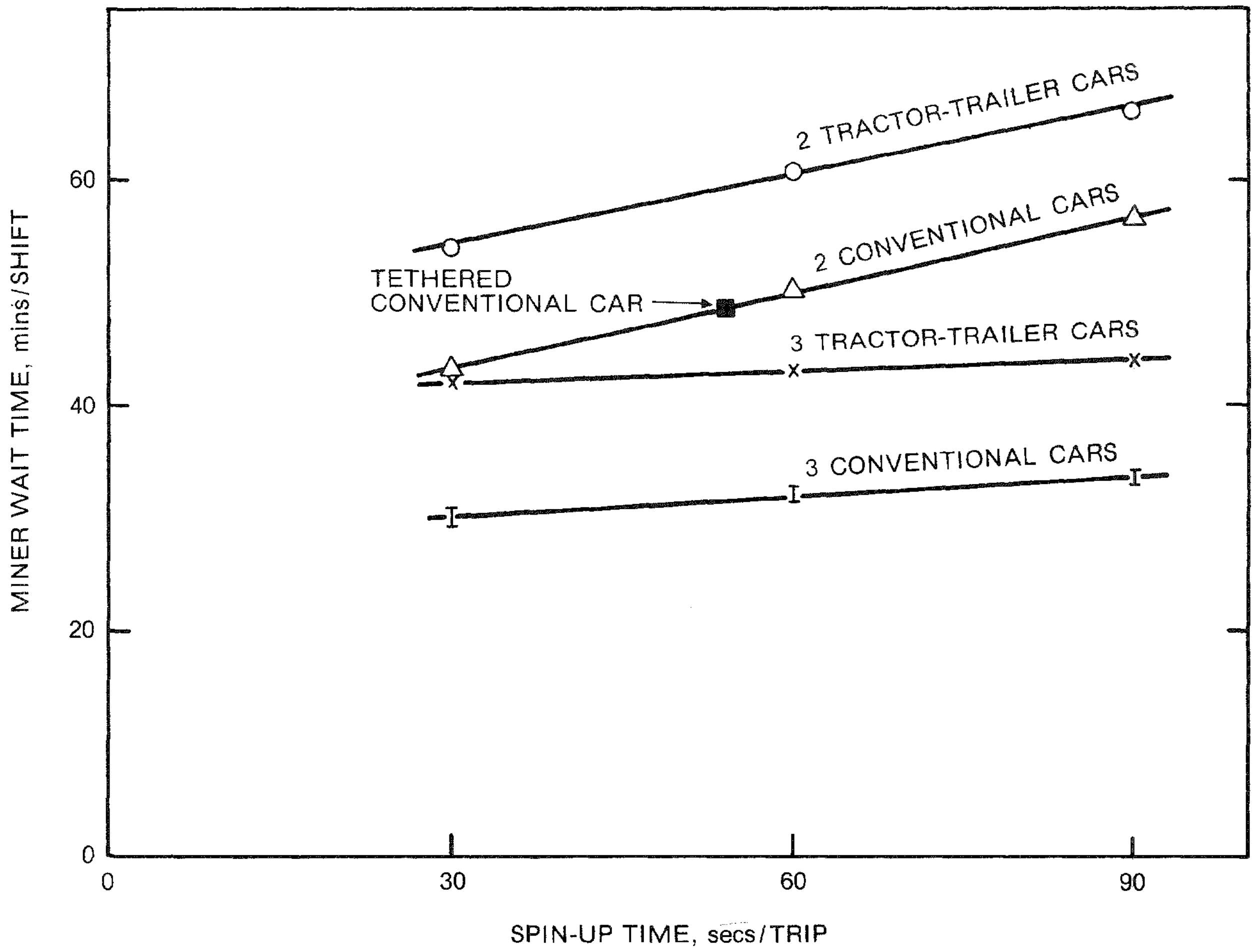


Figure 65. Miner Wait on Shuttle Car Versus Spin-Up Time

control is activated. It is estimated that such a system could reduce stand-by, or waiting time, losses by 70%. Both of these energy savings are reflected in the numbers shown in Table XXIX. The 300 pounds per ton rolling resistance has been retained, as have the energy requirements of the 5% grade. It will be noted that the maximum flywheel energy storage requirements have dropped to about 3.6 kW hrs and the average spin-up (or charge) time is now 47 seconds.

For a conservative estimate of the maximum energy storage requirements of the flywheel, 300 pounds rolling resistance over the entire distance of all tramming routes in the cut plan is not an unreasonable value. However, in actual practice it seems unlikely that such poor bottom conditions would persist over all entries in all cut plans. One might expect 100 to 200 feet of bad bottom in one or two entries or even bad bottom for 1 or 3 entire cut plans out of 10. To assess the effects of less severe bottom conditions, in terms of averages, a rolling resistance of 200 pounds per ton is assumed for the figures in Table XXX. Here it can be seen that the maximum energy requirement has decreased to 2.6 kW hrs and the average charge time is down to 35 seconds with a spread of 24 to 46 seconds.

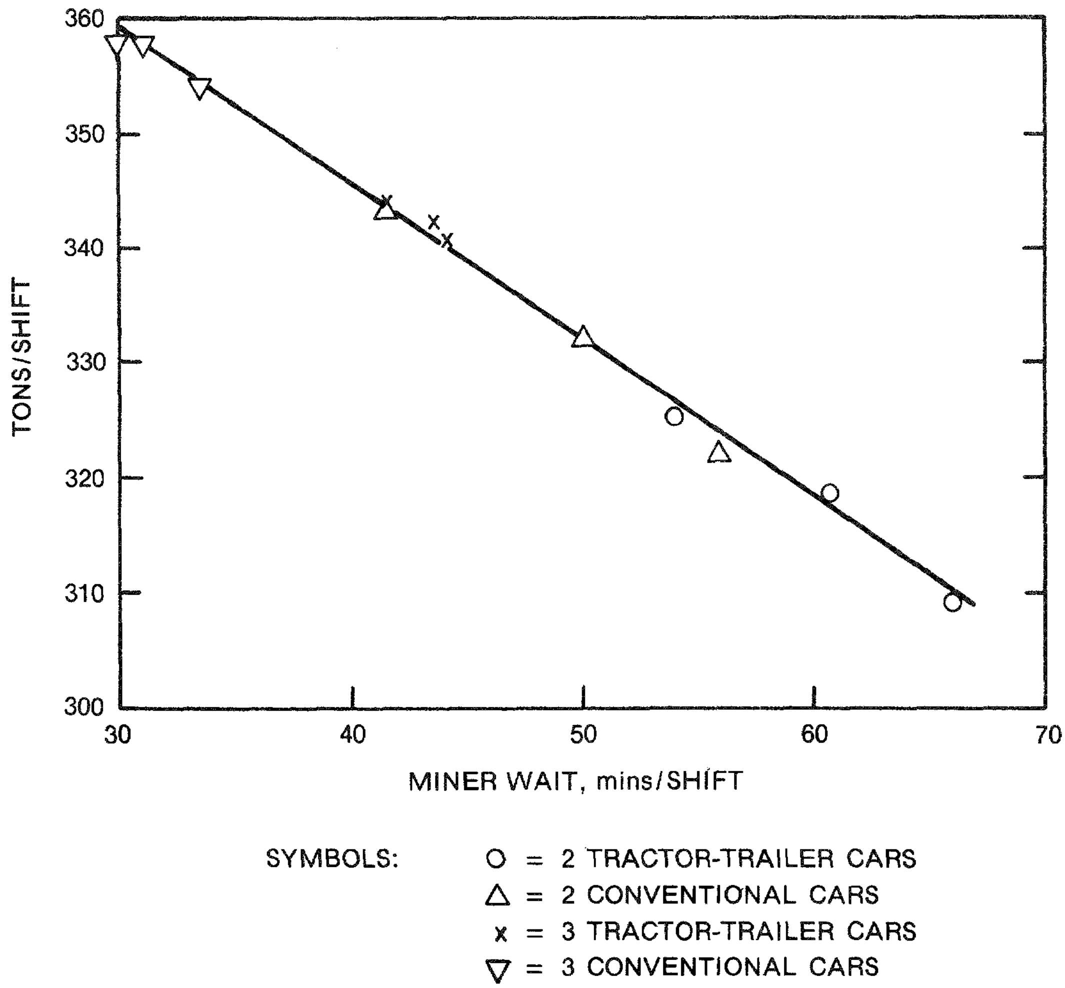


Figure 66. Productivity (Tons/Shift) Versus Miner Wait on Shuttle Car

In like manner the rolling resistance was further decreased to 100 pounds per ton and the results are shown in Table XXXI. In this case the energy storage requirement is down to 1.64 kW hrs and the average charge time is 24 seconds.

The effects of rolling resistance on spin-up time are depicted in Figure 67. Note that the curve is not quite a straight line and it does not intersect the origin. The residual spin-up time, or energy replacement, at zero rolling resistance is due to flywheel windage loss and auxiliary power requirements.

Figure 68 shows the frequency distribution of spin-up times for 17 shifts of the cut plan. Spin-up time values for the 200 pounds per ton rolling resistance case are plotted. Since all of the other energy requirements, or spin-up times, were derived from the same base case data the basic character of the frequency distribution will not change greatly for the other rolling resistance cases.

Table XXIX

Energy Usage in kW hrs from Base Case Simulation
(Assuming: Charge while unloading and reducing stand-by losses 70%)

SHIFT	300#/T ROLLING RESISTANCE	5% GRADE	TOTAL TRAM ENERGY	WAITING LOSSES	TOTAL CAR ENERGY	TOTAL FLYWHEEL ENERGY*	SECS CHARGE TIME**
1	1.27	.36	1.63	.23	1.89	2.07	37
2	1.42	.36	1.78	.26	2.04	2.27	40
3	0.87	.36	1.23	.26	1.49	1.66	29
4	1.16	.36	1.52	.26	1.78	1.98	35
5	1.31	.36	1.67	.26	1.93	2.14	38
6	2.16	.36	2.52	.25	2.77	3.08	55
7	1.94	.36	2.30	.25	2.55	2.83	50
8	2.11	.36	2.47	.31	2.78	3.09	55
9	1.64	.36	2.00	.23	2.23	2.48	44
10	1.89	.36	2.25	.28	2.53	2.81	50
11	1.67	.36	2.03	.26	2.29	2.54	45
12	1.54	.36	1.90	.29	2.19	2.43	43
13	1.71	.36	2.07	.25	2.32	2.58	46
14	1.94	.36	2.30	.27	2.57	2.86	51
15	2.62	.36	2.98	.25	3.23	3.59	64
16	2.36	.36	2.72	.26	2.98	3.31	59
17	2.62	.36	2.98	.23	3.21	3.57	63
					Average	2.66	47

* Assumes 90% alternator efficiency
**With 203 kW (80 sec) charging motor

Table XXX

Energy Usage in kW hrs from Base Case Simulation
(Assuming: Charge while unloading and reducing stand-by losses 70% and 200#/T rolling resistance)

SHIFT	200#/T ROLLING RESISTANCE	5% GRADE	TOTAL TRAM ENERGY	WAITING LOSSES	TOTAL CAR ENERGY	TOTAL FLYWHEEL ENERGY*	SECS CHARGE TIME**
1	0.85	.36	1.21	.23	1.44	1.60	28
2	0.95	.36	1.31	.26	1.57	1.74	31
3	0.58	.36	0.94	.26	1.20	1.33	24
4	0.77	.36	1.13	.26	1.39	1.54	27
5	0.87	.36	1.23	.26	1.49	1.66	30
6	1.44	.36	1.80	.25	2.05	2.28	40
7	1.29	.36	1.65	.25	1.90	2.11	37
8	1.41	.36	1.77	.31	2.08	2.31	41
9	1.09	.36	1.45	.23	1.68	1.87	33
10	1.26	.36	1.62	.28	1.90	2.11	37
11	1.11	.36	1.47	.26	1.73	1.92	34
12	1.03	.36	1.39	.29	1.68	1.87	33
13	1.14	.36	1.50	.25	1.75	1.94	34
14	1.29	.36	1.65	.27	1.92	2.13	38
15	1.75	.36	2.11	.25	2.36	2.62	46
16	1.57	.36	1.93	.26	2.19	2.43	43
17	1.75	.36	1.22	.23	2.34	2.60	46
					Average	2.00	35

* Assumes 90% alternator efficiency
**With 203 kW (80 sec) charging motor

Table XXXI

Energy Usage in kW hrs from Base Case Simulation
(Assuming: Charge while unloading and reducing stand-by losses 7% and 100#/T rolling resistance)

SHIFT	100#/T ROLLING RESISTANCE	5% GRADE	TOTAL TRAM ENERGY	WAITING LOSSES	TOTAL CAR ENERGY	TOTAL FLYWHEEL ENERGY*	SECS CHARGE TIME**
1	.42	.36	0.78	.23	1.01	1.12	20
2	.47	.36	0.83	.26	1.09	1.21	21
3	.29	.36	0.65	.26	.91	1.01	18
4	.39	.36	0.75	.26	1.01	1.12	20
5	.44	.36	0.80	.26	1.06	1.18	21
6	.72	.36	1.08	.25	1.33	1.48	26
7	.65	.36	1.01	.25	1.27	1.41	25
8	.70	.36	1.06	.31	1.37	1.52	27
9	.55	.36	0.91	.23	1.14	1.27	23
10	.63	.36	0.99	.28	1.27	1.41	25
11	.56	.36	0.92	.26	1.18	1.31	23
12	.51	.36	0.87	.29	1.16	1.29	23
13	.57	.36	0.93	.25	1.18	1.31	23
14	.65	.36	1.01	.27	1.28	1.42	25
15	.87	.36	1.23	.25	1.48	1.64	29
16	.79	.36	1.15	.26	1.41	1.57	28
17	.87	.36	1.23	.23	1.46	1.62	29
					Average	1.35	24

*Assumes 90% alternator efficiency
**With 203 kW (80 sec) charging motor

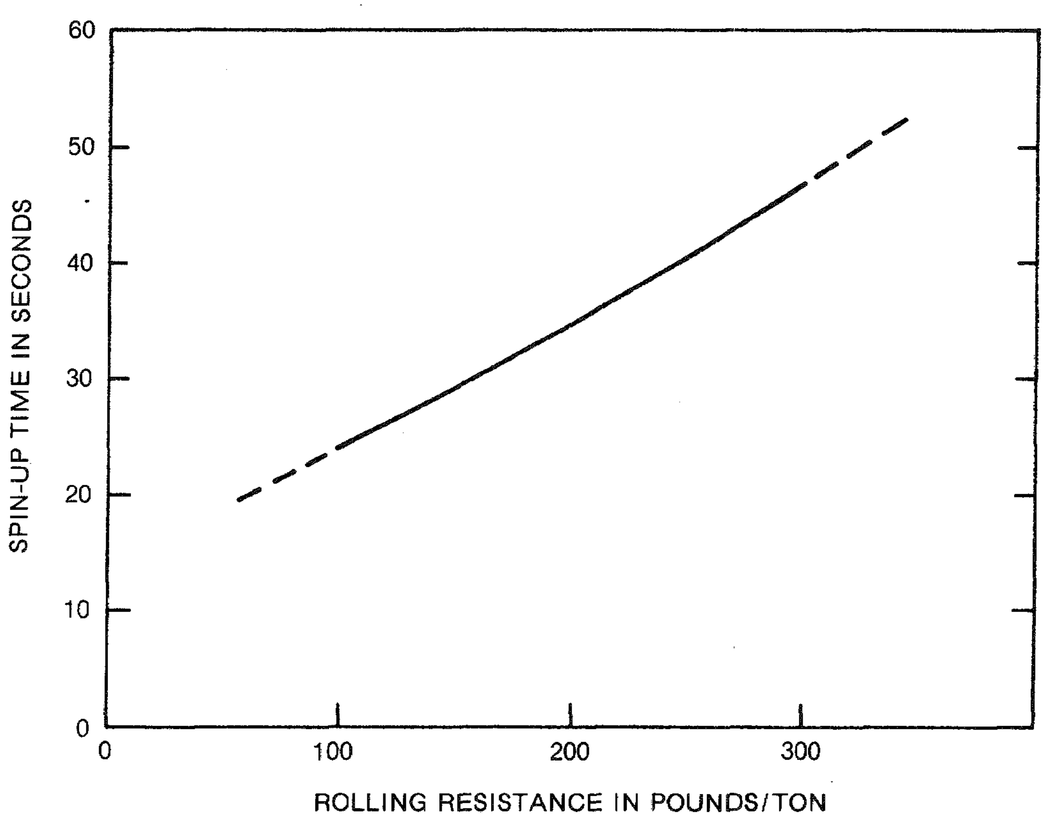


Figure 67. Effect of Rolling Resistance on Flywheel Spin-Up Time

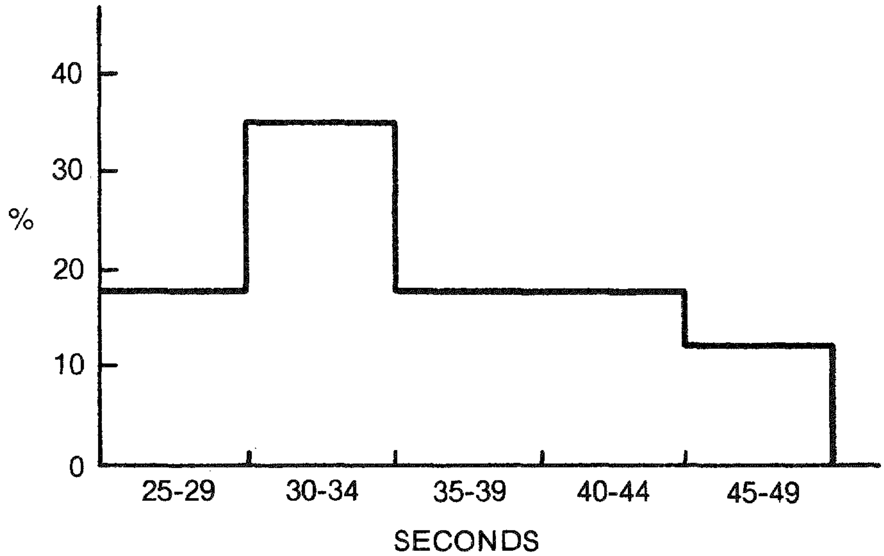


Figure 68. Distribution of Spin-up Times (200 lbs/Ton Rolling Resistance)

5.5.3 Use of 2-1/2 Cars

It was suggested that there might be an economic advantage to be realized from starting into a new cut block with two shuttle cars in operation and then at some later time adding a third car. The theory of the case is that early in the development of the cut block, tramming distances are short, and energy usage is low. Therefore, spin-up times are short and the third car really is not needed. In a mine with more than one or two active sections, third car operators could be "floated" from section to section to reduce miner wait times caused by longer tramming times with attendant longer spin-up times.

To investigate the feasibility of this scheme it is only necessary to tabulate miner wait times for two and three-car systems as is done in Table XXXII. A study of the numbers indicates a significant productivity gain for three cars for every shift except Shift 5 for 30 and 60 second spin-up times. The only workable plan might be to operate the first five shifts of the 30 second spin-up case with only 2 cars and then add the third car in Shift 6. This would result in an average loss of 6 tons per shift for the first 5 shifts. Based on this performance, the concept was not studied further.

Table XXXII

Miner Wait Times for Two and Three Car Operations

	(Miner Wait Times in/Shift)					
	30 S. SPIN-UP		60 S. SPIN-UP		90 S. SPIN-UP	
Shift	2 Cars	3 Cars	2 Cars	3 Cars	2 Cars	3 Cars
1	47.4	35.2	53.6	35.2	62.5	36.3
2	49.3	46.5	52.4	46.5	60.8	47.9
3	40.9	34.0	46.6	34.0	54.6	36.9
4	45.6	42.0	49.9	42.0	48.5	43.6
5	50.6	55.1	50.1	55.1	59.6	55.1
6	55.3	37.8	64.9	40.3	67.7	42.2
7	52.7	40.8	58.6	40.2	66.3	41.1
8	50.4	43.3	53.4	45.7	60.7	45.6
9	56.9	32.8	67.2	33.4	68.1	37.0
10	54.5	41.3	65.2	39.9	71.6	41.5
11	65.2	50.3	71.1	51.4	67.6	54.5
12	57.6	36.1	62.4	38.0	74.0	40.7
13	51.7	44.9	57.4	43.9	63.5	43.2
14	55.1	52.6	54.6	54.1	59.5	55.4
15	61.5	33.8	70.9	37.6	67.0	41.8
16	58.0	44.1	64.2	47.1	77.5	43.1
17	60.5	39.0	68.3	41.3	71.3	46.7
18	62.1	30.3	65.0	36.7	73.2	47.8
19	55.7		77.2		77.6	
20			8.5		60.9	

5.5.4 Shuttle Car Payload Versus Productivity

Thus far, all productivity simulations have been based on the maximum payload capacity of the Joy 18SC13DC shuttle car used in the PSU/USBM simulator. Its payload capacity is 236 cubic feet, 11,800 pounds. At least two other cars are potential candidates for flywheel installation: a car of the capacity of the Jeffrey steam car chassis at 220 cubic feet (or 11,000 pounds) and a car of the capacity of the Jeffrey 404L battery car at 269 cubic feet (or 13,450 pounds). To investigate the effects of shuttle car payload capacity on productivity, the following calculations were made.

If it is assumed that traction motor power is adequate to maintain a 440 foot per minute tram speed and that higher capacity will result in fewer shuttle car trips, then a further assumption can be made that miner wait on shuttle car times will be reduced in direct proportion to the increase in payload capacity. With these assumptions and the productivity versus miner wait time plot of Figure 66 curves of productivity in tons per shift versus payload capacity, Figure 69, was constructed. The curves are exaggerated by the choice of scale factors to permit easy reading. Only the curves for two conventional and two tractor-trailer cars, both with 30 second spin-up times, are shown. Curves for two car cases with longer spin-up times would have the same general characteristics, but at lower productivities. Since miner wait time has already been reduced to near minimum relizable limits with 3 car systems, these cases were not studied though it is believed that even there increased payload capacity will yield some increase in productivity. The curves show decreasing gains in productivity as capacity is increased since miner wait time can not be completely eliminated. Computer simulation runs confirm that the general slope and shape of these curves are correct and that the assumptions used to calculate them lead to conservative results. In other words, the effect of shuttle car payload capacity on productivity is even greater than shown.

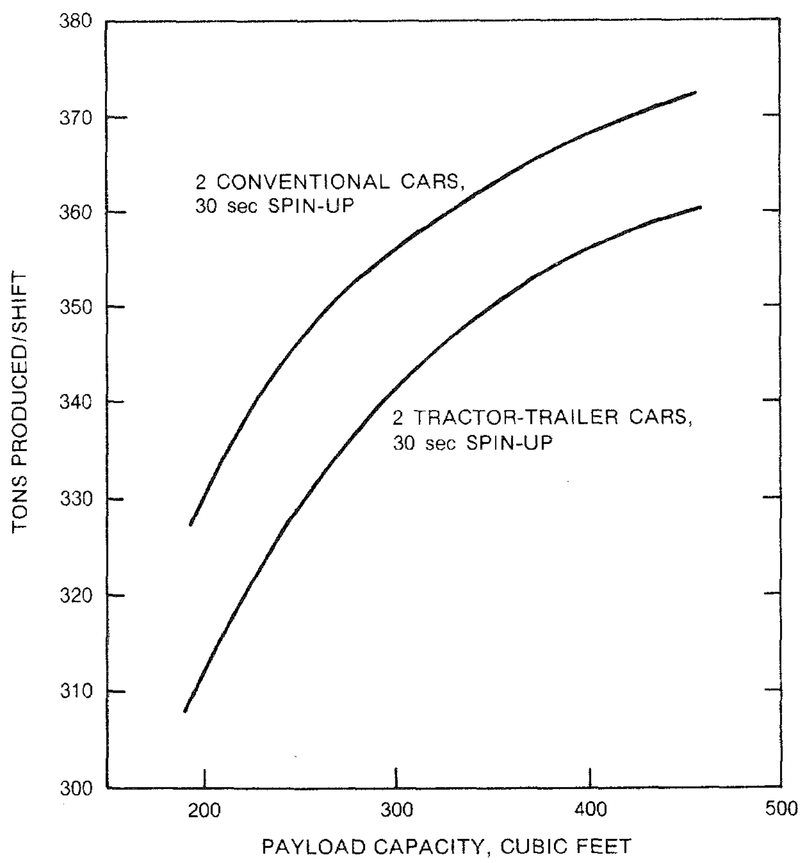


Figure 69. Productivity Versus Payload Capacity

5.5.5 Effect of Spin-up Time on Power Center

The power required to spin-up a flywheel may place a heavy peak load on the normal mine supply. Figure 70 is a plot of the average number of kilowatt hours per trip required for tramming only versus the average tramming time per trip as recorded in the PSU/USBM "base case" simulation. The linearity of this plot makes it quite reasonable to assume that the tramming distance is directly proportional to tramming time. There are some "overhead" energy losses while waiting and loading which must be added to the tramming energy (Section 2). The total kilowatt hours used per trip, including the "overhead" energy versus the average tramming time per trip, is shown on the lower plot of Figure 71. This represents the electrical energy delivered from the flywheel to the car load. To establish spin-up station capacity and its power consumption, these load energy requirements must be increased by the efficiency factors of the flywheel output generator, the charging motor, and inverter. These assumed efficiencies, which differ to a small degree from data derived from more detailed work, are:

- Generating efficiency $\eta 1 = 92.1\%$
- Motor efficiency $\eta 2 = 91.9\%$
- Inverter efficiency $\eta 3 = 90.0\%$

This equation is:

$$\text{Input kW hrs} = \frac{\text{Load kW hrs required}}{\eta 1 \times \eta 2 \times \eta 3}$$

The results of factoring these efficiencies to determine the spin-up energy requirements are shown on the upper plot of Figure 71.

The spin-up time is of major importance in determining the peak load requirements on the spin-up station and the mine power center. The peak load required by the flywheel charging equipment becomes:

$$\frac{\text{kW hrs/Trip} \times 60 \text{ min/hr}}{\text{mins Charge Time}} \times \frac{1}{\eta 1 \times \eta 2 \times \eta 3} = \text{kW Load}$$

The impact of charge time on power center capacity is most easily appreciated by plotting peak load capacity required versus charge time as in Figure 72. Here the capacity requirements are bounded by 3 and 6 kW hrs, the minimum and maximum charges required as defined by the "base case" simulation.

It is quite apparent that to put 4.5 kW hrs usable energy into the flywheel in 30 seconds will require a peak load capacity of 720 kW when the system losses are considered. The peak load capacity drops to 360 kW for the same 4.5 kW hrs charge delivered

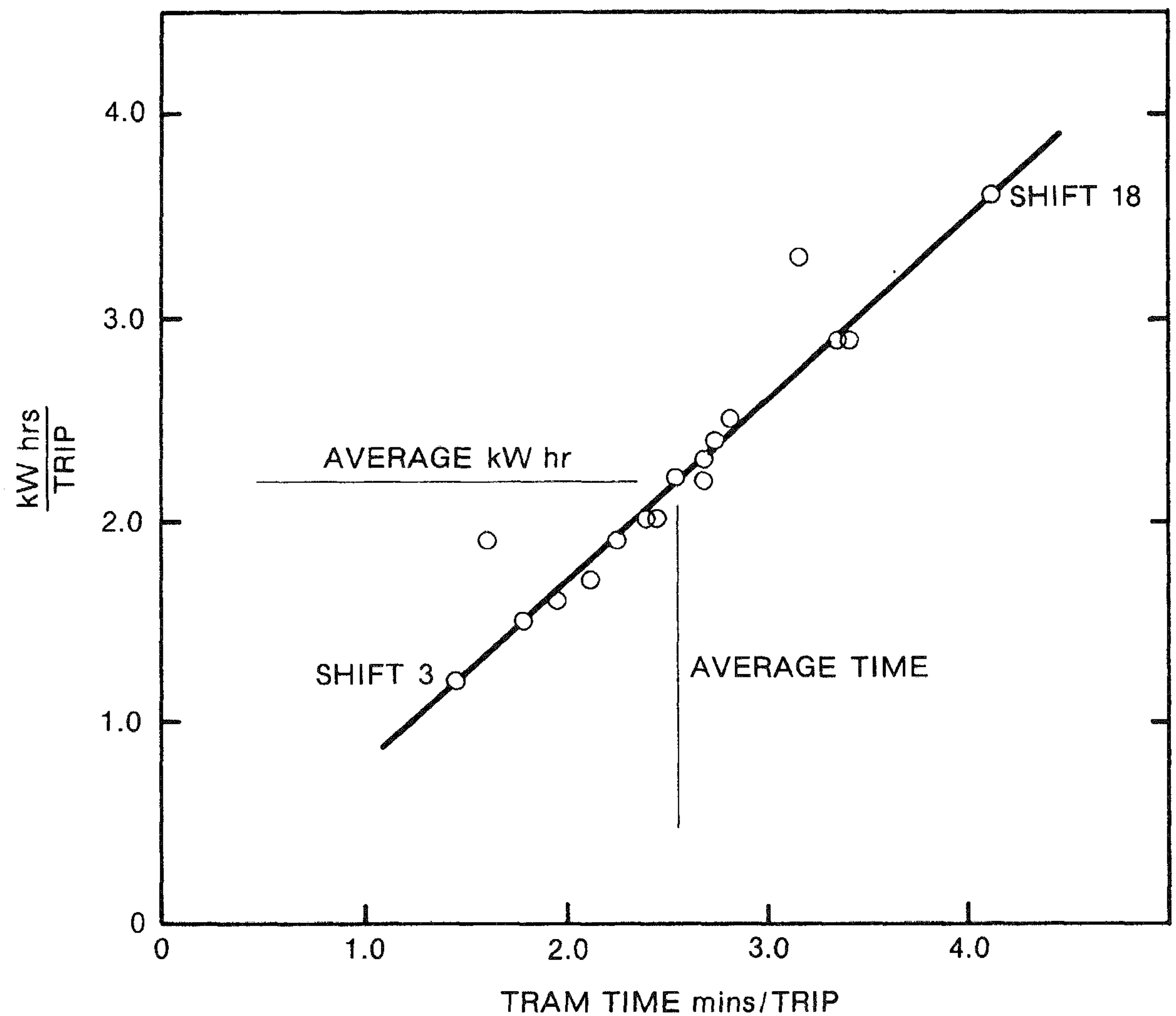


Figure 70. PSU/USBM Simulator - 6 ft Run kW hrs Versus Time Tramming Energy

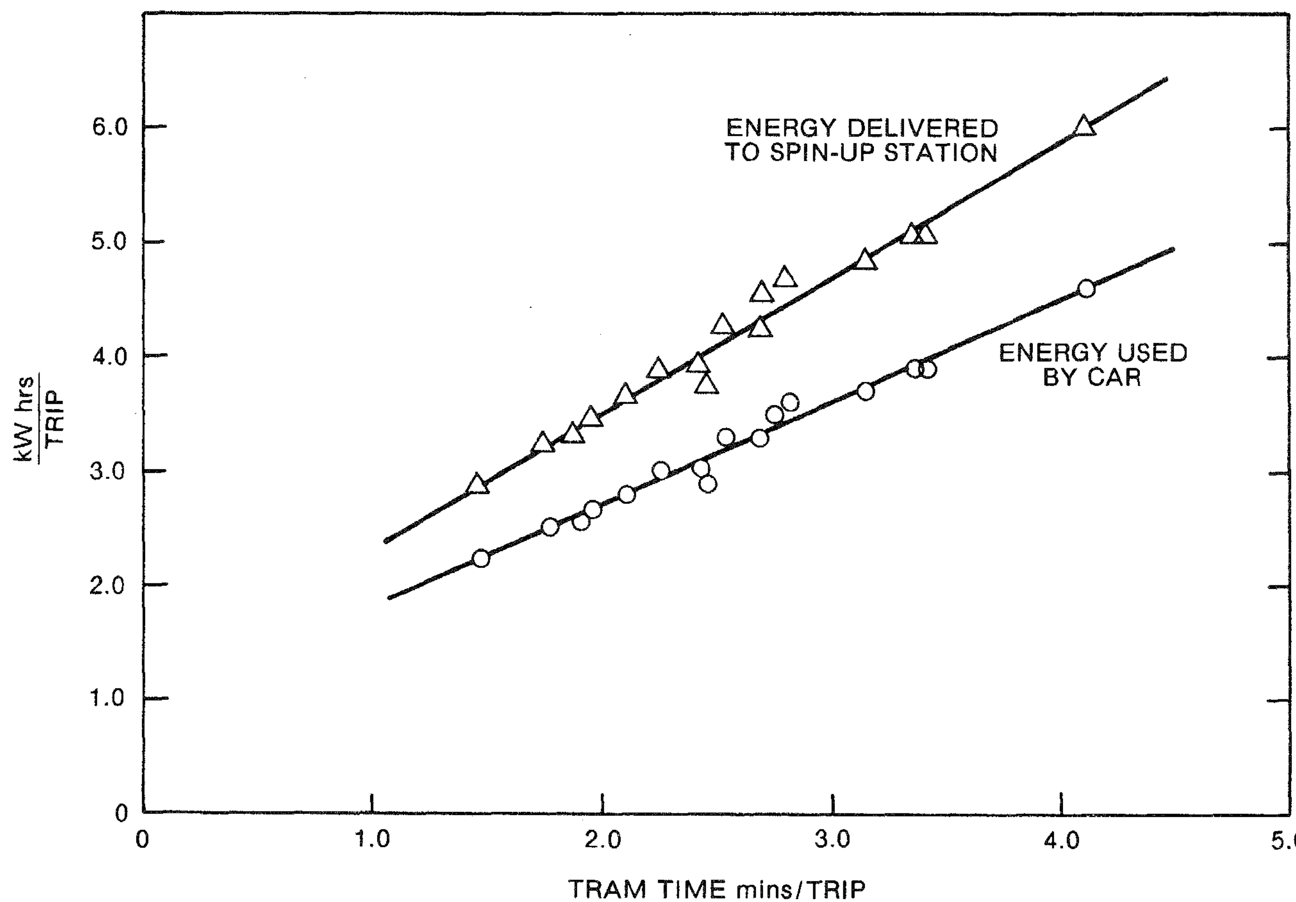


Figure 71. PSU/USBM Simulator - 6 ft Run Total Energy Used - kW hrs/Trip Versus Time/Trip

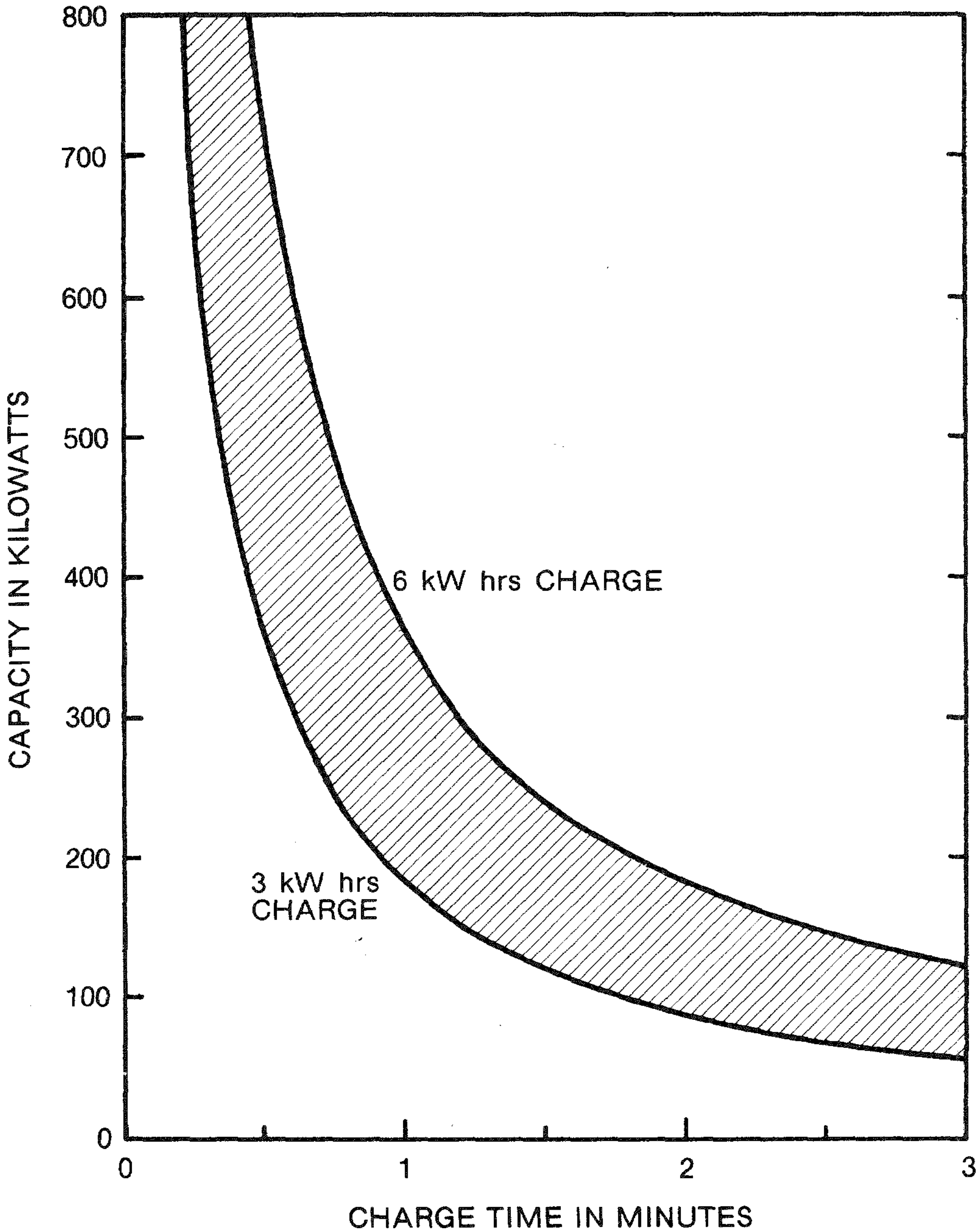


Figure 72. Charge Station Peak Capacity Required Versus Charge Time

in 60 seconds. In like manner 3 kW hrs of usable energy can be loaded into the flywheel in 30 seconds with a peak load capacity of 360 kW.

Underground mine power centers are available in a number of ratings from 150 to 1000 kVA with 750 kVA being a frequently used size. The power centers are rated to withstand 100% overload for one minute. However, other pieces of mining equipment such as the continuous miner, roof bolter, etc., share the output of the power center, and the voltage delivered to them will suffer some drop due to the regulation characteristics of the power center and its supply lines. A surging load such as that represented by the spin-up station of say 200 to 250 kW will not be too serious and can be tolerated, but a 720 kW surging load will probably cause objectionable voltage drop and require a dedicated power center with independent supply lines.

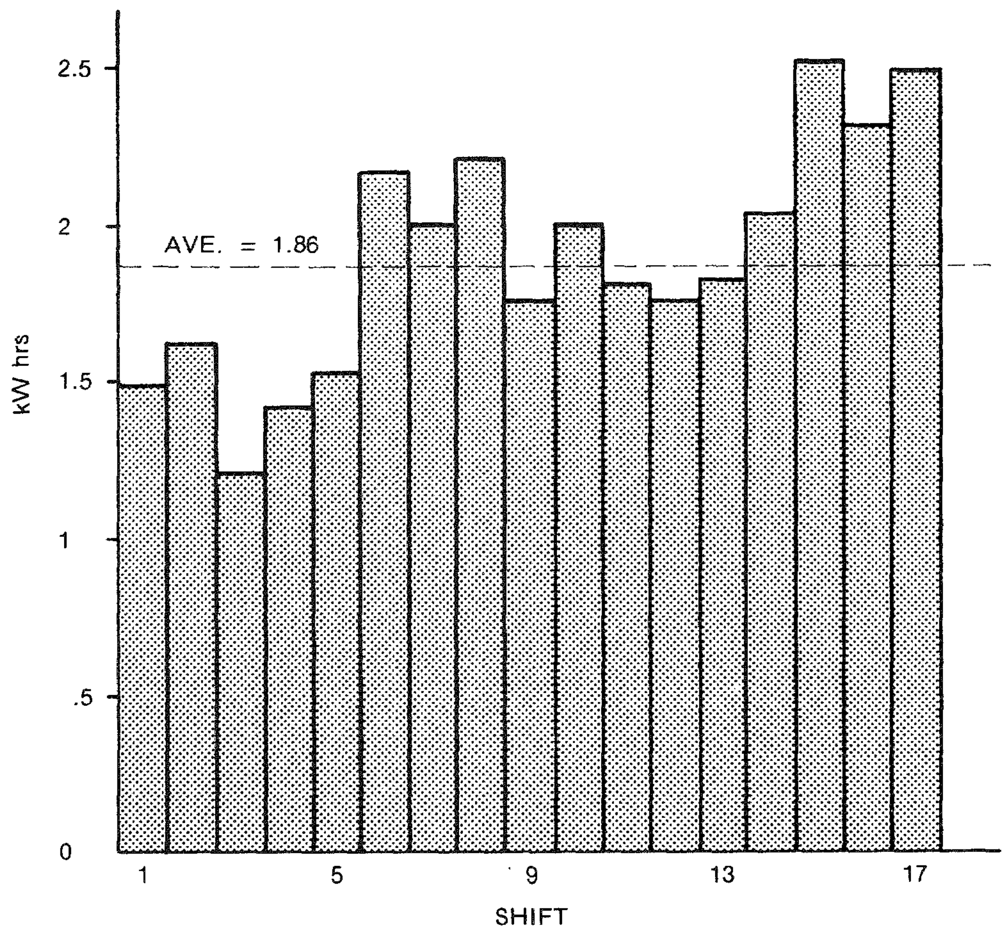
5.5.6 Charging Station Requirements

In Section 2 the maximum usable energy storage requirements for the flywheel were determined to be 4.5 kW hrs. This maximum was based on: rather poor bottom conditions, the longest tram path in the cutblock, energy required to unload and full stand-by power drain during all wait periods. Section 4.3 strongly indicates penalty in the size, weight and cost of the spin-up motor as a function of charging time. The penalty in these factors is approximately 2:1 to recharge 4.5 kW hrs in 30 seconds versus 80 seconds. Sections 5.4 and 5.5.5 indicate the same sort of penalty for wayside equipment, so there are strong incentives to minimize the size and capacity of spin-up station requirements. Balanced against this, the results of Section 5.5.1 show that every minute of miner wait-time is worth 1.4 tons of production, and miner wait-time is almost directly proportional to spin-up time.

The previously indicated study conducted by C.B. Manula for the U.S. Bureau of Mines indicates average bottom conditions of 165 pounds per ton rolling resistance and 2.07% grade. The same study further shows that about 80% of the bottom conditions actually encountered have less than 200 pounds per ton rolling resistance and 3% grade. The bar graph of Figure 73 shows the recharge energy required per trip by shift for 17 shifts required to work the base case cut-block with the following assumptions.

- Rolling resistance: 200 lbs/ton
- Grade: 3%
- Unload energy supplied by wayside
- Reduced wait time losses

On this basis it is evident that the average spin-up or recharge energy per trip will be 2 kW hrs, or less, for 80% of the shuttle car missions encountered in actual mine operations.



ASSUMED: 200 lbs/TON ROLLING RESISTANCE
3% GRADE
UNLOAD POWER FROM WAYSIDE
REDUCED STAND-BY LOAD

Figure 73. Energy Usage Per Trip by Shift

Figure 74 shows the time required to recharge 2 kW hrs of energy as a function of spin-up station, and motor/alternator, capacity. From this curve it is clear that the 203 kW (80 sec) motor determined in Section 4.3 will recharge 2 kW hrs in 36 seconds. Figure 74 also shows the cost of spin-up capacity based on the cost information in Appendix A. Capacity costs shown include: mine power center with rectifier, load commutating inverter and the motor/alternators for two cars. The estimated production costs per ton predicated on the time to recharge 2 kW hrs are also shown on Figure 74. Since the improvement in cost per ton diminishes below 30 seconds spin-up time while the cost of capacity continues to increase for shorter spin-up times, there is little incentive to select a spin-up capacity much greater than 200 kW.

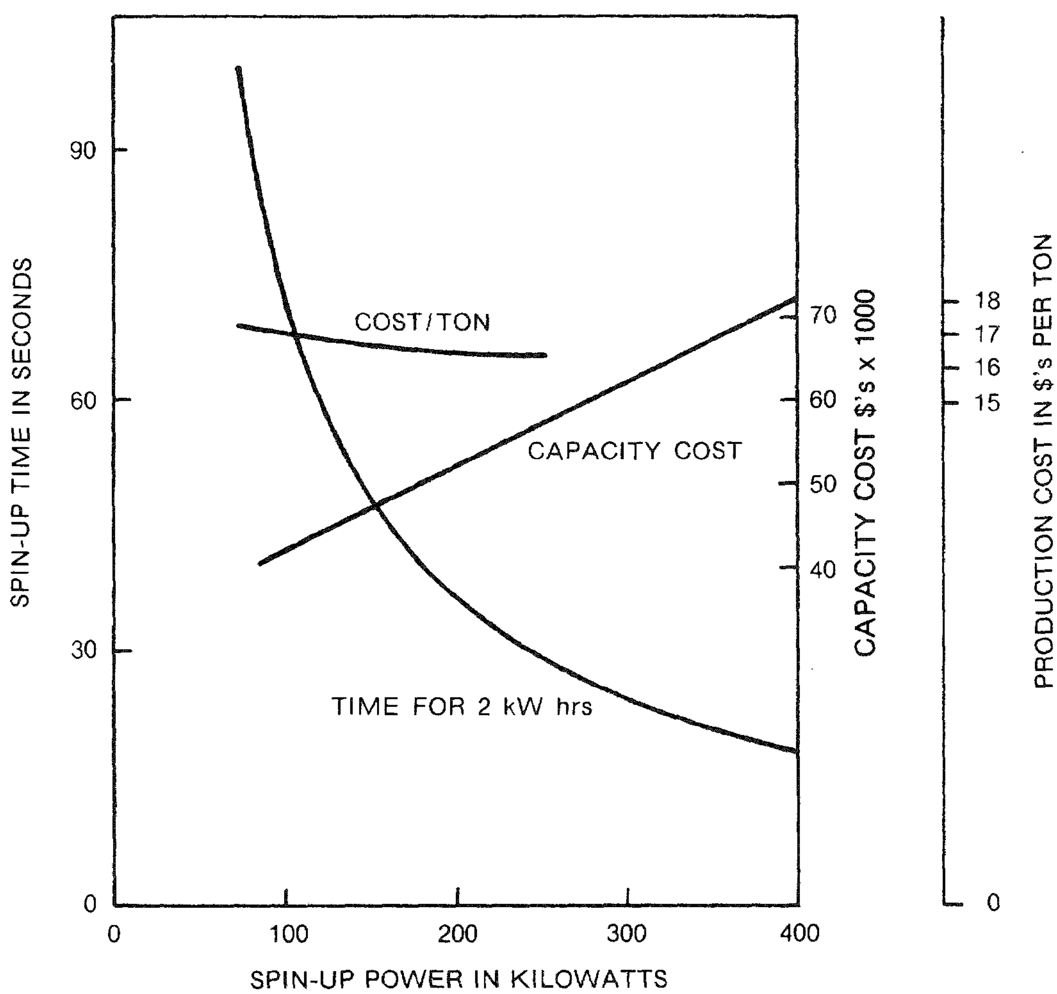


Figure 74. Cost of Spin-Up Capacity

The costs included in the cost per ton curve shown in Figure 74 include a large proportion of costs which are fixed and independent of the capacity of the spin-up equipment. Hence the only thing which can reduce cost per ton is increased productivity. Productivity in turn can only be improved by decreasing the miner wait on the shuttle car, of which the spin-up time is but a small fraction in the three-car systems assumed here. Even though theoretically a ram car can discharge in 15 seconds, practically a minimum unload time of 30 seconds has been assumed. As a consequence spin-up times of less than 30 seconds will show no improvement in productivity. These factors account for the relatively small change in production cost shown.

5.6 COST EFFECT ANALYSIS

Thus far the annual operating costs for various types of cars have been considered and the detailed composition of these costs as they are affected by parametric variables in flywheel powered shuttle car designs. The effects on productivity, in tons per shift, of the more significant design parameters such as flywheel spin-up time, turn around time, conventional cars versus tractor-trailer cars and car payload capacity have also been considered. This section assesses the cost effectiveness in terms of dollars per ton, for many of the practical design options which are available. This part of the report also compares the cost effectiveness of various types of cars such as: conventional tethered cars, battery-powered cars, diesel-powered cars, the developmental steam-powered cars and flywheel powered cars. An attempt has been made to account for all of the variables between the various car types. However, the secondary intangibles and interrelationships between tethered and untethered cars, conventional and tractor-trailer cars, payload capacities which vary by 2 to 1, and the operational impact of three cars in the face haulage system cannot be completely eliminated. While the calculations are carried out and shown to an accuracy of one tenth of a cent, the accuracy of the basic numbers, many of which are estimates, do not warrant considering the bottom line numbers to be any more accurate than ± 5 cents. A cost difference of approximately $0.30 per ton begins to look like a worthwhile improvement.

The first step in generating the cost effectiveness figures is to establish the basic cost numbers and the methodology used to determine the cost per ton numbers shown later in this section. These cost benefit figures are delineated below.

1. Labor costs are calculated on the basis of $90.00* per man day with an average of 21* men per section with two shuttle cars (22 men for three shuttle car cases). Total labor costs are then divided by productivity in tons per shift.

2. Supplies include all mine repair and provisioning items. Cost/ton does not vary as a function of tons produced since the faster mining operations progress the faster supplies are consumed. Supply costs are calculated at $3.962* per ton plus one of the following:

   a) Cable repair and replacement at $5800 - per year x 2 cars/327 tons x 440 shifts = $0.081 per ton.

   b) Battery replacement at $6000 per year x 2 cars/440 shifts per year = $27.27 per shift.

   c) Diesel engine overhaul at $12,800 per year x 2 cars/440 shifts = $58.18 per shift.

   d) Flywheel repair parts at $1400 - per year x 2 cars/327 tons x 440 shifts = $0.019 per ton.

*Numbers provided by C.B. Manula, State College, Pennsylvania See Appendix B.

e) Steam engine repair parts at $1500 per year x 2 cars/ 327 tons x 440 shifts = $0.021 per ton.

3. Electric power and fuel oil, like supplies, are also assumed to be constant values per ton. Energy costs are calculated at $0.475* per ton plus one of the following:

   a) Electricity for the tethered car at $1,170 per year x 2 cars/327 tons x 440 shifts = $0.016 per ton

   b) Electricity for battery car at $1780 per year x 2 cars/327 tons x 440 shifts - $0.025 per ton

   c) Electricity for flywheel car at $1540 per year x 2 cars/327 tons x 440 shifts = $0.021 per ton

   d) Fuel oil for diesel cars at $1400 per year x 2 cars/440 shifts = $6.36 per shift

   e) Fuel oil for steam cars at $1550 per year x 2 cars/440 shifts = $7.05 per shift

4. Health and welfare benefits are calculated on the basis of $12.77* per man per shift with 21* men per section for two shuttle car operations and 22 men for three shuttle cars. Total health and welfare benefits are then divided by tons per shift.

5. Compensation and black lung are calculated as in Number 4 on the basis of $18.00* per man per shift.

6. Administration costs are $245.25* per shift and are assumed to be constant and independent of number of tons produced and not affected by the change from 21 to 22 men per shift.

7. Insurance and taxes are also assumed to be fixed at $122.952* per shift.

8. Depreciation costs are calculated on the basis of a 600 foot deep, approximately one million ton per year mine with an average of 8.3 sections active over the year. Although this is a big mine, it is assumed that capital investment costs will scale linearly in proportion to size over the range of sizes of interest. An average depreciated life of 12 years is used for all items. For this mine the total capital investment, less shuttle cars, is detailed in Table XXXIII.* With 8.3 sections at 327 tons per shift per section, working

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*Numbers provided by C.B. Manula, State College Pennsylvania
See Appendix B

Table XXXIII

Capital Cost Summary ($'s x 1000)

SECTION EQUIPMENT		
Face (less shuttle cars)	$517	
Haulage	319	
Electrical	60	
General Haulage	162	
General Electric	21	
Miscellaneous	28	
+ 10% contingency	111	
Total per Section		$1,218
GENERAL INSIDE		
Mobile Equipment	$656	
Tools and Miscellaneous	215	
		871
SHAFT AND SLOPE		5,105
SURFACE EQUIPMENT		10,695
MOBILE SURFACE EQUIPMENT		415
INITIAL DEVELOPMENT COST		800
DEVELOPMENT LOSSES		5,000
+ 10% contingency		2,289
Total non-Section Related		$25,175

*Note data supplied by C. B. Manula, State College, Pennsylvania (See Appendix B).

For convenience the depreciation costs are broken down to cost per shift per section as follows:

- Total section related items, less shuttle cars = $1,218,000/12 years x 440 shifts/yr = $231 per shift.
- Total general mine depreciation costs = $25,175,000/ 12 years x 8.3 sects. x 440 shifts = $754 per shift.
- Total fixed depreciation costs $805 per section per shift.

To this $805 total must be added the cost of the shuttle cars plus a 10% contingency factor, divided by 12 years and 440 shifts. The grand total is then divided by the number of tons per shift produced. Shuttle car purchase costs are taken as:

a)	Tethered car	$80,000 each
b)	Battery car	$84,000 each with batteries and charger
c)	Diesel car	$111,100 each
d)	Flywheel car	$100,000 each + $20,000 wayside equipment
e)	Steam car	$130,000 each

9. Royalties are held constant at $0.50 per ton.

5.6.1 Flywheel-Powered Tractor-Trailer Car Haulage

With the above costs the cost effectiveness for the tractor-trailer car cases modeled two and three car operations with 30, 60 and 90 second flywheel spin-up times is considered in Table XXXIV.

Labor and its associated compensation expenses are the most dominant cost items. While the flywheel-powered cars cost more, they contribute an increment of only $.05 to $.10 to the depreciation cost/ton. The major factor in assessing cost/ton is productivity in tons per shift. As a consequence, it is very important to minimize shuttle car delays, due to any cause, since they reduce productivity.

Considering the estimating accuracy, two flywheel-powered tractor-trailer cars with 30 second spin-up times and three tractor-trailer flywheel powered systems show little change or a slight cost improvement over the base case tethered car. However, flywheel-powered cars offer the benefits of greatly reduced hazards and freedom of movement, both of which will result in indirect cost improvements, which cannot be quantified without actual experience.

5.6.2 Flywheel-Powered Conveyor Car Haulage

The next variable to examine is the cost effectiveness of conventional shuttle car systems employing two and three cars with 30, 60 and 90 second flywheel spin-up times. The results of these calculations are shown in Table XXXV. The improvement of conventional cars over tractor-trailer cars is approximately $.50 per ton for all cases. This improvement is due exclusively to the elimination of two-one quarter minute turn around delays and serves to highlight the importance of minimizing shuttle car delays.

5.6.3 Capacity Improvement

Based on the assumptions described in Section 5.5-4, Shuttle Car Payload Versus Productivity, it seems worthwhile to examine the cost effectiveness of capacity variations. For this purpose the Productivity Versus Payload Capacity curves of Figure 5.5-5 are used to examine tractor-trailer and conventional cars at 30 second spin-up times with payload capacities of 200, 250 and 300 cubic feet (10,000 - 12,500 & 15,000 pounds). Table 5.6-4 shows the results. From this it is quite clear that even a small change in car payload of 35 cubic feet (15%) produces noticeable results in cost. For example, the 220 cubic foot tractor-trailer car fitted with a flywheel will show a slightly negative cost improvement while a flywheel-powered tractor-trailer car at 269 cubic foot capacity will show some positive cost improvement.

5.6.4 Mine Economic Analysis

Table 5.6-5 is a complete economic analysis of flywheel-powered shuttle and tractor-trailer cars as an alternative investment opportunity, prepared by C.B. Manula, State College, Pennsylvania.

Table XXXIV

Cost Effectiveness for Simulated Cases

	TETHERED 2 CARS BASE CASE	2 CARS 30 sec SPIN-UP	2 CARS 60 sec SPIN-UP	2 CARS 90 sec SPIN-UP	3 CARS 30 sec SPIN-UP	3 CARS 60 sec SPIN-UP	3 CARS 90 Sec. SPIN-UP
Tons/Shift	327.0	325.0	319.0	309.0	343.0	342.0	341.0
Manning/Section	21.0	21.0	21.0	21.0	22.0	22.0	22.0
Cost/Man Day	90.0	90.0	90.0	90.0	90.0	90.0	90.0
Tons/Man Day	15.6	15.5	15.2	14.7	15.6	15.5	15.5
Costs In $'s/Ton							
Labor	5.780	5.815	5.925	6.117	5.773	5.789	5.806
Supplies	4.043	3.981	3.981	3.981	3.991	3.991	3.991
Power & Fuel	0.491	0.496	0.496	0.496	0.496	0.496	0.496
Health & Welfare	0.820	0.825	0.841	0.868	0.820	0.821	0.824
Compensation & Black Lung	1.156	1.163	1.185	1.123	1.155	1.158	1.161
Administration	0.750	0.755	0.769	0.794	0.715	0.717	0.719
Insurance & Taxes	0.376	0.378	0.385	0.398	0.358	0.360	0.361
Depreciation	2.564	2.618	2.667	2.754	2.541	2.549	2.556
Royalty	0.500	0.500	0.500	0.500	0.500	0.500	0.500
Total Cost/Ton	16.48	16.53	16.75	17.13	16.35	16.38	16.41
Cost Improvement/Ton	-	(.05)	(.27)	(.65)	.13	.10	.07

Notes:
- All cars have 236 cubic foot capacity.
- Base case includes 1 min at 40% frequency for cable delays.
- All tractor-trailer cars include 2 x 0.25 min at 100% frequency for turn around at each end of tram.

Table XXXV

Cost Effectiveness for Simulated Cases

	TETHERED	UNTETHERED					
	2 CARS BASE CASE	2 CARS 30 sec SPIN-UP	2 CARS 60 sec SPIN-UP	2 CARS 90 sec SPIN-UP	3 CARS 30 sec SPIN-UP	3 CARS 60 sec SPIN-UP	3 CARS 90 sec SPIN-UP
Tons/Shift	327.0	343.0	332.0	322.0	358.0	358.0	354.0
Manning/Section	21.0	21.0	21.0	21.0	22.0	22.0	22.0
Cost/Man Day	90.0	90.0	90.0	90.0	90.0	90.0	90.0
Tons/Man Day	15.6	16.3	15.8	15.3	16.3	16.3	16.1
Costs In $'s/Ton							
Labor	5.780	5.510	5.693	5.870	5.531	5.531	5.593
Supplies	4.043	3.891	3.891	3.891	3.991	3.991	3.991
Power & Fuel	0.491	0.496	0.496	0.496	0.496	0.496	0.496
Health & Welfare	0.820	0.782	0.808	0.833	0.785	0.785	0.794
Compensation & Black Lung	1.156	1.102	1.139	1.174	1.106	1.106	1.119
Administration	0.750	0.7515	0.739	0.762	0.685	0.685	0.693
Insurance & Taxes	0.376	0.358	0.370	0.382	0.343	0.343	0.347
Depreciation	2.564	2.481	2.563	2.642	2.436	2.435	2.462
Royalty	0.500	0.500	0.500	0.500	0.500	0.500	0.500
Total Cost/Ton	16.48	15.84	16.20	16.55	15.87	15.87	16.00
Cost Improvement/Ton	-	.64	.28	(.07)	.61	.61	.48

Notes:
- All cars have 236 cubic foot capacity
- Base case includes 1 min at 40% frequency for cable delays.

Table XXXVI

Cost Effectiveness for Simulated Cases

	TETHERED BASE CASE 236 ft^3	UNTETHERED TRACTOR-TRAILER CARS			UNTETHERED CONVENTIONAL CARS		
		200 FT^3	236 ft^3	270 ft^3	200 ft^3	236 ft^3	270 ft^3
Tons/Shift	327.0	312.0	325.0	335.0	330.0	343.0	351.0
Manning/Section	21.0	21.0	21.0	21.0	21.0	21.0	21.0
Cost/Man Day	90.0	90.0	90.0	90.0	90.0	90.0	90.0
Tons/Man Day	15.6	14.9	15.5	16.0	15.7	16.3	16.7
Cost In $'s/Ton							
Labor	5.780	6.058	5.815	5.642	5.727	5.510	5.385
Supplies	4.043	3.981	3.981	3.981	3.981	3.981	3.981
Power & Fuel	0.491	0.496	0.496	0.496	0.496	0.496	0.496
Health & Welfare	0.820	0.859	0.825	0.800	0.813	0.782	0.764
Compensation & Black Lung	1.156	1.211	1.163	1.128	1.145	1.102	1.077
Administration	0.750	0.786	0.755	0.732	0.744	0.7k5	0.699
Insurance & Taxes	0.376	0.394	0.378	0.367	0.373	0.358	0.350
Depreciation	2.564	2.727	2.618	2.540	2.578	2.481	2.424
Royalty	0.500	0.500	0.500	0.500	0.500	0.500	0.500
Total Cost/Ton	16.48	17.01	16.53	16.19	16.36	15.93	15.68
Cost Improvement/Ton							

Notes:
- All cases are 2 car systems.
- Base case includes 1 min at 40% frequency for cable delays.
- All untethered cases are with 30 second spin-up while unloading.
- Tractor-trailer cars include 2 x .025 min turn around delays.

Table XXXVII

C.B. Manula
Mine Economic Analysis
Flywheel Cars as an Alternative Investment

		SHUTTLE CARS				TRACTOR-TRAILER CARS			
		2 CARS 30 sec	2 CARS 60 sec	3 CARS 30 sec	3 CARS 60 sec	2 CARS 30 sec	2 CARS 60 sec	3 CARS 30 sec	3 CARS 60 sec
Added Investment Over 2 Th. Cars	($'s)	70,000	70,000	170,000	170,000	70,000	70,000	170,000	170,000
Life	(yrs)	10 yrs	10	10	10	10	10	10	10
Deprec/yr	($'s)	7,000	7,000	17,000	17,000	7,000	7,000	17,000	17,000
Added Profit/yr	($'s)	100,160	27,230	142,820	142,820	(20,140)	(61,370)	42,810	35,380
Tax @ 40%	($'s)	40,060	10,900	57.130	57.130	(8,160)	(24,550)	17,120	14,150
After-Tax Profit	($'s)	60,100	16,330	85,690	85,690	(12,250)	(36,820)	25,690	21,230
Plus Depreciation	($'s)	7,000	7,000	17,000	17,000	7,000	7,000	17,000	17,000
Cash Flow/yr	($'s)	67,100	23,330	102,690	102,690	(5,250)	(29,820)	42,690	38,230
Payback Period	(yrs)	1.04	3.00	1.66	1.66	-	-	3.98	4.45
Approx. Rate of Return	(%)	96	31	60	60	-	-	22	18

It shows an attractive cash flow and payback period for all of the practical cases under consideration except for two tractor-trailer cars replacing two conventional tethered shuttle cars.

Table XXXVIII, also prepared by C.B. Manula, is an economic analysis of flywheel-powered shuttle and tractor-trailer cars when applied on a replacement basis. It also shows a favorable cash flow for all cases except for two tractor-trailer cars. However, payback periods of 6½ to 7½ years are not generally considered to be especially attractive to industry. This furnishes a strong incentive to strive for design goals which would lead to the installation of flywheel energy storage in conventional shuttle cars.

Table XXXIX shows the build-up of cost per ton figures necessary for the preceding economic analyses. The conclusions from these finds have already been discussed in Section 5.3, Productivity or Cost Effectiveness Trade-Offs.

Capital cost details and average cost calculations are itemized in Appendix B.

5.6.5 Alternate Shuttle Car Comparisons

The following paragraphs compare conventional tethered shuttle cars with battery-powered tractor-trailer diesel powered tractor-trailer flywheel-powered cars and the Jeffrey developmental steam powered Ramcars. Although an attempt has been made to minimize differences between the cars, such as capacity, the comparisons should be considered as only rough approximations. Bottom conditions have been held constant at a rolling resistance of 300 pounds per ton over all tramming routes, and with a 5% uphill grade over all laden tramming routes. All cars shown, except the base case, are tractor-trailer cars with a 15 second turn around delay at each end of the tram. The base case is a conventional tethered car with a delay of one minute at a frequency of occurrence of 40% for elt repair delays. All cases are two-car systems. The results are shown in Table XL. Except for the diesel, all cars have essentially the same weight. No corrections have been factored into energy consumption due to the slight changes in weight. The results of such a correction would be of the order of one or two-tenths of a cent. Energy consumption of the diesel car is based on actual reported values. All cars are assumed to have a 30 second unload time although tractor-trailer cars can unload in 15 seconds in the secondary haulage can accept such a surge.

The battery cars show a cost benefit of $0.31 per ton. The Jeffrey 404L tractor trailer car is assumed to be equipped with heavy duty batteries which permit it to get through the worst case shift on one charge, at least while the batteries are in relatively good condition. Accordingly, no delays have been included for battery change-out during the shift. The cost improvement is due largely to the productivity improvement which, in turn, is brought about by the greater payload capacity, 269 versus 236 cubic feet.

Table XXXVIII

C.B. Manula
Mine Economic Analysis
Flywheel Cars as Replacements

		SHUTTLE CARS				TRACTOR-TRAILER CARS			
		2 CARS 30 sec	2 CARS 60 sec	3 CARS 30 sec	3 CARS 60 sec	2 CARS 30 sec	2 CARS 60 sec	3 CARS 30 sec	3 CARS 60 sec
Replacement Investment	($'s)	220,000	220,000	320,000	320,000	220,000	220,000	320,000	320,000
Live	(Yrs.)	10	10	10	10	10	10	10	10
Deprec/yr	($'s)	22,000	22,000	32,000	32,000	22,000	22,000	32,000	32,000
Added Profit/yr	($'s)	85,160	12,230	127,820	127,820	(35,410)	(76,370)	27,810	20,380
Tax @ 40%	($'s)	34,060	4,890	51,130	51,130	(14,160)	(30,550)	11,120	8,150
After Tax Profit	($'s)	51,100	7,340	76,690	76,690	(21,250)	(45,820)	16,690	12,230
Plus Depreciation	($'s)	22,000	22,000	32,000	32,000	22,000	22,000	32,000	32,000
Cash Flow/yr	($'s)	73,100	29,340	108,690	108,690	750	(23,820)	48,690	44,230
Payback Period	(Yrs.)	3.01	7.50	2.94	2.94	-	-	6.57	7.23
Approx Rate Of Return	(%)	31	6	32	32	-	-	9	7

Table XXXIX

C.B. Manula
Cost Per Ton for Simulated Cases

	TETHERED BASE	SHUTTLE CARS 2-30 sec	SHUTTLE CARS 2-60 sec	SHUTTLE CARS 3-30 sec	TRACTOR-TRAILER CARS* 3-60 sec	TRACTOR-TRAILER CARS* 2-30 sec	TRACTOR-TRAILER CARS* 2-60 sec	TRACTOR-TRAILER CARS* 3-30 sec
Tons/Shift	327.00	343.00	332.00	358.00	358.00	325.00	319.00	343.00
Manning/Unit	21/unit	21.00	21.00	22.00	22.00	21.00	21.00	22.00
Cost/Man-Day ($'s)	90.00	90.00	90.00	90.00	90.00	90.00	90.00	90.00
Tons/Man-Day	15.57	16.33	15.81	16.27	16.27	15.48	15.19	15.59
Labor ($'s/Ton)	5.780	5.511	5.693	5.531	5.531	5.815	5.925	5.773
Supplies* ($'s/Ton)	3.997	3.959	3.983	3.928	3.928	4.000	4.014	3.959
Power+ ($'s/Ton)	0.482	0.468	0.477	0.456	0.456	0.483	0.489	0.468
Health & Welf. (p.Ton) ($'s/Ton)	0.820	0.820	0.820	0.820	0.820	0.820	0.820	0.820
Comp. & Black Lung ($'s/Ton)	1.156	1.102	1.139	1.106	1.106	1.163	1.185	1.155
Admin. ($'s/Ton)	0.750	0.715	0.739	0.685	0.685	0.755	0.769	0.715
Insur. & Taxes ($'s/Ton)	0.376	0.359	0.370	0.344	0.344	0.378	0.386	0.359
Depreciation** ($'s/Ton)	3.062	2.965	3.063	2.905	2.905	3.130	3.188	3.032
Royalty/Depl. ($'s/Ton)	0.500	0.500	0.500	0.500	0.500	0.500	0.500	0.500
Total Cost/Ton ($'s)	16.92	16.40	16.78	16.28	16.28	17.04	17.28	16.78
Profit/Ton @20 Price ($'s)	3.08	3.60	3.22	3.72	3.72	2.96	2.72	3.22
Profit/Yr. @ 440 Shf/yr (&'s)	443,150	543,310	470,380	585,970	585,970	422,740	381,780	485,960
Increase Over Base/yr ($'s)		100,160	27,230	142,820	142,820	(20,410)	(61,370)	42,810

* Assumed no change in cost/ton for untethered shuttle cars or tractor-trailer cars.
** Includes added depreciation over basic shuttle car costs for untethered cars.
+ Power rate not adjusted for change in efficiencies of cars -minor change
x Analyses for tractor-trailer cars are conservative since faster dump time and normally expected larger payload were ignored.

Table XL

Cost Effectiveness for Simulated Cases

	TETHERED BASE CASE 1	BATTERY 2	DIESEL 3	STEAM 4	FLYWHEEL 5
Tons/Shift	327.0	335.0	360.0	320.0	335.0
Manning/Section	21.0	21.0	21.0	21.0	21.0
Cost/Man Day	90.0	90.0	90.0	90.0	90.0
Tons/Man Day	15.6	16.0	17.1	15.2	16.0
Costs In $'s/Ton					
Labor	5.780	5.642	5.250	5.906	5.642
Supplies	4.043	4.043	4.124	3.983	3.981
Power & Fuel	0.491	0.500	0.493	0.497	0.496
Health & Welfare	0.820	0.800	0.744	0.838	0.800
Compensation & Black Lung	1.156	1.128	1.050	1.181	1.128
Administration	0.750	0.732	0.681	0.766	0.732
Insurance & Taxes	0.376	0.367	0.342	0.384	0.367
Depreciation	2.564	2.455	2.365	2.600	2.540
Royalty	0.500	0.500	0.500	0.500	0.500
Total Cost/Ton	16.48	16.17	15.55	16.66	16.19
Cost Improvement/Ton	-	.31	.93	(.18)	.29

Notes:
1. Joy 18SC13DC, 236 ft^3 conventional
2. Jeffrey 404L, 269 ft^3 Ramcar
3. Jeffrey 410H, 445 ft^3 Ramcar
4. Jeffrey Developmenta, 220 ft^3 Ramcar
5. Tractor-trailer car, 269 ft^3 Ramcar

The large cost advantage of the diesel cars is due exclusively to increased payload. On the other hand, no allowance has been made for the increased costs, or delays, caused by the requirement for increased ventilation associated with diesel operation

The slightly negative cost effectiveness of the developmental steam car is due to the increase in car costs coupled with the effects of a slightly decreased payload. If operating conditions permit discharging the car in 15 seconds, the cost would probably turn out to about equal the base case. Again, no allowances for increased ventilation requirements, if any, have been factored into the calculations.

Some cost improvement is seen for the flywheel-powered shuttle car. In this comparison a slightly higher (269 ft^3) capacity is assumed along with an average spin-up time of 30 seconds while unloading.

Section 6

ALTERNATE DRIVE SYSTEMS

Several alternate drive systems were considered which could fulfill the requirement of converting flywheel energy to tractive effort. Practicability, reliability, and proven applications were the principal guides, and factors such as efficiency, weight, heat load and control complexity were technical aspects which were also considered. Four systems were investigated:

- Hydromechanical
- Electromechanical
- Hydraulic
- Torque Converter

Each of the systems studied represents an approach which can be implemented through a development effort. In most cases the major components are standard vehicle drive units which can provide a straightforward transmission package. In each application the drive systems could use an existing differential/axle/wheel system. The conceptual design considerations in these mechanical systems are discussed in the descriptions of the systems in Section 6.2 and in the system comparisons of Section 6.3.

6.1 SUMMARY AND CONCLUSIONS

There are three serious considerations which tend to weigh against mechanical drive systems for use with a flywheel energy storage package in a shuttle car. First, and probably most important, is the point that the flywheel should be mounted with its rotating axis vertical to minimize gyroscopic effects. This means that the mechanical energy must be coupled out of the top or the bottom of the flywheel package through a set of right angle gears to the drive train of the vehicle. Since the basic flywheel-motor/alternator package is of the order of 36" long, maintaining bottom clearance, headroom and operator visibility will be difficult. In addition, coupling to drive train components, such as a gear box, will be awkward at best. Second, and of almost equal importance, is the fact that it is desirable to operate the flywheel at reduced pressure to minimize windage losses. As a result, any mechanical energy output shaft must pass through a vacuum seal. Since all rotating vacuum seals have a marked proclivity to leak, the use of mechanical drive systems mandate the added complexity of an on-board vacuum pump. A third, though less serious consideration, is the fact that the flywheel speed continues to decrease throughout the running time of the mission. To compensate for this, the mechanical drive system must employ components which will permit operation over a 2 to 1 variation of input speed.

The preliminary conclusions drawn from a generalized study of mechanical transmissions for use with a flywheel energy storage package are:

- A mechanical transmission system could be designed to transmit flywheel energy to the traction components in a mine shuttle car although component location would be difficult.

- A detailed design study would be required to define with accuracy which of the systems is optimum. During further study, factors such as hermetic sealing, rotating mechanical seals, control complexity, overall efficiency, and component location would require careful analysis based upon the specialized requirements of a mine shuttle car.

- Since an electrical charging system has been chosen over a mechanical charging system (Section 8), there does not appear to be any way to eliminate the on-board a-c machine and thus achieve a major cost and weight benefit. With electrical charging, all the alternate drive systems except electromechanical, the a-c machine is used only for charging, yet must be carried on-board.

- A primary development effort for alternate mechanical systems would be directed toward the vehicle propulsion system, rather than the flywheel system. Thus, the objective of proving the feasibility of a flywheel propulsion system for shuttle cars can best be achieved by concentrating development effort on the flywheel package and using it with an electric propulsion motor and a commercially available mechanical drive.

6.2 DESCRIPTION OF SYSTEMS

6.2.1 Hydromechanical

Figure 75 illustrates a system which transmits the flywheel energy, through a gear reduction, to an infinitely variable transmission (IVT). The IVT is then connected directly to the drive shaft of the existing shuttle car vehicle. The IVT can be either hydrostatic, hydromechanical, or a unit which is a combination of hydrostatic and hydromechanical. The hydrostatic unit is a direct acting, piston-type device using hydraulic fluid as the energy transmitting medium. The hydromechanical unit uses hydraulic fluid acting through a series of gears. Both of these systems are available commercially in power ranges which could be adapted for use in a mine shuttle car. The hydrostatic unit provides greater efficiency in the low speed ranges while the hydromechanical unit provides greater efficiency in the higher speed ranges. At present a unit which combines both hydrostatic and hydromechanical in one smooth acting unit is available only in a size too large for shuttle car application.

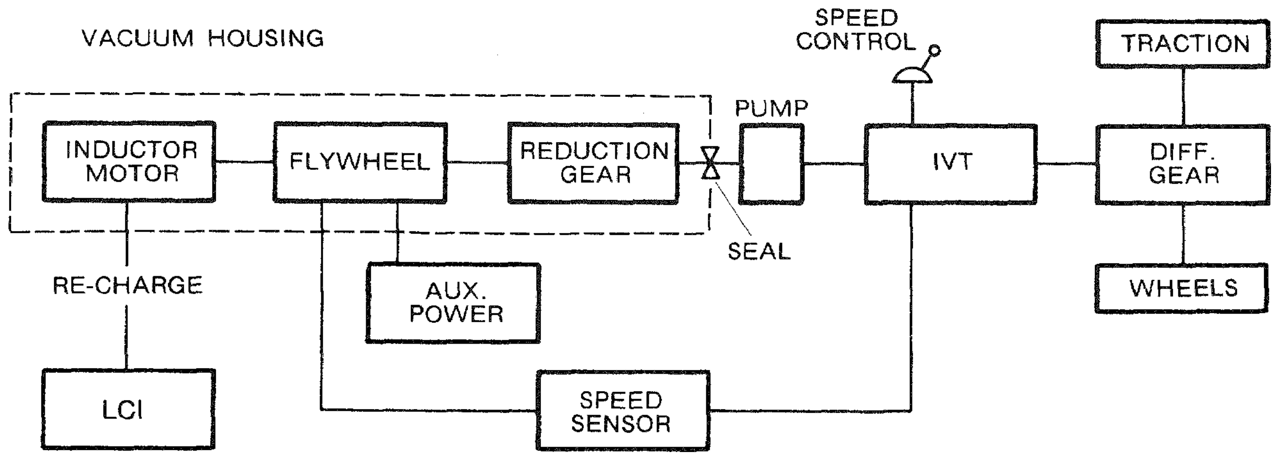

Figure 75. Alternate System - Hydromechanical

The controls for this transmission system become those of the IVT. In comparison to the electrical systems described in Section 4.7, the need for on-board power conditioning and related control is eliminated. The flywheel energy storage unit uses rotating seals and requires a reduction gear. Losses in the IVT and reduction gear unit are partially compensated by the elimination of the electrical losses in an inductor machine, the rectifier, and the traction motor, which are not used in this system.

The simplicity of an IVT system, except for the mechanical complexity of coupling to the flywheel package, and the availability of proven commercial hardware recommend its consideration for further evaluation.

The energy storage unit for this hydromechanical concept is similar to the unit in an all-electric system, Section 4.7, except for the gearing and seals required. Charging of the storage unit is identical to the all-electric unit.

6.2.2 Electromechanical

The electromechanical system illustrated in Figure 76 transmits drive power through a combination of a mechanical and an electrical drive system. A differential gear at the power output end of the flywheel unit links the two systems. In operation the electric drive system serves to augment the mechanical drive system during low speed operation. This tends to increase the efficiency of the system since the efficiency of a direct mechanical coupling tends to be superior to that of a mechanical-electrical-mechanical conversion. The primary drive system links the flywheel through the differential to the dc motor and finally to the traction wheels. During accelerations and cruise conditions the torque transmitted through the electric system combines with the mechanical system. Thus, a portion of the power is always transmitted mechanically.

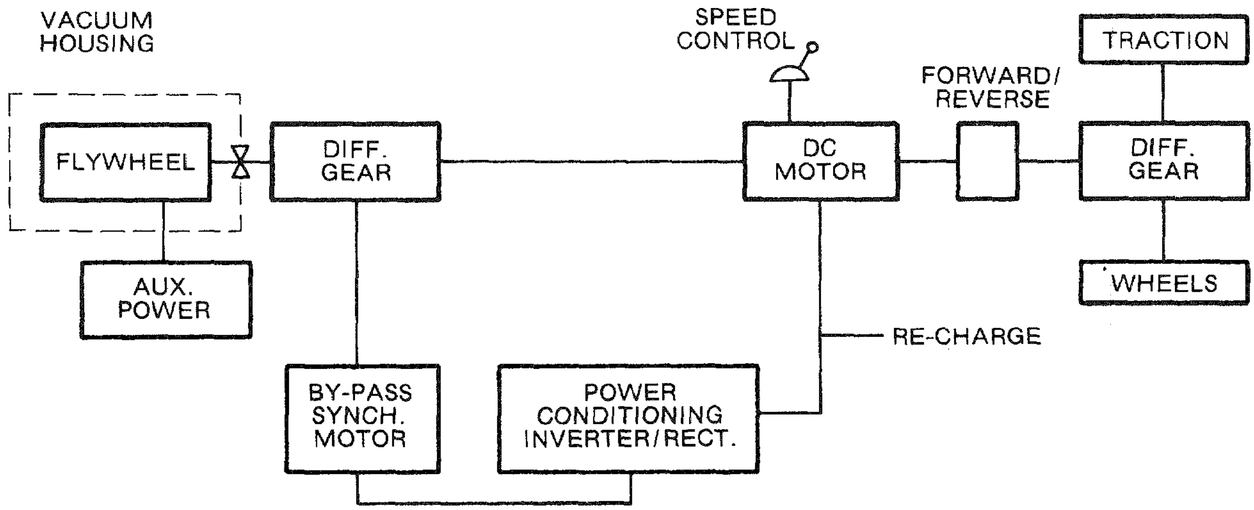

Figure 76. Alternate System - Electromechanical

The synchronous motor and power conditioning units, and a two speed differential complicate the system and add a burden of weight to the drive package size. The problems of hermetic sealing and rotary seals also exist. Recharging of the system does not require a separate inductor motor since the by-pass synchronous motor provides this function.

6.2.3 Hydraulic

Figure 77 illustrates a hydraulic power system. A reduction gear serves to reduce the flywheel output speed to an acceptable speed for driving a hydraulic pump. The pump output can then be utilized in one of two ways. First, the hydraulic pump can drive hydraulic motors which can be contained in the hubs of the traction wheels. This eliminates the need for a differential gear linking the traction wheels. In the second system, not illustrated, the hydraulic pump can power a hydraulic motor which is connected directly to the traction differential of the vehicle. This would be the system most easily adapted to an existing car.

The hydraulic system can use available components and control systems. Hermetic sealing and rotating mechanical seals are disadvantages unless the hydraulic pump is inside the flywheel containment, which makes the flywheel package larger and more difficult to fit in the shuttle car. Reduction gearing is also required. The control system is similar to that used with the hydraulic system.

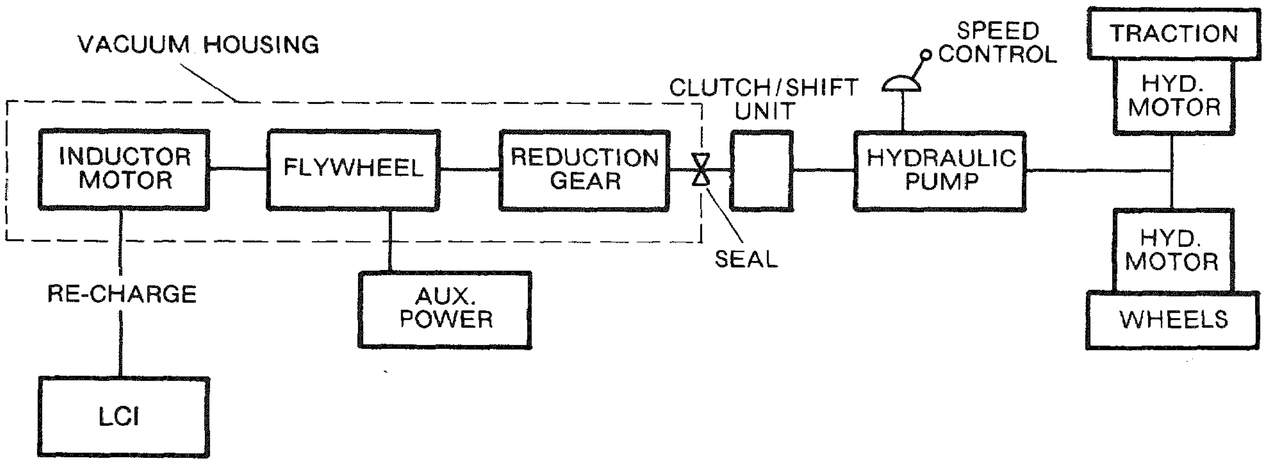

Figure 77. Alternate System - Hydraulic

6.2.4 Torque Converter

The torque converter drive train illustrated in Figure 78 is similar to the infinitely variable transmission system. The torque converter is a hydrokinetic device, which transmits power through the application of kinetic energy to the hydraulic fluid medium. It provides a smooth transfer of power and can be used as a component in a mechanical drive train. The major disadvantage is that the output torque is controlled by the ratio of input to output speed. Thus, an increase of input speed is required to increase the output torque. Since the flywheel speed is decreasing, a gear shift unit must be used or acceleration will be sacrificed to obtain sufficient torque to overcome obstacles.

Simplicity and commercial availability are plus factors, but the coupling of the flywheel to the drive is difficult. A two or three speed gear box may be required to sustain input speed to the torque converter for optimization. Again, sealing of the flywheel energy package and reduction gearing are design problems of the system.

6.3 COMPARISON OF SYSTEMS

In order to assist comparison of the systems, a preliminary sizing calculation is developed in Appendix C.

Based on a rating analysis shown in Tables XLI, XLII and XLIII, the hydromechanical system appears to be a suitable mechanical candidate for further study. Factors such as availability of components, degree of development effort required, volume, weight, and efficiency, all favor a hydromechanical system, but mechanical coupling to the flywheel is an obstacle. The torque converter system, which in essence is a hydromechanical device, would be the second choice. The torque converter system, however, because of its torque/rpm characteristic may require a greater number of transmission components and increased control complexity.

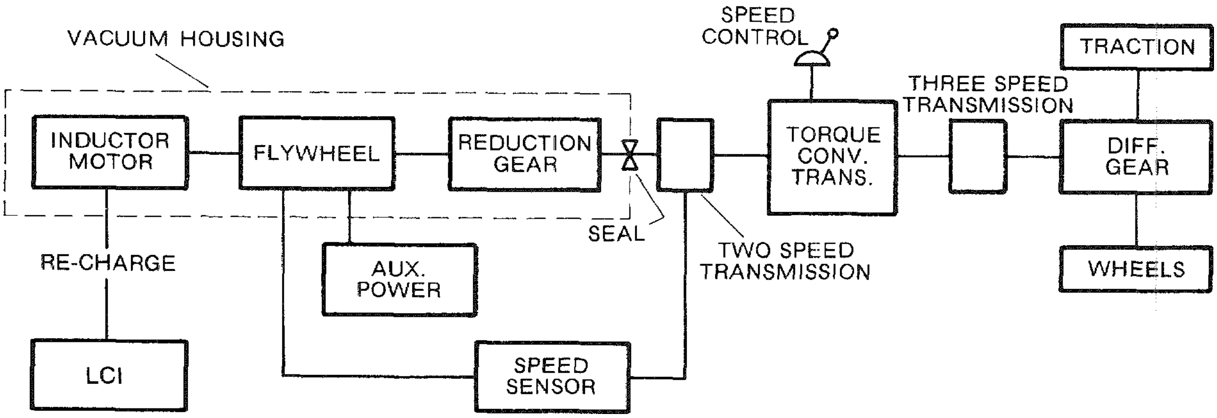

Figure 78. Alternate System - Torque Converter

Similarly the hydraulic drive system promises a straightforward application of commercial units and offers, additionally, a considerable overall drive size reduction. This system could be adapted to a four wheel shuttle car with center conveyor. The electromechanical system is a specialized design and would require a considerable design and development effort. This system makes sealing the flywheel unit especially difficult. Either a high speed rotating seal is required on the flywheel output shaft, or else two seals are required on the output shafts of the differential power split-off gearbox.

Table XLI lists General Rating Factors which compare the several systems to the electrical system described in Section 4.7. None of the ratings or values assigned to any of the systems should be construed as final ratings. They are used to obtain a gross comparison. While Table XLI generates information on a comparative basis, Table XLII refines the evaluation somewhat further by assigning a relative value to the rating factors. Weighting values are also assigned in Table XLIII to more clearly distinguish the comparative ratings of the systems.

Table XLIII lists estimated component values of weight, volume, and efficiency. These values are general in nature and serve only for relative comparison.

The tables show that weight and volume favor the hydromechanical and hydraulic systems, and the efficiency of the hydraulic system appears to be better than the selected electrical system although the overall ratings favor the electrical system by a relatively wide margin. In the analysis in Table XLIII, the relative efficiencies exclude common components such as flywheel losses since the objective is to obtain a relative comparison.

In the mission analysis, Section 2, efficiency of the flywheel systems included only the flywheel losses, inductor motor losses and power conditioning losses, since the objective in Section 2 was to determine flywheel rating.

Table XLI

General Rating Factors

SYSTEM	MECHANICAL COMPLEXITY	MAIN-TENANCE SERVICE	DEGREE OF DEVELOPMENT EFFORT	GENERAL OVERALL EFFICIENCY (SERIES TOTAL)	OVERALL DRIVE SIZE ft^3	OVERALL WEIGHT lbs (W/O COOLING APPARATUS)	AVAILABILITY OF COMPONENTS
Inductor (Rectifier)	Simplest Adaptation	Low	Moderate	.65	16.6	2180	Special Design
Inductor (LCI)	Larger Components On-Board	Low	Moderate	.74	19.0	2600	Special Design
Hydro-mechanical	Difficult Flywheel Mechanical Connection	High	Moderate	.71	5.8	1162	Available
Electro-mechanical	Difficult Flywheel Mechanical Connection	Low	High	.72	18.4	2540	Special Design
Hydraulic	Pump Inside Flywheel Package	High	Moderate	.64	7.2	1800	Available
Torque Converter	Difficult Flywheel Mechanical Connection	High	Low	.66	11.5	1555	Available

Table XLII

System Evaluation

Mechanical Complexity:	Difficult = (-), Moderate (0), Little change (+)
Maintenance:	Low = (+), Medium = (0), High = (-)
Weight:	Low = (+), Moderate = (0), High = (-)
Development Effort:	Low = (+), Moderate = (0), High = (-)
Efficiency:	High = (+), Moderate = (0), Low = (-)
Control Complex.:	Low = (+), Moderate = (0), High = (-)
Availability:	Available = (+), Special Design = (-)
Overall Size:	0 to 10 = (+), 10 to 20 = (0), Over 20 = (-)

Evaluation Factors

SYSTEM	MECH. COMPLEX.	MAINT. REGT.	DEV. EFFORT	EFF.	SIZE	WEIGHT	AVAIL. OF COMPONENTS	TOTAL
Inverter Rect.	+	+	0	0	0	0	-	+1
Inverter LCI	0	+	0	+	0	-	-	0
Hydro-Mech.	-	-	0	+	+	+	+	+2
Electro-Mech.	-	+	-	+	0	-	-	-2
Hydraulic	0	-	0	0	+	0	+	+1
Torque Conv.	-	-	+	0	0	+	+	+1
Weighting for each evaluation factor	2	2	2	2	1	1	1	
Inverter Rect.	+2	+2	0	0	0	0	-1	+3
Inverter LCI	0	+2	0	+2	0	-1	-1	+2
Hydro-Mech.	-2	-2	0	+2	+1	+1	+1	+1
Electro-Mech.	-2	+2	-2	+2	0	-1	-1	-2
Hydraulic	0	-2	0	0	+1	0	+1	0
Torque Conv.	-2	-2	+2	0	0	+1	+1	0

Table XLIII

Transmission Weights, Sizes, Efficiencies

The following are order of magnitude values which can be employed for preliminary definition and evaluation of the drive systems under discussion.

SYSTEMS*	ON-BOARD WEIGHT (lbs)	VOLUME (ft^3)	GENERAL TOTAL EFFICIENCY	
Electric (Fig. 32)				
Inductor Motor	700	3.3	91	
DC Motor	700	2.0	91	
Rectifier & El. Control	400	6.3	97	
Torque Converter	70	3.7	85	
3 Speed Transmission	310	1.3	95	
	2180	16.6	Series Total = .65	
Electric (Fig. 33)				
Inductor Motor	700	3.3	91	
DC Motor	700	2.0	91	
LCI Electrical Control	1000	13.0	91	
Reduction Gear	200	0.7	98	
	2600	19.0	Series Total = .74	
Hydromechanical (Fig. 75)				
Reduction Gear	200	0.7	98	
Pump	159	1.3	80	
IVT	103	0.5		
Synchronous Machine	700	3.3	91	
	1162	5.8	Series Total = .71	
Electromechanical (Fig. 76)			Mech. Path	Elec. Path
Differential	300	1.1	.97	.97
Synch. Motor	700	3.3		.91
Inv/Rect.	400	6.3		.91
Coupling	90	1.3	.99	
Coupling	90	1.3	.99	
Clutch	90	1.0	.80	
Tachometer	70	1.0	.99	
DC Motor	700	2.0		.95
F/R Shift	100	1.1	.95	.95
	2540	18.4	Series Total = .72	
Hydraulic (Fig. 77)				
Reduction Gear	200	0.7	98	
Pump	300	1.4	85	
2 Motors	600	1.8	85	
Inductor Motor	700	3.3	91	
	1800	7.2	Series Total = .64	
Torque Converter (Fig. 78)				
Reduction Gear	200	0.7	98	
2 Speed Trans.	175	1.4	96	
T.C.	70	3.7	85	
3 Speed Trans.	310	1.3	95	
F/R Shift	100	1.1	95	
Inductor Motor	700	3.3	91	
	1555	11.5	Series Total = .66	

*Common equipment not included: flywheel & containment, differential gearing & shafting, auxiliary power

Section 7

CABLE REEL SHUTTLE CAR SAFETY IMPACT

7.1 SUMMARY AND CONCLUSIONS

An assessment of the safety data available on presently used mine shuttle cars was necessary to evaluate the safety considerations of utilizing a flywheel system in a mine. This section discusses the available data covering shuttle car accidents, safety considerations of shuttle cars with trailing cables, and data of electrical accidents including those associated with trailing cables.

The study indicates that of the 750 to 1000, or more, mining electrical accidents and injuries per year some 40% occur in the face area of underground mines (Table XLVIII). Additionally, nearly 31% of the electrical accidents and injuries are caused by cables. While shuttle car cables represent some 20% to 40% of the electrical cables in the face area, the very nature of their use -- constantly flexing and scrubbing against ribs and floors -- makes them much more prone to failure. It is conservatively estimated that 50% of the 90 to 120 electrical accidents and injuries caused by electrical cables in the face area of underground mines may be attributed to shuttle car tether cables. Therefore, it may be concluded that the introduction of the internally powered flywheel shuttle car holds the promise of eliminating at least 45 to 60 electrical accidents and injuries per year in underground mines. The fatality data in Table L reveals that 38% (3 out of 8) of the cable related fatalities were due to shuttle car operations. At least two of these were the direct result of splicing activities.

The matrix of Table XIIV represents an attempt to summarize the major advantages of the various shuttle cars under consideration in this study: tethered, battery, diesel (open & closed cycle), flywheel, and steam powered. No attempt has been made to put weighting factors on the various hazards, but it seems pretty clear that there are no obvious winners or losers among the various car power systems.

The accident potential of cable reel shuttle cars has been recognized, and efforts to eliminate or reduce this potential have been underway since 1969. The development of automatic guidance systems, and the introduction of canopies are two such efforts. The trailing cable has also been recognized as a major safety and maintenance hazard, and alternate power systems that will not depend on the trailing cable, such as this work on a flywheel system, are being investigated. While diesel powered shuttle cars have been recommended by some studies, diesel engines call for increased efforts on engine maintenance and on the quality of the mine atmosphere. All of this suggests the need to develop other

Table XLIV

Shuttle Car Hazard Factors

HAZARD/TYPE OF CAR		TETHERED	BATTERY	DIESEL OPEN CYCLE	DIESEL CLOSED CYCLE	FLYWHEEL	STEAM
Hazards Associated with Normal Operations:							
Cable, electrical & mechanical	(1)	X					
Presence of hazardous fluids	(2)		X	X	X		X
Products of combustion	(3)			X			X
High temperatures (burns)	(4)			X	X		X
Electrical shock	(5)	X	X			X	
Presence of high pressures	(6)				X		X
Flywheel burst	(7)					X	
Fire-electrical origin	(8)	X	X			X	
Fire-high temp. origin	(9)			X	X		X
Endurance-limited energy	(10)		X			X	
Mechanical damage	(11)	X	X	X	X	X	X
Hydraulic fluid leakage	(12)	X	X	X	X	X	X
Tire blowouts	(13)	X	X	X	X	X	X
Hazards Resulting from Mine Incidents Damaging Car:							
Electrical shock	(14)	X	X				
Fire	(15)			X	X		X
Explosion	(16)				X	X	X
Inability To Use Car for Emergency Escape	(17)	X	X			X	

Footnotes to Table 10.1-1

1. Hazards unique to the cable, e.g. electrical shock from faulty insulation or while splicing, tripping over the cable or being knocked down by it while car is in motion.
2. Hazards created by accidental spills while refilling or spills from ruptured tanks or battery cases.
3. Presence of toxic and noxious fumes which impose increased ventilation requirements.
4. Presence of high temperatures in engines, boilers, and exhaust systems which could result in serious burns if contacted accidentally.
5. Presence of high voltages in the car which could result in shock from exposed wiring etc., resulting from damage and improper maintenance.
6. Presence of high pressures, large quantities of stored energy in oxygen tanks and steam boilers which could result in explosions from accidents and/or improper maintenance.
7. Flywheel burst resulting from fatigue or an accident causing damage to flywheel containment.
8. Fire resulting from electrical sparks caused by damaged or worn and improperly maintained electrical equipment.
9. Fire resulting from flammables contacting high-temperature engines or boilers.
10. Accidents resulting from attempts to extricate shuttle cars which have become stuck or stalled as a result of limited on-board energy storage.
11. Accidents caused by moving vehicles or secondary effects of damage to vehicles' mechanical protective devices, e.g. motor housings, drive train components, brakes, etc.
12. Leakage of flammable and slippery hydraulic fluid caused by damaged or worn and improperly maintained hydraulic systems.
13. Accidents caused directly by tire blowouts or loss of vehicle control resulting from worn or damaged tires.
14. Hazard of electrical shock resulting from damage to vehicle caused by a mine incident (cave in).
15. Fire hazard resulting from damage to vehicle caused by a mine incident.
16. Potential explosion resulting from damage to vehicle caused by a mine incident.
17. Inability to use the shuttle car as an emergency escape vehicle due to limited cable length or limited on-board energy storage.

types of face haulage equipment that does not degrade the mine environment while at the same time eliminates the requirement for trailing cables.

From these viewpoints a flywheel powered shuttle car appears to be a very attractive possibility. The safety impact of a flywheel system include the possibilities of a flywheel burst and problems associated with an electrical or mechanical connection required to spin-up or charge the flywheel. The application of conservative design practices, particularly on stress and fatigue life factors, will essentially eliminate the possibility of a burst. Also a suitable containment will be incorporated in the flywheel housing to prevent flywheel burst fragments from escaping and causing any external damage or injury. The charging connector must be designed so that safety of operation and maintenance will be primary design considerations.

In underground coal mining, cable reel shuttle haulage in the face area is the most common method of transportation. In 1974, there were over 11,000 rubber-tired vehicles in deep coal mines. Cable reel shuttle cars accounted for 6,050 or 55% of this equipment. Nearly 94% of the coal that was mechanically loaded was moved away from the face areas by shuttle cars (Minerals Yearbook, Reference 5). Since fatalities and/or non-fatal injuries associated with cable reel shuttle cars can be classified under haulage, electricity or machinery, a review of the accident statistics is provided in the following discussion.

7.2 SHUTTLE CAR ACCIDENTS

In 1976, a total of 141 fatalities were reported in the coal mining industry. Haulage, machinery, and electricity accounts for nearly 50% of the total. In 1975, the same categories accounted for 56% of the fatalities (MESA, Reference 4). However, analysis of the accident frequency rate (injuries and fatalities) for recent years is not readily available. The most recent publication mentions statistics for the year 1970 (Moyer and McNair, Reference 6). However, the safety aspects of cable reel shuttle cars has already been studied in some detail by Curth (Reference 2), Theodore Barry and Associates (Reference 12), and Chalpin et al. (Reference 1). Table XLV summarizes the fatality data for the period 1966 - 1970 associated with shuttle cars.

7.3 SHUTTLE CAR SAFETY CONSIDERATIONS

A list of unsafe conditions and unsafe acts dealing with shuttle car operations in general is as follows (Curth, Reference 2):

1. Unsafe Conditions

    a.) Defective equipment including brakes, lights, steering, cables, etc.

Table XLV

Shuttle Car Fatality Accident Data, 1966 - 1970*
(Reference 12)

VICTIM'S ACTIVITY	NUMBER OF VICTIMS AND THEIR JOB CLASSIFICATION
Scaling Roof	1 Repairman
Inspection	1 Mine Foreman 1 Foreman
Repair	1 Repairman 1 Driller 1 Bolter Operator 1 Shuttle Car Operator
Tramming	9 Shuttle Car Operators 5 Repairmen 2 Apprentice Repairmen 1 Mine Foreman 1 Continuous Miner Helper 1 Foreman 1 Bolter 1 General Inside Man
Shuttle	30 Shuttle Car Operators 2 Repairmen 2 Foremen 6 Others
Other Activities	6 Others
TOTAL	74

*According to Mr. Thomason of MESA (202-235-1575) a detailed breakdown of shuttle car related accidents can be obtained from the Health and Safety Analysis Center (HSAC) in Denver. This will involve both time and costs for the search and summary of the HSAC computer data base. It is considered doubtful that the results of such a search and summary would add any information of significance (beyond that already at hand) to a rationale for determining the relative merits of the various shuttle cars under consideration, especially since there is no data on the new cars - flywheel and steam powered.

b.) Environmental conditions involving poor visibility, uneven floor

c.) Protective devices that are either defective or missing

d.) Poor planning such as the application of oversized equipment on adverse grades

2. Unsafe Acts

a.) Improper operation of equipment

b.) Failure to use protective devices

While the above is generally true with regard to all face haulage equipment, tethered shuttle cars have additional problems with trailing cables. These include:

a.) Damage caused by tramming
b.) Excessive cable tension and binding
c.) Improper anchoring of cables
d.) Inadequate short circuit protection
e.) Inadequate or improper application of temporary and permanent cable splices
f.) Other hazards caused by damp and wet floor conditions

The following discussion summarizes the results of several USBM studies dealing with the above mentioned hazards caused by cable reel shuttle cars.

Theodore Barry Study

A study by Theodore Barry and Associates (Reference 12) dealing with the hazards associated with underground coal mine production revealed that the shuttle car operator is the most dangerous job classification. This conclusion was reached on the basis of an analysis of fatal accidents during the period 1966 through 1970. The study also revealed that fatal shuttle car accidents are 3-1/2 times more likely to occur in seam heights less than 5 ft than in seam heights greater than 5 ft. Among the reasons hypothesized are a generally less satisfactory mine environment coupled with poor visibility in low and medium seams.

Major recommendations to minimize these hazards were a central seating arrangement and operator canopy. Other improvements included job tailored response controls, improved lighting and an internally powered shuttle car. The last mentioned improvement was occasioned by an observation that an untethered shuttle car could eliminate a prime safety and maintenance problem, i.e., trailing cables. The problems with trailing cables were not amplified. However, it was recognized that electrical fatalities (which were high for repairmen) can be reduced by the development of a permissible quick-disconnect coupling between the power cable and equipment. One solution mentioned was the need to develop a quick-charging station for use with battery-powered shuttle cars.

Since the elimination of trailing cables is one major recommendation in reducing the hazard potential of tethered shuttle cars, the following studies are summarized to develop objectively relationships between component and system failures in shuttle car operations.

7.4 FAILURE MODE AND EFFECT ANALYSIS ON CABLE REEL SHUTTLE CARS

There are several failure modes associated with shuttle cars in general. However, the trailing cable and the electric power are two that are unique to cable reel shuttle cars. Chalpin et al. (Reference 1) performed a Failure Mode and Effect Analysis on Cable Reel Shuttle Cars from which one concludes that the cable is a weak link with regard to safety since it introduces several safety modes that are not associated with untethered equipment -- specifically 4, out of a total of 23 failure modes identified, are attributed to the presence of the cable.

7.5 FAULT TREE ANALYSIS ON CABLE REEL SHUTTLE CARS

The report by Chalpin et al. (Reference 1) also includes a fault tree analysis on cable reel shuttle cars. The important point here is that the trailing cable introduces hazards which ultimately can result in the loss of lives.

In addition to these two analyses, Chalpin et al. also used a "forced decision analysis" method to arrive at a merit number for four power systems for shuttle cars, open cycle diesel, closed cycle diesel, battery/motor and cable reel/motor. This section of the report reveals that of the four power systems studied, open-cycle diesel is the most desirable for shuttle cars. Ramani and Kenzy (Reference 8) have summarized the health and safety aspects with regard to diesels, and point out that the maintenance of diesel engines and monitoring of the mine atmosphere for NO_x and CO are two important factors that need further attention.

7.6 ELECTRICAL ACCIDENT STUDY

Sinha, Stefanko and Ramani (Reference 10) have analyzed fatal electrical power accidents in coal mines for the period 1955 - 1970. Underground mines accounted for over 84 percent of the 200 accidents. Here, the face area including the shuttle car operations accounts for over 33 percent of the number of fatalities (Table XLVII). Table XLVI summarizes the major causes of fatalities.

Table XLVI

Causes of Electrical Fatalities
(1955 - 1970)
(Reference 10)

Lack of Suitable Technology	33
Personal Action/Inaction on the Part of the Victim	41
Lack of Adequate Supervision	12
Bad Design/Equipment	63
Improper Maintenance/Repairwork	51
Total	200

Table XLVII

Electrical Accident Analysis by Operations
(Fatalities, 1955-1970)
(Reference 10)

OPERATIONS	ELECTRICAL ACCIDENTS 1955-1970					
	DISTRIBUTION		SUB-TOTAL		TOTAL	
	NO.	%	NO.	%	NO.	%
Underground Mining						
a) Face Area	57	40				
b) Haulage System	64	45				
c) Others	22	15				
Total Underground Accidents	143	100	143	85		
Total Surface Accidents			26	15		
Total Underground Mining			169	100	169	85
Strip Mining						
Mining Area	11	46				
Others	13	54				
Total Strip Mining	24	100			24	12
Total Non-Mining					7	3
GRAND TOTAL					200	100

OPERATIONS:

Underground Mining

Face Areas: includes working faces, crosscuts, rooms, shuttle car track areas
Haulage System: includes main transport system, such as trolley haulage, belt conveyor, etc.
Others: includes power centers, pumping station, repair shops, etc.
Surface Accidents: includes surface substation, main repair shops, etc.

Strip Mining

Mining Areas: includes all working areas
Others: includes power substation, repair shops, etc.

Non-Mining

Includes preparation plants, long distance transportation services, etc.

Trailing Cable Accident Statistics

Mason (Reference 3) has reported on the electrical hazards in underground bituminous coal mines for the years 1972 and 1973. Electrical arcs and burns, burns (heat), and electrocution and shock accounted for over 96 percent of the accidents and injuries (Table XLVIII). Additionally, nearly 31 percent of the electrical arcs and burns are caused by cables (Table XLIX), usually as a result of defective splices and breaks in insulation. Sopko (Reference 11) has summarized the non-fatal and fatal cable accident reports (Table L).

Table XLVIII

Electrical Accidents and Injuries, 1972 and 1973
(Reference 3)

DESCRIPTION	1972 Number	1972 Percent	1973 Number	1973 Percent
Electrical arcs.............	242	37.35	156	20.63
Electrical burns............	276	42.60	452	59.79
Burns (heat)................	56	8.64	72	9.52
Electrocution and shock.....	49	7.56	53	7.01
Puncture....................	9	1.38	0	0
Chemical burn...............	6	0.93	4	0.53
Contusion...................	3	0.46	4	0.53
Sprain or strain............	2	0.31	6	0.79
Multiple injuries...........	1	0.15	8	1.06
Unclassified................	4	0.62	1	0.14
	*648	100.00	**756	100.00

*Represents a 63.84 percent sampling of all electrical accidents for 1972.

**Represents a 100 percent sampling of all electrical accidents for 1973.

Among the eight fatalities in Table L, three were associated with shuttle car haulage. In one case, the shuttle car frame was energized by a bare phase conductor as a result of the damage in the insulation and jacket of the cable near the entrance to the cable reeling unit. In the other two cases, the fatalities were a result of working on splices.

Table XLIX

Main Causes of Electrical Arcs and Burns
(Reference 3)

CAUSES	1972 Number	1972 Percent	1973 Number	1973 Percent
Cables........................	155	29.92	197	32.40
Trolleys......................	81	15.64	94	15.46
Switches......................	58	11.20	80	13.16
Haulage equipment.............	35	6.76	60	9.87
Electrical apparatus..........	25	4.83	31	5.10
Power and lighting circuits...	18	3.47	21	3.45
Mining machinery..............	15	2.90	31	5.10
All other.....................	131	25.28	94	15.46
	518	100.00	608	100.00

Table L

Cable Related Accident Summary
(Reference 11)

YEAR	NUMBER OF (NON-FATAL) DISABLING INJURIES	NUMBER OF FATALITIES
1972	81	0
1973	64	2
1974	27	3
1975	Not Available	3

REFERENCES

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11. P.A. Sopko, "Shielded Low-Voltage Trailing Cables for Underground Coal Mines," Unpublished M.S. Thesis, Pennsylvania State University, 1977.

12. Theodore Barry and Associates, Industrial Engineering Study of Hazards Associated with Coal Mine Production, Report to U.S. Bureau of Mines, 1971.

Section 8

WAYSIDE-TO-VEHICLE INTERFACE AND ALTERNATE CHARGING SYSTEMS

This section includes a study of alternate wayside to vehicle charging systems including mechanical and electrical interface methods. A mechanical interface implies that the spin-up motor is on the wayside, and the wayside-to-vehicle interface is mechanical. The electrical interface is necessitated by an on-board location of the spin-up motor.

Included in the discussion are these considerations:

- Charging station location
- Car location when charging
- Charging station requirements
- An evaluation of alternate vehicle-to-wayside connection systems

The impact on critical vehicle components is considered and a conceptual design of an electrical connection scheme is illustrated.

8.1 SUMMARY AND CONCLUSIONS

- The use of forms of energy other than electrical or mechanical from the wayside was considered. While a different form of energy would change the type of vehicle to wayside connector, it would not significantly improve the interface alignment problem.

- To accomplish an early demonstration of a practical flywheel-powered shuttle car, an electrical interface for spin-up power appears to offer the simplest and least risk development program.

- In the longer term, large scale use and production of flywheel shuttle cars might warrant the larger development effort of a completely mechanical flywheel system including spin-up power coupling and vehicle drive train.

- A wayside-to-vehicle interface system for the transmission of spin-up power has been defined. An initial concept of an electrical connector has been developed and is described in Section 8.4.4. Its size is roughly estimated in Section 8.4.4.

- As currently envisioned the wayside charging station includes three principal elements: a 250 kVA feed from a 750 kVA mine power center, a 25 to 30 cubic foot enclosure containing the load commutating inverter, and an assembly consisting of the car alignment guideway, the wayside connector, and its automatic engagement actuator.

- While the load commutating inverter and its control logic is a fairly complex piece of equipment, it is a straight forward design task. It is not intended to understate the design effort, but similar equipments operating at these power levels have been designed and operated. No new inventions are required and no major difficulties are anticipated.

8.2 ALTERNATE CHARGING SYSTEMS

8.2.1 General Considerations

The basic problem is to transfer a block of energy (up to 4.5 kW hrs) across a physically indefinite boundary, i.e., from a semi-fixed location of the wayside equipment to a variable car position.

A first consideration is the form of energy to be handled. Rotating shaft mechanical energy is required for the end application. Furthermore, the thrust of the study at hand eliminates consideration of primary fuel conversion systems. This leaves the following possible energy forms:

- Mechanical
- Electrical
- Hydraulic
- Pneumatic
- Electromagnetic
- Optical
- Thermal
- Acoustic

The first four of these energy forms require some sort of a mechanical connection for transmission while the last four are energy forms which may be radiated. It would be very convenient to use a radiant energy form to bridge the gap between shuttle car and wayside but unfortunately equipment to use these radiant energy forms is complex, inefficient, and/or low power.

8.2.2 Charging Station Location

There are three possible locations for the charging station:

- At the unload point
- At an intermediate change-out point
- At the miner

Locating the spin-up station at the unloading point is desirable for several reasons. First, the flywheel can be recharged simultaneously with the unloading activity thereby making dual use of the time required for unloading. Second, the unloading location is semi-permanent, i.e., it does not change very often, usually remaining fixed during the period of working a cut-block. Third, the car is stationary during the unload activity. Finally, the car is relatively carefully positioned at the unload point.

Locating the spin-up station at an intermediate change-out point suffers a major disadvantage in that the car is out of service for the entire time required for recharging. For two-car systems, this will reduce car availability thereby increasing miner wait time and reducing productivity. With three-car systems, runs on the PSU/USBM Simulator indicate no significant increase in miner wait time for spin-up times up to 60 seconds (plus 30 secs for unload). Charging at the change-out point requires a second accurate parking of the car in addition to parking at the unload point. Depending on the mine operation, it seems reasonable to expect that it might be desirable to change the location of the change-out point more frequently than changes in the unload point.

Locating the spin-up station at the miner has the advantage of dual usage of the load time. However, there are several drawbacks. First, the miner and shuttle car are in almost constant motion during the loading operation, and this adds significant complexity to the connection problem. Second, the "wayside" charging equipment would have to be located on, or at least near, the miner which presents awkward logistics problems. Finally, if electrical energy is used for charging, the connector would have to be permissible. Because of these reservations, the miner has been discarded as a possible location for the spin-up station.

8.2.3 Charging Regimes

There are two possible charging regimes. First, where the car is stationary (not tramming) at the unload point or at a change-out. The second possible regime is to charge while the car is in motion over a short distance - for instance, from the nearest cross-cut to the unload point. There are two advantages to the latter regime. A small amount of charging time is acquired while the car is tramming to the unload point, and the activities of acquiring the charging connector and unloading the car are separated in time and space. However, the use of a short heavy-duty cable, or multi-line trolley is not very attractive.

8.2.4 Charging Connection Requirements

- Assuming a 203 kW charging rate (4.5 kW hrs in 80 seconds) an electrical charging connector must be rated at

  500 volts dc at 406 amps

  or

  500 volts (line to neutral) three-phase, 234 amps per line, at a frequency of approximately 200 - 400 Hz.

- A mechanical charging connector must be rated at

275 horsepower

or

438 ft lbs torque at 3,300 rpm

The dc charging system implies that each car is equipped with its own load commutating inverter, but only two power connections are required. An ac charging system connector requires four power connections and up to six control signal connections (see Section 4), but the inverter is on the wayside and shared with other cars. The control signals could be multiplexed or telemetered if there is any technical or economic advantage to this approach.

- Preferably the connection system should be fully automatic with little or no operator intervention, other than driving the car into position.
- It should not be necessary for the operator to dismount from the car to assist the connection operation.
- The connect-disconnect function should preferably be accomplished with no delay to tramming or unloading activities.
- The connector must be so constructed and interlocked that no mechanical or electrical power flows until the connector is fully made up and no hazardous voltages or rotating components are accessible. This is primarily for safety but also to avoid arcing to improve reliability and minimize connector size.
- The design of the system for aligning and mating the two parts of the connector must include every consideration to minimize the hazards to personnel.
- For alignment of the vehicle to the connectors (see Section 8.4 for definition), six degrees of freedom must be accommodated: H, L, V and, to a limited extent, three axes of rotation
  - Forward-backward motion (H) has the greatest tolerance, tentatively ±12 inches.
  - Left-right position (L) has the next largest tolerance, tentatively ±6 inches.
  - Up-down location (V) has the highest tolerance, tentatively ±1.5 inches.

8.2.5 Connector Location

There are five possible locations for the connector system relative to its position on the car:

- Car bottom and floor - probably the least desirable location since it is most vulnerable to dirt, water, and damage. It is also inaccessible and not visible during the alignment operation.
- Car side and rib - visible and accessible but quite vulnerable to damage.
- End opposite discharge - accessible, not visible, vulnerable to damage.
- Discharge end - accessible, not visible, vulnerable to damage and in an awkward location where it would interfere with load/unload operations.
- Car top - most desirable, least vulnerable to damage, accessible and visible. May be difficult to find space where it does not interfere with visibility or other car functions.

8.2.6 Alternate Vehicle-to-Wayside Connection System

Several potential solutions to the spin-up power connection interface have been identified. These were evaluated against six different criteria as shown in Table LI. The identified systems are described below with appropriate discussions of the evaluations.

Table LI

Evaluation of Alternate Vehicle-to-Wayside Connection Systems

	SUITABLE FOR MINE	SAFETY	LITTLE DEVEL. REQD.	LOW COMPLEX-ITY	OPER. EASE	LOW REL. COST	OVERALL
1. Semi-Automatic Car Positioning	5	5	3	3	5	5	26
2. Elevating Connector on Car	5	5	3	2	5	4	24
3. Sliding Contactor/ Trolley	3	1	3	3	5	5	20
4. Manual Connector Positioning	5	5	3	4	1	5	23
5. Split Transformer	5	5	1	2	5	1	19
6. Change-Out Flywheel	5	4	1	1	1	3	15
7. Treadmill Drive Cars Traction Wheels	1	2	4	3	5	3	18
8. Clutch Drive Traction Wheel	5	1	4	3	4	3	20
9. Fifth Wheel	5	1	4	3	4	3	20

Note: Methods 1, 2, 4 & 6 could conceivably be either electrical or mechanical energy transfer.
Ratings, 5 = most desirable, 1 = least desirable

Method 1 - Semi Automatic Car Positioning

Method #1 envisions both the car and wayside connectors in fixed positions on the car and wayside. Coarse alignment is achieved by mechanically guiding the vehicle into position with a ferryboat or car wash slip type of berth. Vernier alignment is accomplished either by connector tolerances or by a second guideway acting on the connector parts as they engage. This solution appears to be the most practical of those conceived thus far. An electrical version of this concept is described in detail in Section 8.4.

Method 2 - Elevating Connector on Car

Method #2 envisions an elevating connector mounted on a trolley pole or pantograph type of mechanism on the top of the car. At the spin-up station the car connector raises to engage a rail mounted on the mine roof transverse to the direction of car travel. The wayside connector would then automatically traverse the rail to mate with the car connector. Due to the extended movements of both parts of the connector this scheme is probably only practical with an electrical interface. Since this concept is somewhat more complex than Method #1, it is given lower complexity and cost ratings.

Method 3 - Sliding Connector/Trolley Line

Method #3 is a short trolley line or sliding contactor. Because the number of circuits add complexity, this scheme is probably only practical for a two-wire dc type of charging system. This in turn adds the cost of a load-commutating inverter on each car. The method is given a low safety rating since the "trolley wires" will have partially exposed high voltage at least during the spin-up time.

Method 4 - Manual Connector Positioning

Method #4 attempts to trade the complexities of automatic alignment for manual intervention. The car connector would be mounted in a convenient fixed position while the wayside connector might be mounted from the roof or rib, counterbalanced with flexibility and freedom to be moved to the mating location. Some means of automatically providing insertion force would probably have to be included. This method is not rated very highly because of the need for operator intervention and the fact that little complexity is avoided.

Method 5 - Split Transformer

Method #5 involves a physically divided transformer with the secondary windings and half of the core mounted on the car. The primary windings and core are affixed to the wayside. Advantages include no open electrical connections and somewhat easier alignment with greater tolerance. Disadvantages include added car complexity and the cost of an on-board load commutated inverter on each car.

Method 6 - Flywheel Package Interchange

Method #6 would be to physically transfer a discharged flywheel package from the car to the wayside spin-up station in much the same manner that battery packs are transferred. This scheme reduces the spin-up station peak power requirements but introduces an awkward handling problem. The method eliminates spin-up time delays, but since the changeout probably could not be performed at the unload point, it will add a fixed delay of many seconds for flywheel transfer. Finally, this method does not greatly alleviate the connector design or alignment problems.

Method 7 - Treadmill Drive

Method #7 involves positioning one or more of the car's traction wheels on a treadmill and pumping mechanical energy back to the flywheel through a bi-directional drive train. While there is a certain elegance in the simplicity of this scheme, it is not highly rated for several reasons. Ideally, the treadmill should be buried in the mine floor, requiring added installation difficulty and exposing it to water and dirt. The presence of the powerful high-speed treadmill would create a safety hazard. Finally, the need for bi-directionality may add cost to the drive train. An on-board load commutating inverter would be required if the flywheel output is electrically coupled.

Method 8 - Individual Vehicle Wheel Drive

Method #8 is an alternate to Method #7. It involves automatically jacking up one of the traction wheels and coupling mechanical energy to the wheel through some device such as a large jaw clutch or driving dog. While this method is more suitable for mine service in that it gets the spin-up equipment up off the mine floor, it still suffers from the safety hazards and bi-directional drive train drawbacks of Method #7.

Method 9 - Fifth Wheel Drive

Method #9, another means of coupling mechanical energy to the car, might be to employ a fifth rubber tire wheel mounted on a vertical shaft on the car. The fifth wheel would be driven from a wayside wheel or roller and could couple energy directly to the flywheel through suitable clutching and gearing. This scheme trades whatever problems might be associated with a bi-directional traction drive train for a new completely separate mechanical power linkage. The safety concerns of high-speed high-power rotating machinery remains unchanged.

8.2.7 Critical On-Board (Vehicle) Components

For the case of an on-board electric motor spin-up, the most critical component associated with spin-up is the motor/alternator itself, and this is discussed in detail in Section 4.3 in this report. The electrical interface connector between the vehicle and wayside equipment is another critical component. The mechanism for making up this connection poses some design challenges, and these will be treated in Section 8.

For the case of mechanical spin-up from a motor located at the wayside the critical on-board components are the clutch between the flywheel and the input power shaft and the power input connector. Three different types of clutches have been considered: a dry plate friction clutch, an overriding or centrifugal clutch, and a synchronized jaw clutch. Any one of the three types will work. The synchronized jaw clutch is favored since for a given torque rating it is the smallest. The power input connector represents a design challenge since for optimum performance and minimum size it should have a very tight minimum tolerance fit with the driving socket. Weighing against this, the plug will be exposed to the dusty abrasive mine environment and subject to considerable abuse. Both of these factors mitigate against tight tolerances. The design considerations for the splined plug are:

$$\text{Horsepower} = 2\pi NT/33{,}000$$

or

$$\text{Torque (in ft lbs)} = \frac{33{,}000 \times \text{hp}}{2\pi \times \text{rpm}}$$

and substituting

$$T = \frac{33{,}000 \times 275}{2\pi \times 3{,}300} = 438 \text{ ft lbs}$$

A 2-1/4 inch diameter, 10-splined plug three inches long will transmit this torque.

These considerations indicate that alternate mechanical spin-up methods would effectively reduce overall system complexity only if direct mechanical coupling is employed between the flywheel and the car's propulsion drive train. In addition, the development program required for an "all mechanical" flywheel-powered shuttle car is perceived to be longer, more costly, and have more high risk elements than a program based on an electromechanical approach.

These nine methods are evaluated against six criteria in Section 8.1. No attempt has been made to put weighing factors on the rating criteria, and, therefore, the spread in overall ratings is not very great.

To accomplish an early demonstration of a practical flywheel-powered shuttle car, an electrical interface for spin-up power between the car and wayside equipment appears to offer the simplest and least risk development program. This is due to the inherent simplicity and flexibility of electrical power transmission and connection systems coupled with the availability of system components such as: cables, circuit breakers, and controls. Of the interface methods conceived thus far, Method #1 (Semi-Automatic Car Positioning) is the most attractive.

In the longer term, large scale use and production of flywheel cars might warrant the larger development effort of a completely mechanical flywheel system including spin-up power coupling and vehicle drive train. A completely mechanical system shows promise of being simpler, more efficient and ultimately less costly.

8.3 MECHANICAL SPIN-UP SYSTEMS AND INTERFACE CONNECTOR

8.3.1 Alternate Systems

Figures 79 and 80 portray the mechanical components associated with mechanical spin-up interface concepts. Figure 79 shows a mechanical spin-up system used in conjunction with a flywheel employing electrical output to the vehicle propulsion system. The two universal joints and splined sleeve allow the splined driving socket to have the necessary degrees of freedom for alignment with the splined receptor plug of the flywheel. The synchronized jaw clutch is exactly the same principle as that employed in an automatic "stick shift" synchromesh transmission. As a matter of fact, a prototype implementation of this concept might very well use a synchronized jaw clutch from an urban bus or truck transmission; the horsepower and torque ratings are in the right range. Although no reduction gears are shown in Figure 79, it would probably be desirable to step down the 10,000 rpm of the flywheel by something like 2 or 3 to 1. An input shaft operating in the range of 3,000 to 5,000 rpm would alleviate problems with a high-speed shaft operation through the case vacuum seal and high-speed drive components external to the flywheel enclosure.

Figure 80 depicts a completely mechanical system using mechanical spin-up power and mechanical output to the vehicle propulsion system. In this Figure a 3 to 1 reduction gear is coupled to the flywheel. To accommodate gyroscopic action induced by vehicle pitch and roll, the flywheel must be "soft mounted" to allow it some limited amount of tilt in any direction. Accordingly, universal joints and a splined sleeve are shown at the output of the flywheel package to allow for relative motion. Spin-up power input is the same conceptually as shown in Figure 79. A second clutch is shown to disconnect power to the traction drive train components. The principal reason for this clutch is to conserve energy during waiting periods. The clutch could be either synchronized jaw-type or a conventional dry disc automotive type. Traction speed and power would not be modulated by the clutch, as it is in an automobile, since a clutch is very inefficient in this mode. Tractive effort and speed along with flywheel speed changes would be accomodated by the power split, hydrostatic, constantly variable transmission, and, if required, a gear box. A difficulty with this concept is the potentially awkward location of the power take-off shaft from the flywheel package.

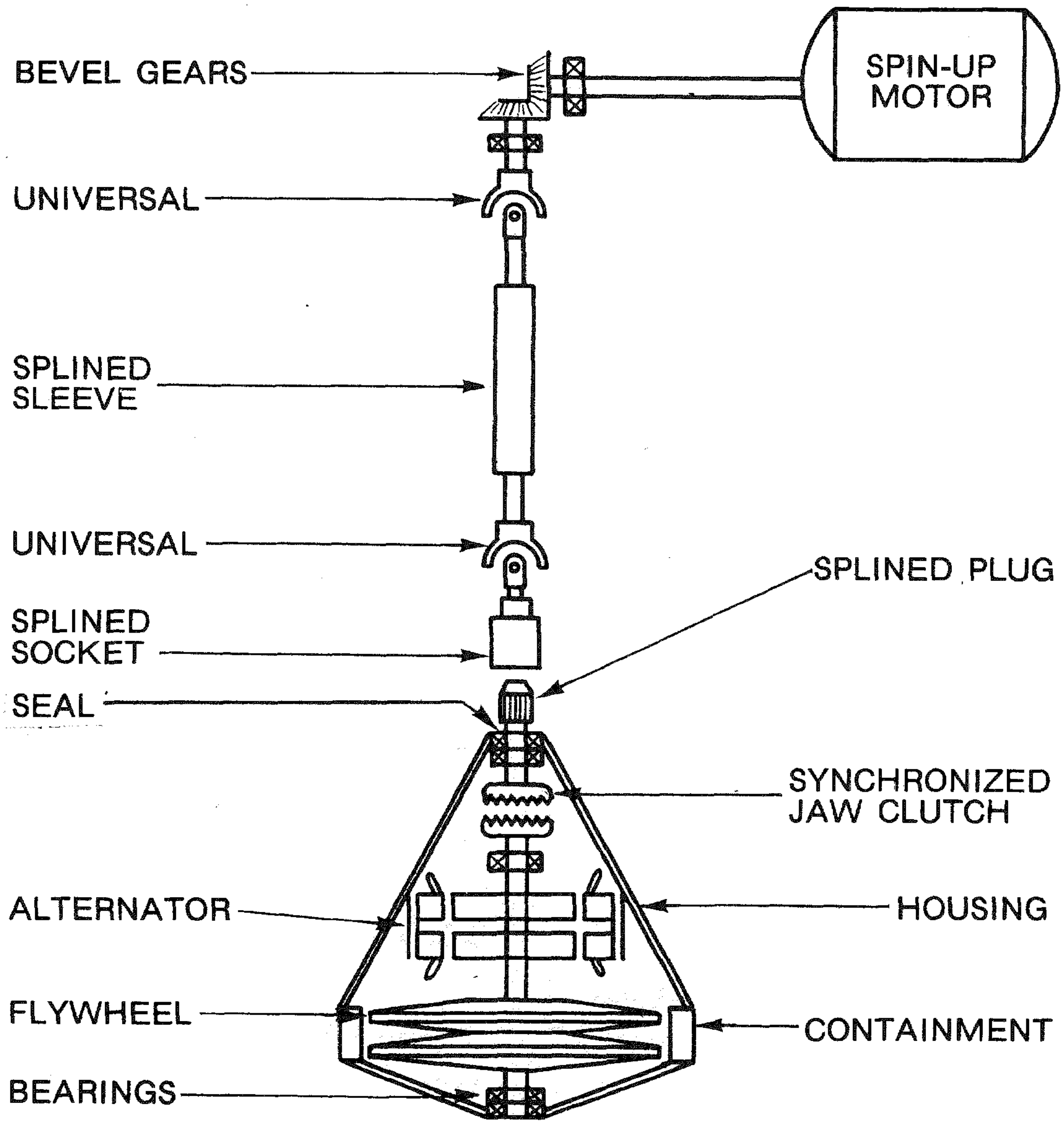

Figure 79. Mechanical Spin-Up with Electrical Vehicle Pictorial

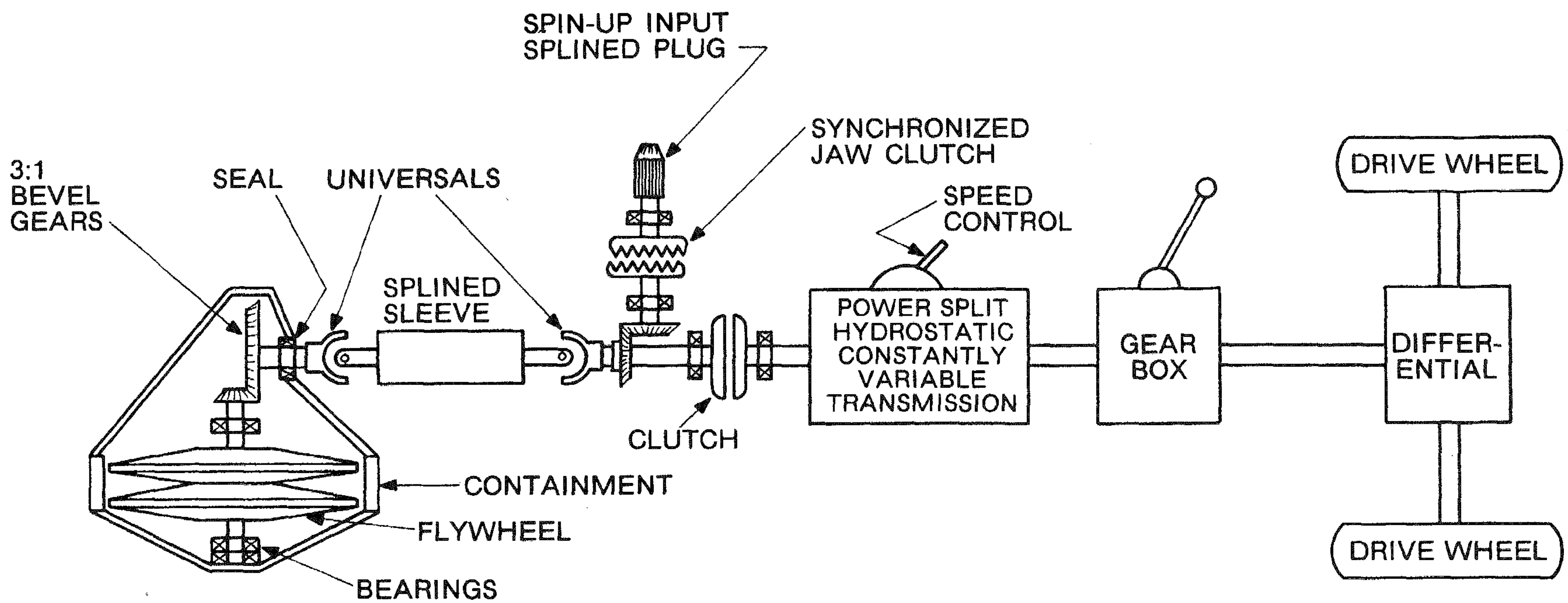

Figure 80. All-Mechanical Drive -- Spin-Up and Power Train Elementary Pictorial

8.3.2 Wayside Spin-Up Motor

If a mechanical spin-up system is used, a wayside spin-up motor is required. The logical type of motor to use is a dc commutator-type because of its commercial availability and control simplicity. While larger than an ac motor, physical size is much less of a constraint than for an on-board application.

The motor speed/power can be controlled with a simple phase-controlled rectifier rather than the more complicated load commutated inverter. A traction-type series-wound dc propulsion motor could be employed. Because of the intermittent duty, a motor selected for the duty cycle may be used. Brush/commutator wear would not be a serious concern. Due to the high torque variable speed characteristics of a traction motor the speed monitoring and control system is greatly simplified. For example, no special circuitry is required when starting up a completely discharged flywheel from zero rpm.

A General Electric 1254 Motor rated at 275 horsepower in railroad traction duty is suitable for this application. Detailed motor specifications are shown in Table LII. Since this motor is rated at 3750 rpm maximum, a suitable gear box would have to be designed to provide an output of 10,000 rpm for direct coupling to the flywheel. However, interface considerations, which were discussed in Section 8.3.1, indicate the desirability of mounting the reduction gear box on the shuttle car. The development effort for this alternative, other than the mechanical linkage interface to the car, is very attractive since the motor- and phase-controlled rectifier are essentially off-the-shelf designs.

Table LII

Application Information GE-1254-A1 Traction Motor

Ratings:	One hour starting cold	- 295 hp, 425 V, 560 A, 2240 rpm
	Continuous dc Full Field	- 275 hp, 425 V, 520 A, 2300 rpm
Temperature Rise:	140°C Armature, 155°C Field	
Maximum Speed:	3750 rpm	
Weight:	1996 pounds	
Size, Case	25-3/8 inches diameter 28-inches long	
Ventilation:	Separate 860 CFM	

The estimated costs of a dc motor wayside spin-up station are shown below:

Power Center (Transformer)	$25,000
Phase Controlled Rectifier	4,000
dc Motor	5,000
	$34,000

This figure cannot be directly compared with the cost of the car-mounted ac motor and wayside load commutated inverter since it does not include the added complexity of the mechanical interface or on-board components. At this point in time the major trade-off is between the magnitude of the development programs required for an all mechanical versus an electromechanical vehicle drive and spin-up system.

Another alternative which should be considered would be to use an essentially constant speed motor coupled through a variable speed transmission to accommodate the speed range of the flywheel. Such a system might well use components of the automatic transmissions found on urban buses. As with the dc traction motor, this system would require a step-up gear box to match the 10,000 rpm top speed of the flywheel. This alternative does not offer any significant advantage over the traction motor drive. While the motor starter/controller is simpler than a phase controlled rectifier, the added complexity of the automatic transmission and its control system more than offset the saving.

8.4 ELECTRICAL INTERFACE CONNECTOR REQUIREMENTS AND CONCEPT

8.4.1 Basic Environment Guides

Selected features of the environment which bear on the connection system concept and design are:

(a) The ambient atmosphere is not explosive* but may be moist and may contain sulfur-bearing gases such as H_2S and SO_2.

(b) The ambient temperature is lower than $40^{o}C$.

(c) The contacts must engage and disengage promptly, such as within 2 or 3 seconds, but this may establish the need for mechanized final alignment and power drive to engage and disengage the contacts.

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*Note: Although the mine atmosphere is not explosive at the location of live operation and the connector will be de-energized at all other times, it may be necessary to make the portion of the connector attached to the car "permissible" since it does enter the face area of the mine.

(d) Contacts will engage and disengage without circuit power on them, i.e., non-arcing.

(e) When engaged, the contacts will be protected from easy access by people and from easy accidental contact with machinery or metal objects.

(f) Spin-up recharge must be simultaneous, or nearly so, with some part of the normal mining cycle. The only appropriate time appears to be the coal unloading time.

(g) Contact renewal or maintenance shall be infrequent.

8.4.2 <u>Secondary Guides</u>

Secondary guides to design concepts which would be compatible with the basic guides are:

(a) The contacts will engage in a butt-type of motion with a minimum or very low component of abrasive action. Relative motion in the abrasion direction of about 3 to 5 thousandths of an inch may be desirable if the atmosphere contains enough sulfur-bearing gas to tarnish copper visibly in 24 hours.

(b) The contacts shall make initial engagement with definite force determined by the preloading of the movable contact springs. The recommended amount of initial contact force is an empirical value related to the current through the contacts. The values in Table LIII are compatible with switchgear and contactor practice and will lead to low-contact temperature rise, low wear and tolerance of occasional fault conditions. The contact force on final contact engagement will be larger by the product of spring gradient and wear allowance (new contacts). This increase will not be critical for electrical performance but will be determined by reasonable spring design, space and actuator considerations. Anticipated values of force increase are about 50% of the initial contact force.

(c) Contact wear allowance will be about 3/16" - 1/4". This represents the amount a pair of contacts could wear away before failing to engage. Practically, the contacts are renewed before the allowance is fully worn away. (The total amount of allowance is called "wipe" in contactor and switchgear terminology. Note that it does not imply abrasive motion.)

(d) The contact face will be backed by sufficient copper to hold the average temperature rise over an operating cycle to less than 65°C. In a simple design, this metal will be identical to the contact face. Cross section values are recommended in Table LIII.

Table LIII

CROSS-SECTION VALUES

CONTACT CURRENT RATING (AMPERES)	CONTACT INITIAL FORCE (POUNDS)	CONTACT MINIMUM CROSS SECTION NEAR THE CONTACT FACE (SQUARE INCHES)
300	8±1	1
600	16±2	1-1/2
1200	32±4	2

(e) The contact life under normal operating conditions will be more than 500,000 engagements and disengagements.

(f) The described design concepts will be applicable to continuous ratings of effective current from about 300 to 1200 amperes with appropriate variation of size of components indicated by Table LIII.

The study thus far has not fixed the parameters which determine the current but the range considered there varies from 375 amperes at 80 to 90 second spin-up, dc at 600 volts to 1111 amperes at 30 second spin-up, dc at 600 volts to an on-board inverter. Note, however, that it will be permissible to establish the required current rating on an RMS basis combining the spin-up time with the minimum off time to the subsequent spin-up.

8.4.3 Alignment Tolerances

This is the general arrangement of contacts that is proposed:

- A set of contacts mounted as an assembly on a wayside site
- A set of contacts mounted as an assembly on a site on the flywheel car
- Alignment of the car contacts with the wayside assembly as part of the controlled entry of the car into its coal-unloading berth but possibly requiring response to a simple operator act such as pushing a button
- Motion of one contact assembly, to be called the "movable* contact assembly" so that its contacts will engage with the contacts of the other contact assembly, to be called the "stationary contact assembly"

*The terms "movable" and "stationary" refer to the motion of the contact assemblies with respect to their supports. They have no necessary relation to the movable car and the stationary wayside.

- Subsequent disengagement of the contacts in response to a simple electrical or mechanical signal.

This conception includes the use of control contacts in the contact assemblies to prevent contact engagement unless the power circuit is de-energized by a suitable interrupter, which permits closing of this interrupter after engagement and which requires opening of this interrupter as a condition for contact disengagement.

Alignment motions will be discussed on the basis of three mutually perpendicular axes:

H - the Horizontal direction parallel to the forward and back motion of the longitudinal axis of the car near its unloading position.

L - The Lateral direction in a nominally horizontal plane, but at right angles to H.

V - The Vertical direction at right angles to both H and L.

It is assumed that:

- There are several unloading positions in the mine.
- These remain fixed for time intervals long enough to justify reasonably stable road surfaces and relatively simple guide structures at each unloading position.
- Such guide structures will be positioned identically in each unloading berth with respect to the unloading position of the car.
- The wayside contact assembly at each unloading berth will be positioned identically with respect to the guide structure.
- Each contact assembly on the car will be positioned identically on the car with respect to some reference such as the front end and height above ground.

An alternative to a specially prepared hard road surface at each coal unloading position, positioning of the leading wheel guide elements in each such surface and accurate alignment of the wayside portion of the connection apparatus with the road surface would be the use of a simple, portable steel base carrying the guideway elements as integral parts and mounting the wayside portion of the connection system on a stanchion-like support integral with the portable base. Under this arrangement, the alignment of the wayside and car portions of the contact assemblies would not be sensibly disturbed by wear or shifting of the base because both portions would shift together. The portable base structure is envisioned as made, principally, of two structural steel channel bars nearly a car-length long, tied together by bolted steel pieces. The bolted construction would make it practical to relocate the base to new unloading positions by carrying it on a shuttle car without protruding from the sides.

It is intended to seek alignment of the contacts before engagement by car positioning only for all three axes, H, L and V is possible, but the contemplated design will permit additional fine adjustment in one of these axes should it be necessary. The design conditions required for alignment depend on tolerance of contact position deviations from nominal along the three axes. These are examined herewith:

(a) H Axis. The car will be trammed to a fixed berth at the unloading dock and remain there throughout the unloading process. The operator will brake the car to final stop at a position indicated to him by visual alignment. An aid to the operator will be some suitable marks on the car and dockside, or if it is desired to aid the operator more, a simple electrical installation of limit switches or photocell or induction relays will generate a signal to him or automatically stop the car in the manner of elevator levelling. The length of an acceptable stopping zone along the H axis, thought to be feasible with manual stopping, is about 12 to 24 inches. It is estimated that manual stopping within 24 inches would not add any significant time to that required to tram the car to its assigned unloading position, distances within 12 inches would not add more than 2 seconds to this time, but distances within 12 inches might add significantly to the docking time.

These estimates will require experimental confirmation.

(b) L Axis. As the car enters its unloading berth, steering of the car will be taken over from the operator by passive ground equipment which will compel the leading end of the car to travel in an assigned path. This equipment will be a guideway for the leading wheels, much on the principle of the simple guideways commonly used in carwash installations for passenger automobiles, but somewhat more sophisticated. In as much as each flywheel car will enter its unloading berth many times during the normal life of its tires, it is important to minimize abrasion of the tire sides by the guides and any potential increase in driving power. Therefore, instead of making the guideway of simple horizontal metal bars, the guide members will be free metal rollers on vertical axes disposed and fixed on the ground so the rollers will press against the outside of the front wheel tires when they deviate from the preassigned path. The positions of the rollers will form a tapered entrance, so that the operator will not have to exercise any extraordinary skill to bring his car into the guideway.

The amount of clearance that must be provided in the guideway will depend in part on the behavior of the tires under load. This is examined with the aid of Figure 81. It is clear that if tire inflation is low, the width of the tire will change from W to (W + ΔW). However, if the inflation is enough to expand the tire to its nominal shape under maximum load, the width will not change much with reduced load. The correct inflation, cold, will be that which will hold the tire's nominal shape (full tread, only, in contact with the road) under maximum load. It will be necessary for maintenance practice to assure this minimum inflation, but good practice for reasons independent of alignment would also demand this.

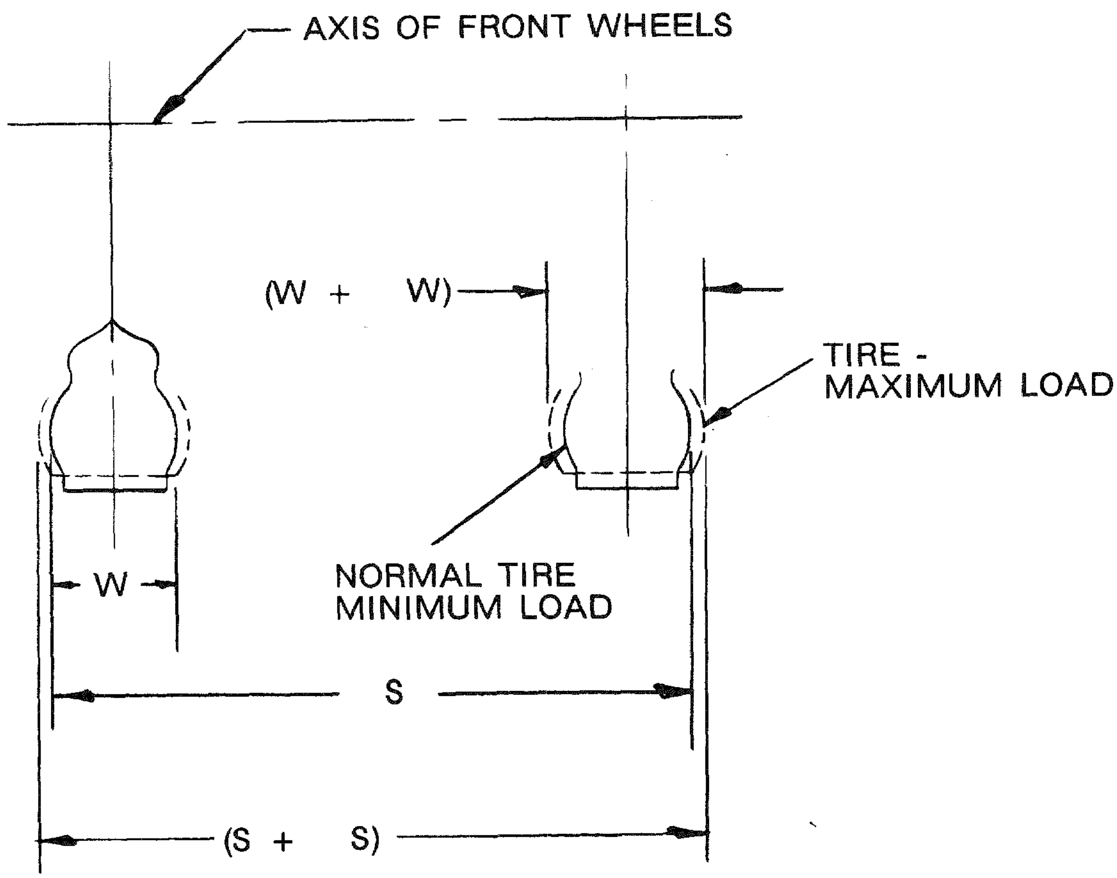


Figure 81. Tire Deflection Under Load

If the mine practice permits somewhat lower inflation, alignment still could be maintained by somewhat larger contact components, but the proposed conception makes no provision for alignment in the event of a complete or nearly complete deflation.

The tolerance of position in the L-direction must allow for the factors shown in Table LV. The meaning of the estimated ΔW and ΔS will be evident from Figure 81.

Table LIV

L Direction Factors and Tolerance

Factor	Tolerance Inches ΔW	ΔS
Tire width as a function of manufacturing and tire grades	1/8	1/8
Tire width as a function of side wear (outside only)	1/8	1/4
Tire width as a function of effective inflation, including the effect of temperature	1/4	1/4
Tire width as a function of car load from empty car to maximum load	1/4	1/4
Manufacturing tolerance of front wheel separation		-
Manufacturing and installation tolerance of guideways (all unloading berths in one mine)		1/8
Manufacturing tolerance in placement of contact assemblies on car with respect to wheels and on wayside with respect to guideway		1/4
Clearance		1/4
Total		1-1/2"

(c) V Axis. The variation of relative positions of the contact assemblies parallel to the V Axes will not be affected directly by tramming but must include th factors in Table LV.

Table LV

V Axis Factors and Tolerance

Factor	Tolerance Inches
Tire diameter as a function of manufacture or tire grades	1/8
Tire tread wear	1/2
Tire tread deformation under load variation, empty car to maximum load	1/8
Tire diameter as a function of inflation, including the effect of temperature	1/8
Change of position of road surface with respect to wayside contact assembly as a result of road wear or ground shifting (Adjust contact assembly position, if necessary, to hold tabulated value)	1/4
Manufacturing and installation tolerances of contact assembly mountings	1/8
Total	1-1/4

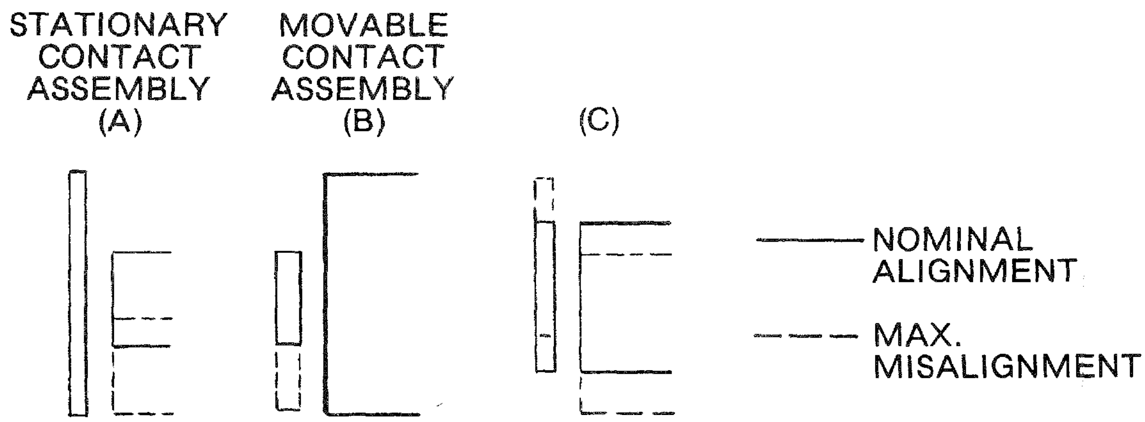
8.4.4 Design Concept

The following salient points seem to be firm guides to establishing a design concept:

- The movable contact assembly should be on the wayside, and the stationary contact assembly on the car because the movable contact assembly is more complex, larger, requires power and there will be fewer of them if so located.
- The movable contact assembly should contain the resilient contacts. It is simpler to have the flexible conductors that are needed for contact resilience and those that are needed for the stroke of engagement and disengagement on the same assembly. It may be possible to combine them. This is a general practice in switchgear. Double-break contacts are not considered because they would not be simple under the condition of the relatively large adjustment needed for alignment.
- The large tolerance in car positioning in the H direction is easily and simply accommodated by track-shaped stationary contacts such as copper bars, 24 inches long (possibly 12 inches long). This appears preferable to providing means for fine adjustment of car position.

From these considerations a decision can be made to use track-shaped stationary contacts, 24 inches (possibly shorter) on the cars and resilient, movable contacts on the wayside. The natural position for the stationary track-shaped contacts is on the side of the car, as in Figure 83 where they may be recessed into the car body and possibly even covered with an automatic door closure for safety and permissibility. In this case, the movable contact assembly extends and retracts from the wayside in a direction parallel to L. The stroke of the movable contact assembly in the L-direction will include the L-direction tolerance, contact wipe (wear allowance) and sufficient air gap to provide electrical and mechanical clearance. A pair of limit switches sensing when the movable contact assembly moves beyond the point of initial contact of unworn contacts through the initial wipe of unworn contacts will stop the actuator in the contact-closing direction and automatically compensate for the particular value of L-direction tolerance that may prevail on each closing operation.

The potential misalignment in the V-direction may be accommodated by making the stationary or movable contacts, or both, wider than the dimension required for contacts not misaligned. For example, if the potential misalignment in the V-direction is 1-1/4 inches, as estimated in Table 83, and the contact width in the V-direction that would be required if the contacts were perfectly aligned is one inch, the contact extension in the V-direction could be 5/8-inch above and below on either the stationary or movable contact, or 5/16-inch above and below on each as illustrated in Figure 82, or other intermediate combinations. If the value of



CONTACT EFFECTIVE WIDTH = 1 in
MAXIMUM MISALIGNMENT = 5/8 in
MAXIMUM SPACE REQUIREMENT = 1-5/8 in

Figure 82. Contact Alignment

required tolerance in the V-direction is found to be substantially greater than 1-1/4 inches, it will be undesirable to use this method of accommodation. In this case by means of a guideway or by an adjustment of the movable contact assembly vertical position, responding to a sensor measuring the V-direction misalignment will be necessary.

The movable contact assembly is illustrated in Figure 84 From Figure 83 the overall size of the connector, with signal contacts and protective hood, is roughly estimated at 24 inches high by 24 inches long (along the side of the car). The stationary (car-mounted connector) will be about 4 inches deep. The movable wayside connector will be about 24 inches deep including 12 inches of engagement travel, actuating mechanism and mounting stanchion.

8.5 WAYSIDE CHARGING STATION CONCEPT

As currently envisioned, the wayside charging station includes three principal elements:

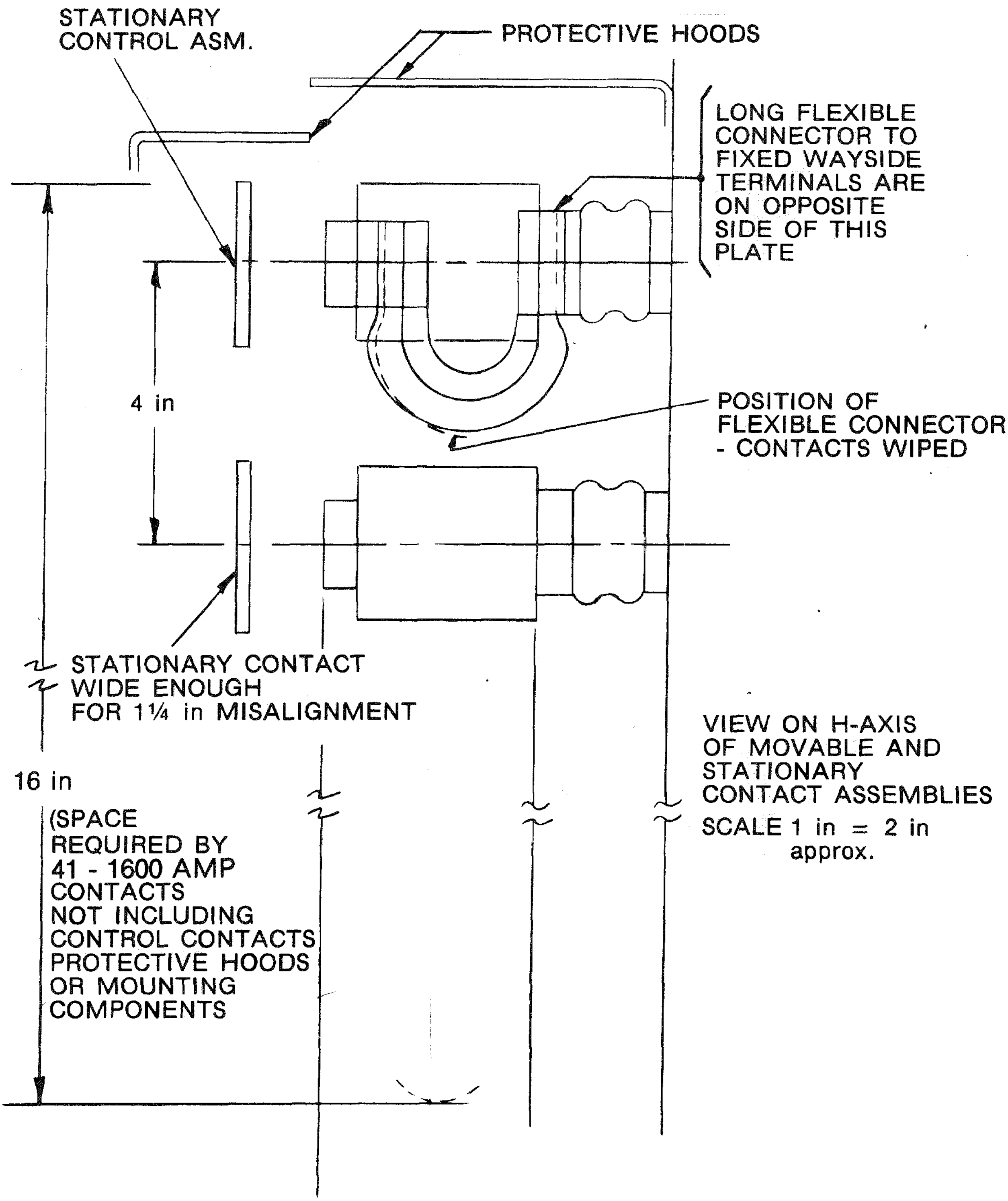
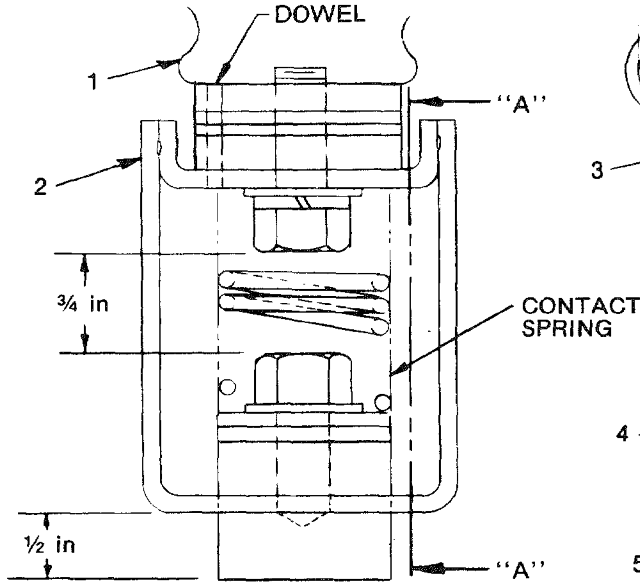
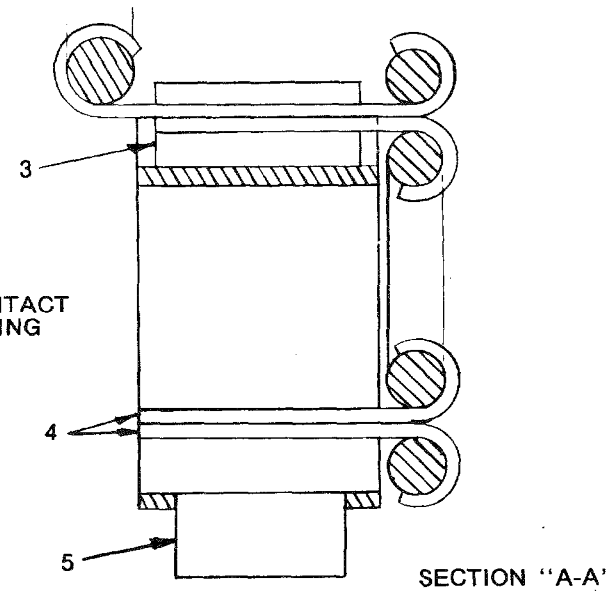
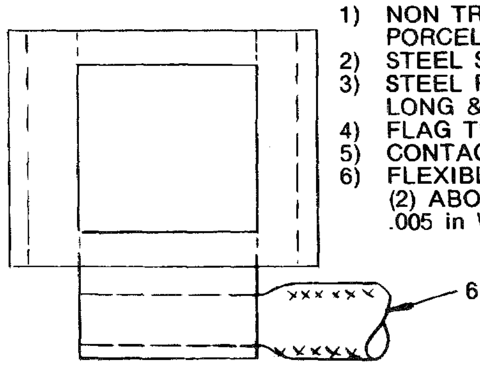


Figure 83. Charging Connector Details





1 MOVABLE CONTACT ASM.-600 AMP
SCALE 1 in = 1 in (APPROX.)



PART NO.

1) NON TRACKING INSULATOR PORCELAIN PREFERRED
2) STEEL SUPPORT - RESISTANCE WELDED
3) STEEL PLATE TO MOUNT THE LONG & SHORT FLEXIBLE CONDUCTOR
4) FLAG TERMINAL FOR FLEX. CONDUCTOR
5) CONTACT - COPPER 1¼ in X 1¼ in – 1¼ in X 1¾ in
6) FLEXIBLE CONDUCTOR CABLE (2) ABOUT 144000 cm EA, .005 in WIRE BARE

Figure 84. Moveable Contact Assembly - 600 Amperes

- A 250 kVA feed from a 750 kVA mine power center
- A 25 to 30 cubic foot enclosure containing the load commutating inverter and control logic
- An assembly consisting of the car alignment guideway, the wayside connector and its automatic engagement actuator.

For convenience of discussion, these three elements are divided into seven major components as shown in the single line block diagram of Figure 85. Each component of the diagram will be discussed.

8.5.1 Mine Power Center

The mine power center is essentially a special purpose three-phase step down transformer mounted in a suitable permissible housing. Power centers are available with a wide variety of options such as: high voltage interrupters, low voltage circuit breakers, surge arresters, surge capacitors, permissible connectors and rectifiers. For this application the power center must have a high enough rating such that the surge load (approximately 250 kVA of the wayside charging station does not seriously exceed the rating of the transformer. It was previously determined that a 750 kVA rating would be adequate. This rating was based on the assumption that other mining equipment at the face did not present a load much in excess of 500 kVA. It is also assumed that the power center to be used is equipped with a rectifier rated for at least 250 kVA continuous duty. It is not important that the recitifier bank be included in the mine power center; it can just as well be considered as part of the load commutating inverter and included as a part of that package.

8.5.2 Load Commutating Inverter

The load commutating inverter is discussed in some detail in Section 4.7. Basically it is a bank of silicon controlled rectifiers arranged to invert the dc input power to a variable frequency, variable voltage ac power to drive the spin-up motor. This is not a piece of commercially available equipment. A load commutating inverter must be designed and built to meet the requirements of this application.

8.5.3 Control Logic

The control logic is really a part of the load commutating inverter and would be mounted in the same enclosure. It is shown as a separate block to emphasize its importance. The control logic performs several important functions: first, by means of a suitable circuit breaker, it prohibits the application of power to the interface connector until a control signal is received

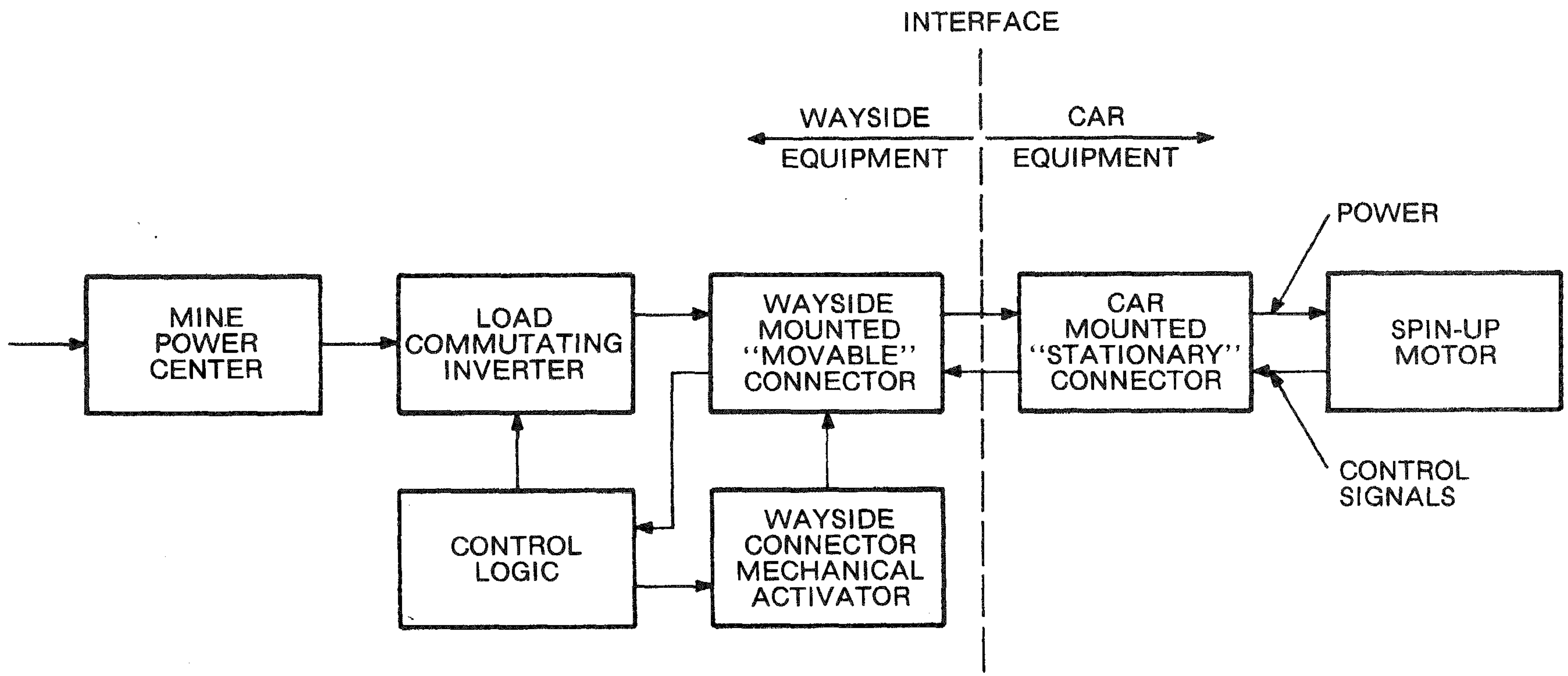


Figure 85. Wayside-to-Vehicle Electrical Interface Block Diagram

indicating that the wayside and car connectors are fully made up or mated; second, the control logic, acting on a signal from a suitable sensor, operates the wayside mechanical activator; third, the control logic generates the required trigger pulses for the SCRs. The trigger pulses are varied in phase and frequency depending on speed signals received from the spin-up motor. Since the control logic knows the speed of the flywheel, it also knows when the flywheel is at full charge. At this point it stops firing the SCRs, opens the circuit breaker, and operates the wayside connector mechanical activator to disengage the interface connector.

There is another aspect to the disconnect logic sequence. First, suitable control signals must be provided to inhibit car movement while the interface connector is engaged and especially while spin-up power is flowing across the interface. Second, some sort of override function must be provided to allow operator intervention to terminate the spin-up cycle at any time and initiate the disconnect sequence. This may be a separate control or it may be connected to the cars acceleration control with a signal indicating that full charge has not been achieved.

The flywheel proper is a highly stressed component. Although adequate safety factors have been employed in its design, it is essential that its speed never be allowed to exceed 10,000 rpm. As a consequence it is important that redundant speed sensing and control circuitry be included in the control logic of the wayside spin-up station.

8.5.4 Wayside Connector Mechanical Activator

This is a mechanical device which physically moves the wayside connector into engagement with the car connector. The actual motion may be accomplished with a pneumatic or hydraulic cylinder or possibly with an electro-mechanical device such as an electric motor-gear train-rack and pinion. A pneumatic cylinder would be the best choice except for the requirement for a source of compressed air.

8.5.5 Wayside and Car Mounted Connectors

An initial concept of the connectors is described in Section 8.4.2. The connectors are presently envisioned to consist of four power carrying contacts plus a number (perhaps as many as six) of signal contacts all suitably guarded. The contacts are conceived to be of the butt type, held in pressure contact with a force of eight, or more, pounds per contact. The need for substantial engagement/disengagement force will exist with any type of connector capable of handling this power level (200 kW). Also, any connector at this power level will be fairly big and heavy. These considerations, coupled with the requirement for fast operation, conspire to mandate the need for an automatic mechanical activator.

8.5.6 Wayside System Considerations

The load commutating inverter, the control logic and possibly the three-phase rectifier, if it is not included in the mine power center, may be packaged in an enclosure of from 25 to 30 cubic feet (see Section 4.6). It is not necessary nor desirable (because of its size) to locate this package right at the unload station. On the other hand, because of the special multi-conductor cable required between the LCI and the wayside connector equipment, it is desirable to locate the LCI/control package as near as possible. Perhaps it can be situated outby down the same entry as the unload station or in a nearby unused crosscut.

As described in Section 8.4.3 the wayside connector and its mechanical activator must be located at the unload station. The wayside connector and car alignment guideway will be an integral unit, fixed-mounted or perhaps attached to the ratio feeder, hopper, or whatever equipment is employed to receive coal at the unload station.

When in operation, the LCI will be handling some 200 plus kW of power. Assuming 90% efficiency, heat losses will amount to 20 kW (28 horsepower). Even at 50% or less duty cycle, there is still a lot of heat to be dissipated. Furthermore, the junction temperature of the silicon controlled rectifiers in the LCI must be held to 75°C or less. A forced air-cooled heat exchanger will be required.

Section 9

DESIGN CONSIDERATIONS OF A FLYWHEEL PACKAGE ON A SHUTTLE CAR

The design considerations involved with incorporating a flywheel energy storage system in a suitable face haulage vehicle have been studied by the contractor and by Jeffrey Mining Machinery Division of Dresser Industries under subcontract. Jeffrey's work is reported in detail in Appendix D of this report.

The car system design considerations, the selection of a vehicle, and the identification of problem areas are discussed. Also included are recommendations for further study in subsequent design phases.

9.1 SUMMARY AND CONCLUSIONS

- To accommodate the physical size of a 4.5 kW hr flywheel package, at least 270 cubic feet (13,500 pound) load-carrying capacity and four-wheel drive - the basic chassis and drive train of the Jeffrey steam powered Ramcar - offer a minimum development-effort vehicle for a mine demonstration.

- Visibility studies have shown that, because the flywheel package projects above the tractor structure, a smaller energy flywheel yields better visibility than a larger flywheel. The flywheel does not change in diameter with size but increases in height. Higher seam heights (above about 60 inches) allow raising the operator seat, thereby increasing visibility.

- The RAMCAR traction motor must be operated during spin-up to provide hydraulic power for unloading. Means must be provided to allow the traction motor to operate with the variable voltage power provided by the wayside equipment for spin-up.

- There is a trade-off between high voltage - low current versus low voltage - high current for the traction motor and flywheel motor/alternator. The trade-off involves cost, size, and practicality of the wayside-to-vehicle connector versus the cost and complexity of the load commutating inverter, the traction motor, and the motor/alternator. An evaluation of the alternatives should be made before finalizing the voltage.

- Vehicle drive train components have been selected to minimize the amount of car and drive train development effort. There is need to evaluate the impact of drive train efficiency on face haulage system cost effectiveness including initial cost versus mine productivity ($/ton).

- Two independent (General Electric and Jeffrey) conceptual design approaches to the vehicle equipment cooling requirements concluded that a single, series-operated heat transfer system would be adequate for cooling the on-board components. This conclusion and the entire cooling system should be reviewed in the next phase of the program.

- The average temperature rise of the rotating components (flywheel and motor/alternator rotor) has been calculated for a worst-case mission (4,5 kW hrs) and found to be within acceptable limits (65°C). The dynamic or transient temperatures present a more difficult analytical problem. This analysis is best done during the subsequent design stages.

9.2 SELECTION OF CAR

The work of Section 2, Mission Analysis and Energy Storage Requirements, led to the conclusion that a flywheel with a useful energy storage capability of 4.5 kW hrs was required to meet the worst case mission profile. This subsequently led to the work in Section 4.4, Flywheel Design and Construction, which defined a flywheel of approximately 42.5 inches in diameter. A survey of available shuttle car designs quickly led to the conclusion that a tractor-trailer type of car would have to be used. The volume normally occupied by a battery pack or engine in a tractor-trailer car is most readily adaptable to accommodate the flywheel package and the necessary electrical equipment.

In Section 5.5.4, Shuttle Car Payload Versus Productivity, it was concluded that the greater the payload capacity of the car the more economically attractive a flywheel-powered shuttle car system would become. This led to the conclusion that the payload capacity should be at least 270 cubic feet (13,500 pounds).

To provide optimum results in any subsequent demonstration of a flywheel-powered shuttle car system, it was concluded that a four-wheel drive system should be used. This minimized the possibility of loss of traction in bad bottom conditions.

The vehicle requirements just mentioned narrow the choice of options to three:

- Design a new vehicle specially for the job

- Modify an existing design, say by adding four-wheel drive to an existing two-wheel drive vehicle

- Utilize the chassis and drive train of the recently developed Jeffrey steam-powered RAMCAR, substituting the flywheel and an electric motor for the steam engine

The third option represents the least amount of development effort and hence cost. As far as known at this point, choosing the third option does not compromise any other part of the system nor the overall performance of the demonstration vehicle.

9.3 CONSIDERATIONS IN CAR SYSTEM DESIGN

Figure 86 is a single line block diagram showing the major components of the car system and its interface with the flywheel system. The individual components represented by blocks will be discussed briefly.

9.3.1 Wayside Connector

The functions of the wayside connector are to transmit spin-up power from the wayside charging station to the vehicle and to provide control signals, such as flywheel speed and vehicle status, to the wayside control logic. It may be desirable to provide an automatic, or semi-automatic cover for the car mounted connector. This cover would provide added safety and protection to the connector from dirt and mechanical damage. A main circuit breaker is shown immediately after the wayside connector. Its purpose is to disconnect flywheel generated power from the wayside connector during normal car operation.

9.3.2 Flywheel Package

Power flow to the flywheel is bi-directional. Power flows into the flywheel motor/alternator during spin-up and out during normal car operation. During car operation the motor/alternator output voltage is held essentially constant by adjusting the alternator field excitation. A difficulty may arise during the spin-up phase of the operating cycle. The hydraulic power required to unload the vehicle is supplied from the traction motor via a power take-off from the transmission. This means that the traction motor must be running, albeit at a fraction of its full horsepower. During spin-up the motor/alternator may be provided with a variable frequency/variable voltage input to produce maximum torque at all speeds and hence minimum spin-up time. Traction motor speed can be maintained with reduced input voltage by reducing its field excitation. Reduced field means reduced torque, but there will still be ample horsepower for the unloading operation. An alternative might be to switch the traction motor supply to a separate constant voltage supplied by the wayside equipment. A disadvantage of this approach is the require-

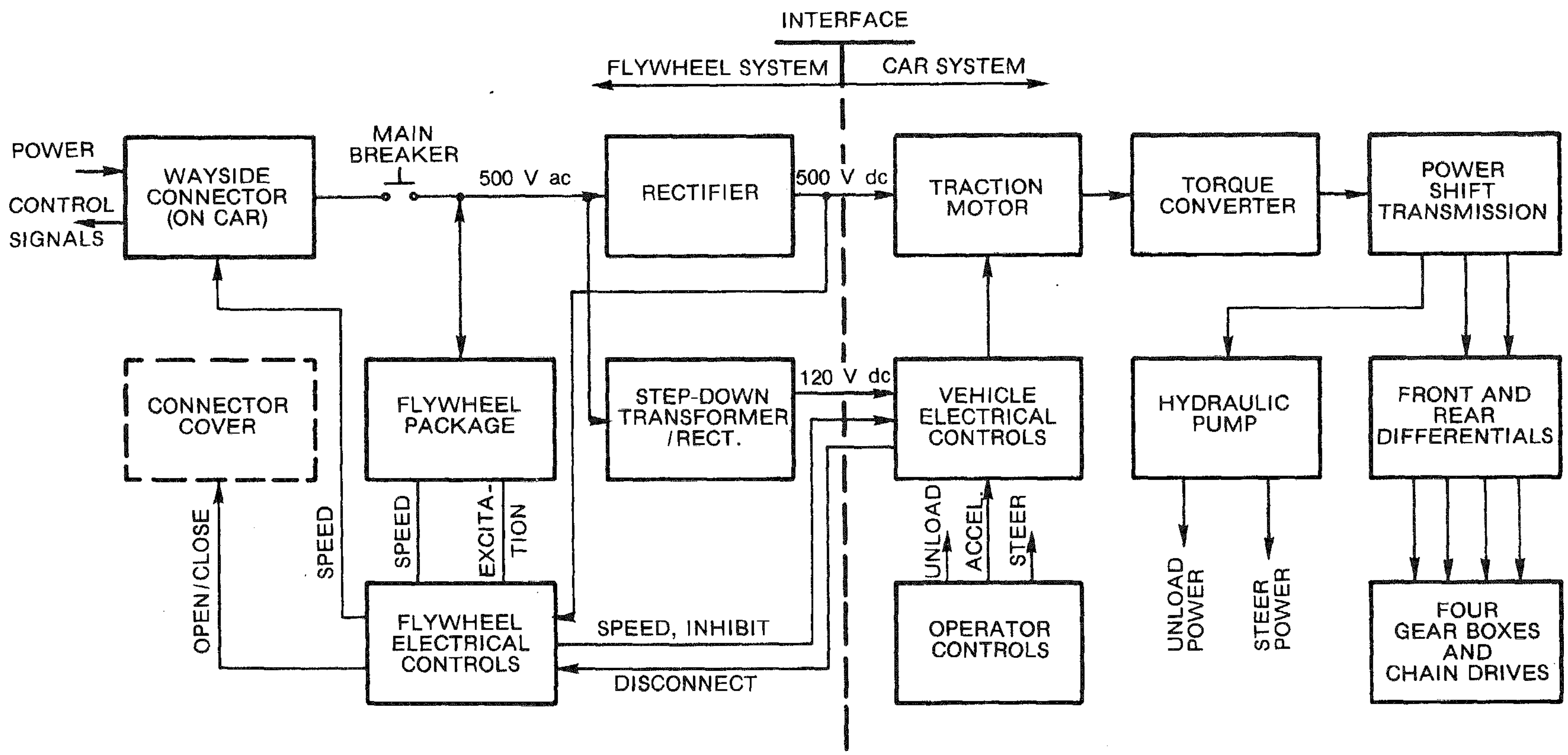


Figure 86. Car Systems Block Diagram

ment for additional power contacts in the wayside connector. Another alternative is to utilize the variable voltage through a phase-controlled rectifier to power the traction motor with constant voltage. This is discussed further in Section 9.3.4.

9.3.3 Flywheel Electrical Controls

The first of the flywheel electrical controls is the motor/alternator field excitation control. This circuit holds the alternator output voltage to the traction motor constant as the flywheel slows down. It also provides the proper excitation to the motor during spin-up. Another control function is to provide redundant speed signals back to the load commutating inverter and to the vehicle operating console (fuel gauge). Those vehicle operator commands which affect the wayside equipment, e.g., interrupt charging before completion, are routed through the flywheel controls package and the wayside-to-vehicle connector. Using signals from suitable sensors, the flywheel controls would open and close the wayside connector cover, if one is employed. Flywheel controls would sense the spin-up mode, flywheel speed, and dc voltage to the traction motor and provide information to properly adjust the field excitation of the traction motor during spin-up.

9.3.4 Rectifier

The rectifier is needed to convert the variable frequency alternating current output of the motor/alternator to direct current for the traction motor. At this time it is envisioned as a straight forward three-phase full-wave rectifier consisting of six diodes. Another solution to the difficulty presented by the variable voltage during spin-up noted in Section 12.3.2 would be to use a phase-controlled rectifier comprised of six silicon controlled rectifiers. It would be desirable to avoid this added complexity, but it may be the best solution when all other factors are considered.

9.3.5 Step-Down Transformer

The car electrical controls such as the traction motor starting contactor, lights, and indicators require three to five hundred watts at 120 volts dc. Accordingly a step-down transformer and rectifier will be required to reduce the nominal 500 volts ac output from the alternator. Output voltage can be held constant during spin-up by the use of a simple SCR phase control similar to those commonly available for dimming lights.

9.3.6 Traction Motor

The traction motor is presently sized at 75 horsepower, 500 volts. The 500-volt rating is higher than necessary, or even desirable, in a commutated motor of this size. A lower voltage traction motor would mean a lower output voltage from the alternator. This in turn would require lower input voltage and higher current during the spin-up phase. Higher input current would mean larger contacts in the wayside to vehicle connector. Of even greater significance, higher input current will increase the losses in the load commutating inverter. Higher current would also require larger, or even doubling the number of, SCRs in the LCI. A trade-off study to determine the optimum traction motor voltage should be conducted before the design is finalized.

9.3.7 Vehicle Mechanical Drive Train

Details of the mechanical components in the vehicle drive train along with the criteria and rationale for the selection of each component are covered in Appendix D. The major mechanical components of the vehicle drive train are listed below:

- Hydraulic torque converter
- Automatic powershift transmission
- Front and rear differentials
- Reduction gear boxes and chain drives (one for each wheel)

The design criteria used in selecting these components are:

- Maximum torque required with bad bottom conditions
- Highest performance (maximum speed) achievable with good bottom conditions
- Maximum efficiency realizable over a variety of mission profiles
- Availability of "standard" drive train components

In reviewing the recommendations of Appendix D, it will be noted that emphasis has been placed on minimizing the amount of car and drive train development effort. This is in consonance with the principle thrust of the program which is to determine the feasibility of a flywheel-powered face haulage vehicle. On the other hand, the impact of possibly lower drive train efficiency on the economic viability of the system has not been evaluated. (Lower efficiency means higher power consumption per mission. This in turn leads to longer average spin-up times which <u>may</u> increase miner wait times thereby reducing productivity.)

9.4 VEHICLE EQUIPMENT COOLING

Four pieces of car-mounted equipment have been identified as requiring consideration with respect to heat dissipation or cooling. These are: the traction motor, the torque converter, the flywheel motor/alternator, and the electronic equipment associated with the flywheel package. In addition, heat transfer from the rotating components of the flywheel package require special consideration since they will be operating in a partial vacuum.

9.4.1 Traction Motor and Torque Converter

The traction motor and torque converter are considered to be a part of the basic vehicle and its mechanical systems. The cooling requirements for these components is covered in Appendix D. The motor is forced-air cooled with an integral fan. The torque converter employs a separate heat exchanger to cool its hydraulic oil. The air used to cool the motor is first drawn through the heat exchanger. Since this heat transfer system for the torque converter is already on-board, it is planned to increase its capacity and utilize it to cool the electronic equipment and flywheel package. Cool oil from the liquid to air heat exchanger would be routed first to the on-board electronics package since it should be operated at as low a temperature as possible. The coolant would next go to the flywheel package since it is desirable to maintain as great a temperature differential as possible between the rotating parts and the flywheel package container. Finally, the oil would pass to the torque converter and then return to the heat exchanger. Other coolant routing systems are feasible including paralleling heat sources and series-parallel combinations. A detailed design study of the heat transfer system is warranted.

9.4.2 Electrical Equipment Cooling

There will be a large number of kilowatt hours of energy passing through the shuttle car and consideration must be given to the removal of energy losses which are in the form of heat. The most significant of these losses are found in the motor-alternator, the ac to dc rectifier, and the mechanical drive train. The purpose of the following discussion is to make a preliminary assessment of cooling requirements. A set of worst-case conditions is assumed:

Energy used per trip	- 4.5 kW hrs
Minimum trip time requiring full energy	- 1.33 mins Charge Time
	.45 mins Load Time
	4.00* mins Tram Time
	5.78 mins Total

*Based on 3 mph ave. tram speed on bad bottom, 1100 ft round trip

Energy use rate = 4.5 × 60/5.78 = 47. kW hrs per hour or 160,000 Btu per hour.

Spin-up motor losses = 8% (approx.)
Alternator losses = 8% (approx.)
Total motor/alternator losses = 16%

or 7.5 kW (10 Horsepower)

The total heat loss from a surface may be expressed as:

$$q = (h_c + h_r)\, A\eta\Delta T$$

where: q = heat loss in Watts

h_c = heat conduction coefficient in still air

h_r = heat radiation coefficient

A = cooling surface in square inches

η = effectiveness factor for heat exchanger (80%)

ΔT = temperature differential in °C (150-20 = 130°C) 150°C represents assumed surface temperature

$$h_c = 2.21 \times 10^{-3} \left(\frac{\Delta T}{L}\right)^{0.25} \text{ watts/in}^2\,°\text{C}$$

where L = vertical length in inches (12")

$$h_c = 2.21 \times 10^{-3} \left(\frac{130}{12}\right)^{0.25} = 4 \times 10^{-3}$$

$$h_r = 1.47 \times 10^{-10}\, \varepsilon\left(\frac{T_s + T}{2} + 273\right)^3 \text{ watts/in}^2\,°\text{C}$$

where: ε = emissivity = 0.8

T_s = surface temperature = 150°C

T_A = ambient temperature = 20°C

$$h_r = 1.47 \times 10^{-10} \times 0.8 \left(\frac{150 + 20}{2} + 273\right)^3 = 5.4 \times 10^{-3}$$

Rewriting the previous equation for heat loss and putting in the values:

$$A = q/(h_c + h_r)\eta\Delta T$$

$$A = 7{,}500/(5.4 + 4) \times 10^{-3} \times 0.8 \times 130 = 7{,}672 \text{ in}^2$$
or 53 ft^2

This result indicates a number of things. First, it will be necessary to use a heat transfer system employing oil, freon, water-ethylene glycol or some fluid to extract heat from the motor-alternator to an exposed radiator. The 53 square foot figure is conservative in that it is based on worst-case tramming energy requirements and on still air. The figure is liberal since it assumes the maximum allowable surface temperature of 300°F (150°C). 150°C is probably too high a temperature for the stator of the motor-alternator. It is certainly too high for a surface which might be accidentally contacted by personnel. The large area and high temperature practically mandates the use of forced air through the heat exchanger. The heat transfer coefficient for forced air is:

$$h_c = 11.2\sqrt{\frac{V}{L}} \times 10^{-4} \text{ Watts/in}^2\text{°C}$$

where: V = air velocity in ft/min

L = length of surface parallel to air flow

assuming $V = 300$ ft/min

$L = 3$ in

$$h_c = 11.2\sqrt{\frac{300}{3}} \times 10^{-4} = 11.2 \times 10^{-3}$$

and decreasing the heat exchanger temperature to 100°C:

$$A = 7500/(11.2 + 4) \times 10^{-3} \times 0.8 \times 80 = 7709 \text{ in}^2$$

$$\text{or } 54 \text{ ft}^2$$

Fifty-four square feet of surface could be obtained in a finned radiator two feet wide, one and one-half feet high and three inches thick.

The heat losses from an ac to dc rectifier on the car is calculated in a similar manner. Losses in the rectifier and traction motor control may be kept quite low, approximately 5 percent. In addition, unlike the motor-alternator, the 4.5 kW hrs of energy only passes through once. Hence the rectifier/control losses = 4.5 kW hrs × 0.05 = 225 W hrs and the energy use rate = 225 × 60/5.78 = 2336 watts.

For optimum performance and reliability the junction temperature of solid state devices should be kept as low as possible, less than approximately 100°C for rectifiers and 75°C for silicon controlled rectifiers. Substituting these numbers in the previous equations and assuming forced air cooling:

$$h_c = 11.2\sqrt{\frac{300}{3}} \times 10^{-4} = 11.2 \times 10^{-3}$$

$$h_r = 1.47 \times 10^{-10} \times 0.8 \left(\frac{100 + 20}{2} + 273\right)^3 = 4.3 \times 10^{-3}$$

$$A = 2{,}336/(11.2 + 4.3) \times 10^{-3} \times 0.8 \times 80 = 2355 \text{ in}^2$$

$$\text{or } 16 \text{ ft}^2$$

This says that a heat exchanger for the propulsion power rectifier with forced air cooling might be of the order of one quarter cubic foot.

In case the load commutated inverter is on board, the same calculations, assuming 75°C operation, 10 percent losses and providing power for the shuttle car, shows a heat exchanger surface area requirement of 64 square feet. Such a heat exchanger would require a volume of about one cubic foot with forced air cooling.

9.5 CRITICAL ON-BOARD COMPONENTS AND DESIGN AREAS

This study has identified the following critical on-board components and design areas requiring additional analysis work:

- Selection of optimum traction motor voltage, considering the cost impact on spin-up connector and wayside equipment.
- Operation of the traction motor during the variable voltage conditions which occur while spinning-up.
- As the program progresses from the concept to the design phase, it will be necessary to review thoroughly the efficiencies of all on-board components with particular attention to the impact of efficiency on flywheel size and system effectiveness.

Section 10

EMERGENCY PROCEDURES - DISCHARGED FLYWHEEL

Flywheel energy storage systems on shuttle cars require a plan to rescue a stranded vehicle in case the flywheel runs down at a point removed from the charging station. This could be caused by a trip requiring an unusual amount of energy, operator error in charging the flywheel, excessive wait time, extended time between usage, as in over a week-end, or by failures of the flywheel equipment.

10.1 SUMMARY AND CONCLUSIONS

- Towing is by far the simplest and most practical solution to rescuing a stranded shuttle car. It is the method most likely to be used in actual practice regardless of what other means are provided.
- In the press of recovering a stranded vehicle there is a high probability that the second shuttle car will exhaust its energy supply. In this event, another means of recovery must be provided.
- An electric cable appears to be the best secondary means of returning an electrically propelled vehicle to the charging station.
- For an all mechanical shuttle car a 15 to 25 horsepower portable electric motor appears to be the simplest secondary means of recovery. The motor could be powered with an electric cable or storage batteries.

10.2 ENERGY REQUIREMENTS

The problem of returning a vehicle to the charging station in the event of a fully discharged flywheel readily reduces to two possible courses of action: tow it or provide some portable source of power. Towing is by far the simplest possible solution since it involves no additional equipment. However, the towing solution has limitations. First, although it is completely feasible, the use of other types of mining equipment is not considered a valid solution. Use of the miner as a tow vehicle would be too slow, awkward, and costly. It is doubtful that a loader or roof bolter would have enough tractive effort available, especially if the spent shuttle car was loaded or mired. So the towing alternative is for practical purposes restricted to the use of the second shuttle car as a towing vehicle. It is anticipated that each shuttle car will be provided with a "fuel gauge" (a tachometer on the flywheel), so that each operator will know at all times how much energy remains in the car. Use of one shuttle car as a towing vehicle for the other is, to a first approximation, restricted to those mission profiles requiring 3 kW hrs of energy. The available 4.5 kW hrs on the towing vehicle will use 1.5 kW hrs tramming out plus 3 kW hrs tramming back with two cars. If a shuttle car runs out of energy at the far end of a 4.5 kW hr mission (and this is bound to happen), then towing is not a completely satisfactory solution for all situations. However, the towing car could tram back to the charging station one or two times before the tow is completed, thereby extending the range.

10.3 ENERGY AVAILABLE FROM RESCUE VEHICLE

One source of portable power supply would be the energy stored in the second shuttle car. Ac or dc power could be coupled, via suitable cable and connectors, between the two vehicles. This scheme is subject to the same mission total energy limitations as previously discussed. There is an alternative recovery procedure involving multiple trips of the rescue vehicle between the spin-up station and the stranded vehicle. To estimate the feasibility of this approach, it is necessary to make the following assumptions:

Vehicle weight, empty	26,000 lbs
Capacity (6 ft seam)	14,000 lbs
Rolling resistance	300 lbs/ton
Grade	5%
Tram distance	550 ft
Drive train efficiency	70%
Rescue vehicle	Unloaded
Stranded vehicle	Loaded
Auxiliary equipment loss	250 Watt-hours
Total energy available	4.5 kW hrs

Then:

Rescue vehicle rolling resistance loss, round trip = $2 \times 550 \text{ ft} \times 26{,}000 \text{ lbs} \times 300/2000 = 4.3 \times 10^6 \text{ ft lbs}$ $= 1.6 \text{ kW hrs}$

Rescue vehicle grade losses cancel on a round trip

Stranded vehicle rolling resistance loss =
550 ft x 40,000 lbs x 300/2000 = 3.3×10^6 ft lbs
= 1.2 kW hrs

Stranded vehicle grade loss =
550 ft x 40,000 lbs x 0.05 = 1.1×10^6 ft lbs = 0.4 kW hrs

And:

Total energy available	4.5 kW hrs
Less auxiliary losses (2 vehicles)	0.5 kW hrs
	4.0 kW hrs
Times 70% efficiency	2.8 kW hrs
Less rescue vehicle rolling resistance	1.6 kW hrs
Energy available for rescue	1.2 kW hrs

Since the stranded vehicle requires 1.6 kW hrs of energy, it could be recovered with two trips of the rescue vehicle. It should be noted that while the conditions assumed do not add up to a 4.5 kW hrs round trip mission for one car, the conditions are severe enough to represent a conservative estimate. With these conditions it would also be possible for the rescue vehicle to tow the stranded car back to the spin-up station in two trips.

If the shuttle cars are each equipped with on-board load commutating inverters, it would be possible to transfer energy from the rescue vehicle to spin-up the flywheel of the stranded vehicle via a temporary jumper cable. The use of the on-board inverter complicates the on-board equipment. In this scheme some energy would be lost due to the efficiencies of the L.C.I. and the inductor motor/alternator. The rescue energy available would become:

1.2 kW hrs x 90% (L.C.I. efficiency) x 92% (motor efficiency) x 92% (alternator efficiency) = 0.9 kW hr

and the recovery could still be accomplished with two trips of the rescue vehicle albeit with a little less energy to spare.

10.4 PORTABLE SPIN-UP STATION

It is also possible to consider transporting the spin-up station to the stranded vehicle. In this case the second shuttle car is the only practical means of moving the spin-up station. Special equipment would probably have to be provided for loading and unloading the spin-up station. And, of course, the primary power supply cable feeding the spin-up station would have to be dragged along. This alternative is not considered to be as attractive as simply taking out an electric cable to provide traction motor power.

To continue the consideration of portable power sources, it is necessary to have some estimate of the horsepower or kilowatt, capacity required. Previously it was determined that some 1.6 kW hrs of energy were required to overcome rolling resistance and grade losses. This energy must be further increased by dividing by the 70% efficiency of the vehicle drive train for a total of 2.3 kW hrs.

Further assuming an average speed of 200 feet per minute, the one way trip time = 550/200 = 2.75 min and:

$$\text{Horsepower} = \frac{2300 \times 60}{746 \times 2.75} = 67$$

Noting that bad bottom conditions have been assumed, it might be reasonable to expect the spent car to "limp" home at 1/3 speed; in which case, about 25 horsepower (19 kW) of emergency power would be required.

10.5 EVALUATION OF ALTERNATE EMERGENCY POWER SOURCES

The provisioning of emergency power to a distressed vehicle immediately divides itself into two aspects: first, the type of power, e.g. electrical or mechanical , and second, how it can be interfaced with the vehicles propulsion system. The following types of emergency power sources suggest themselves:

- Electric cable
- Battery
- Internal combustion engine
- Winch and cable
- Compressed air

A summary evaluation of each of these emergency power supply means is shown in Table LVI. A discussion of each emergency power supply method follows.

Assuming an electric motor is used for car propulsion, an electric cable similar or identical to those currently used for tethered shuttle cars is by far the simplest solution. Nineteen kilowatts load at 600 volts is only some 32 amperes; hence, a rather small wire (AWG 8-10) is adequate. The use of an extension cable suffers from a certain amount of handling problems, but then so do all of the other emergency power schemes. The cable would have to be equipped with a permissible connector at

Table LVI

Emergency Power Source Evaluations

	SAFETY	MINE ENVIRONMENT	COMPLEXITY	DEVELOPMENT EFFORT	INTERFACE WITH CAR*	TOTAL
Electric Cable	1	2	2	2	2	9
Battery	1	2	0	1	1	5
I.C. Engine	2	0	1	0	1	4
Winch	1	0	2	2	2	7
Compressed Air	2	2	0	0	1	5

Notes: Ratings: 2 = good, 1 = fair, 0 = poor
*Assumes electric motor powered car

the car end, of course. This method is only rated fair from a safety viewpoint since all of the hazards of an electrical tether cable are present.

A conventional automotive type lead acid storage battery can readily store 100 ampere hours which at 12 volts equals 1.2 kW hrs. Two or three batteries would provide all the energy required to return the disabled car to the spin-up station. The only catch is that the voltage (36 V) is too low to operate the traction motor. The alternatives are a specially designed high-voltage battery, a dc to dc converter or a portable low-voltage traction motor. All of the alternatives are feasible but not especially attractive from the viewpoint of complexity.

A portable lightweight internal combustion engine such as those used on recreational vehicles or outboard motor boats might be considered. The big difficulty here is lack of compatibility with the mine environment. Such an engine would have to be completely equipped with antipollution devices and fire prevention equipment. Since no lightweight engine is known to be available with such equipment, a development program would be required.

A winch with tow cable appears as a simple straightforward solution. However, if the winch is located at or near the spin-up station, an elaborate assembly of pulleys and anchors would have to be temporarily installed to guide the cable around corners. Such an operation would be time consuming and hence the scheme is rated low in compatibility with the mine environment. A portable winch could be temporarily mounted on the second shuttle car. In this mode auxiliary power would have to be provided for the winch and this alternative reduces to a variant of the recovery schemes previously discussed.

Another means of energy storage is compressed air. A standard high pressure gas cylinder holds about 1 cubic foot at 2000 psig. This represents about 500,000 foot pounds of energy at best. For the trip, 6.3×10^6 foot pounds of energy are required, so as many as 12 standard gas cylinders might be required for one emergency powered trip. Based on logistics alone this alternative is not as attractive as other solutions.

A small amount of emergency power will be required after a prolonged shutdown. Only one car can be left at the spin-up station, and it must be assumed that in 24 hours the flywheels will have lost all their energy through friction and windage. The second car could be towed and/or pushed to the spin-up station, but a short electric cable also appears to be a convenient answer.

Appendix A

DETAILED COST ANALYSIS

A.1 SHUTTLE CAR SPECIFICATIONS

Data for the Joy 18SC13DC tethered car (used for all runs in the PSU/USBM simulator, Section 2) and the Wagner MTT F20 18S Teletram diesel car were supplied by C.B. Manula of Penn State University. Specifications for the Jeffrey 404L and 404H battery "Ramcars," the Jeffrey 410H diesel "Ramcar," and the Jeffrey Steam-Car were provided by Jeffrey Mining Machinery Division, Dresser Industries, Inc. At this time either the basic chassis of the Jeffrey Steam-Car or the Jeffrey 404L appear to be quite suitable for the installation of a 4.5 to 6 kW-hr flywheel. The volume currently occupied by the steam equipment (or battery in the 404L) is approximately 6' W x 6' D x 2.5' H. The width and depth dimensions are more than required to accommodate a 43-inch diameter flywheel plus its containment. It is estimated that the inductor motor/generator portion of the flywheel equipment will require an additional 12 inches in height above the basic car body in the form of a cylindrical structure approximately 36 inches in diameter.

The specifications shown on the characteristics sheet, Table LVII, for the flywheel powered car are based on the following assumptions:

- Use of a basic Jeffrey chassis either 404L or steam-car
- Overall flywheel assembly height of 36"
Basic chassis weight	19,800	lbs
Flywheel	1,000	(4.5 kW-hrs usable energy)
Containment	400	
Inductor Motor/Generator for 30 sec charge	1,500	
Total Empty Car Weight	22,700	lbs
- Use of the same 50 horsepower electric motor - power shift transmission - torque converter as the Jeffrey 404L Battery Car which determines acceleration, speed and drive efficiency.
Basic car chassis price		$80,000
Flywheel and Containment	(4.5 kW-hrs, Fig. A.1-2)	8,000
Inductor Motor Generator	(@$6.50/lb, Note 1)	3,000
Miscellaneous (Note 2)		9,000
Total Flywheel Car Costs		$100,000

Table LVII

Shuttle Car Characteristics

Model	Joy 18SC13DC "Base Case"	Jeffrey 404L Battery	Jeffrey 404H Battery	Jeffrey 410H Diesel
Capacity, ft^3 rated	190	134	208	355
Capacity, lbs rated	9,500	13,450	10,400	22,250
Capacity, ft^3 max. in 6 ft seam	236	269	291	445
Overall length, inches	330	326.5	352	420
Overall width, inches	113	128	120	120
Overall tram height, inches	32	35	42	56
Ground clearance, inches	6.5	6.125	6.0	10
Empty weight, lbs	26,700	27,000	31,500	48,000
Discharge time, mins	0.33	0.25	0.25	0.33
Acceleration, max., ft/min^2	20,000	40,000	40,000	35,000
Deceleration, max., ft/min^2	10,000	40,000	40,000	35,000
No. Drive wheels	2/6	2/4	2/4	2/6
No. Brake wheels	2/6	2/4	2/4	2/6
Speed, max. empty, ft/min	458	440	352	792
Speed, max. full, ft/min	422	396	308	750
Cornering speed, ft/min	250	440	352	792
Speed, reverse, max. ft/min	458	440	352	792
Motor horse power, total	2x15+15	50	2x25+25	146
Gear reduction	33.3	131.3,50, 29	46.5	118.2,62.7, 36
Wheel radius, in	16.2	16.2	20	27.3
Drive efficiency, %	70	70	80	70
Rolling resistance, lbs/ton	100	100	100	100
Electric motor, voltage	250	128	250	-
Electric motor current	200	600	600	-
Battery capacity kW hrs	-	68.5/105	68.5/105	-
Battery weight	-	6,600/9,000	6,600/9,000	-
Battery life, no. cycles	-	1,200	1,200	-
Fuel consumption, gals/hr	-	-	-	2.44
Engine life, mean time between overhauls, hrs	-	-	-	1 year @ 3 shifts
Car price, $'s	80,000	57,000		111,240
Battery price/set $'s	-	10,000/ 12,000	10,000/ 12,000	-
Spare engine price $'s				28,000
Unique spares, $'s				
Complete overhaul, $'s				6-10,000

Table LVII (continued)

Shuttle Car Characteristics

Model	Wagner F20 18S Diesel	Jeffrey Steam Car	Flywheel Car Tr.Tr.
Capacity, ft^3 rated (struck)	454	151	151
Capacity, lbs rated (struck)	22,700	7,550	7,550
Capacity, ft^3 max. in 6 ft seam	454	220	220
Overall length, inches	404	408	408
Overall width, inches	127	124	124
Overall tram height, inches	68	30	42
Ground clearance, inches	9	6-8	6-8
Empty weight, lbs	38,700	26,000	22,700
Discharge time, mins	0.5	0.25	0.25
Acceleration, max., ft/min^2		62,000	40,000
Deceleration, max., ft/min^2		104,000	40,000
No. Drive wheels/Total	2/4	4/4	4/4
No. Brake wheels/Total	4/4	4/4	4/4
Speed, max. empty, ft/min	1,022	528	440
Speed, max. full, ft/min	170	528	396
Cornering speed, ft/min		528	440
Speed, reverse, max. ft/min	1,022	528	440
Motor horsepower, total	146	75	50
Gear reduction		84.8, 47.1, 24.8,	131.3, 50, 29
Wheel radius, in	21	16.2	16.2
Drive efficiency, %	70	70	70
Rolling Resistance, lbs/ton	100	100	100
Electric Motor, voltage			
Electric Motor current			
Battery capacity kW hrs			
Battery weight			
Battery life, no. cycles			
Fuel Consumption, gals/hr	2.44	3.4	
Engine life, mean time between overhauls, hrs	600	NA	NA
Car price, $'s		130,000	100,000
Battery price/Set $'s			
Spare engine price $'s		40,000	
Unique spares, $'s		NA	15,000

Notes: 1. The cost of the flywheel inductor motor/generator at $6.50 per pound is based on General Electric production experience with equivalent machinery.

2. Miscellaneous includes items such as a special high-voltage dc traction motor to match the output voltage of the flywheel inductor machine, a bridge rectifier to convert inductor machine ac output to dc, the possibility of a vacuum pump to maintain low pressure in the flywheel chamber and bearings; flywheel mountings and adaptations to the basic car to accommodate the flywheel equipment.

- The cost of a spare flywheel-inductor machine package, shown under Unique Spares, was estimated by adding the costs of the flywheel, the inductor motor/generator, and about half of the miscellaneous costs. (Figure 87)

Most cars are rated in terms of struck or water level capacity. However, in a high seam (6 ft) mine they are frequently fitted with side boards and piled as full as possible. A version of the Jeffrey 404 Battery Car has a struck capacity of 208 cubic feet compared with 190 cubic feet for the "base case" Joy Car. The Joy Car is used in the Simulator with a load capacity of 236 cubic feet. In a 6-foot seam the Jeffrey cars can easily hold 236 cubic feet (Figure 88).

A.2 INVERTER COSTS

The load commutated inverter forms the heart of the wayside spin-up equipment (Section 4.7). It is provided with dc from the mine power center and converts this power to variable frequency ac required by the inductor machine to spin-up the flywheel. The estimated costs shown in Figure 89 are based on other General Electric design studies and assume a 600 volt dc input. Such equipment is normally rated at constant input power. However, due to the short time transient nature of the spin-up requirements, the continuous duty rating of the inverter may be exceeded in accordance with the transient capability curve of Figure 90. Using these two plots, the estimated cost of the inverter may be determined for any spin-up time in the range of consideration.

A.3 POWER CENTER COSTS

The power required to spin-up the flywheel may place a heavy peak load on the mine supply, possibly to the point where a larger capacity power center must be used. Accordingly, it seems only reasonable to charge off some part of the cost of the power center against the cost of electrically propelled cars. The dc powered

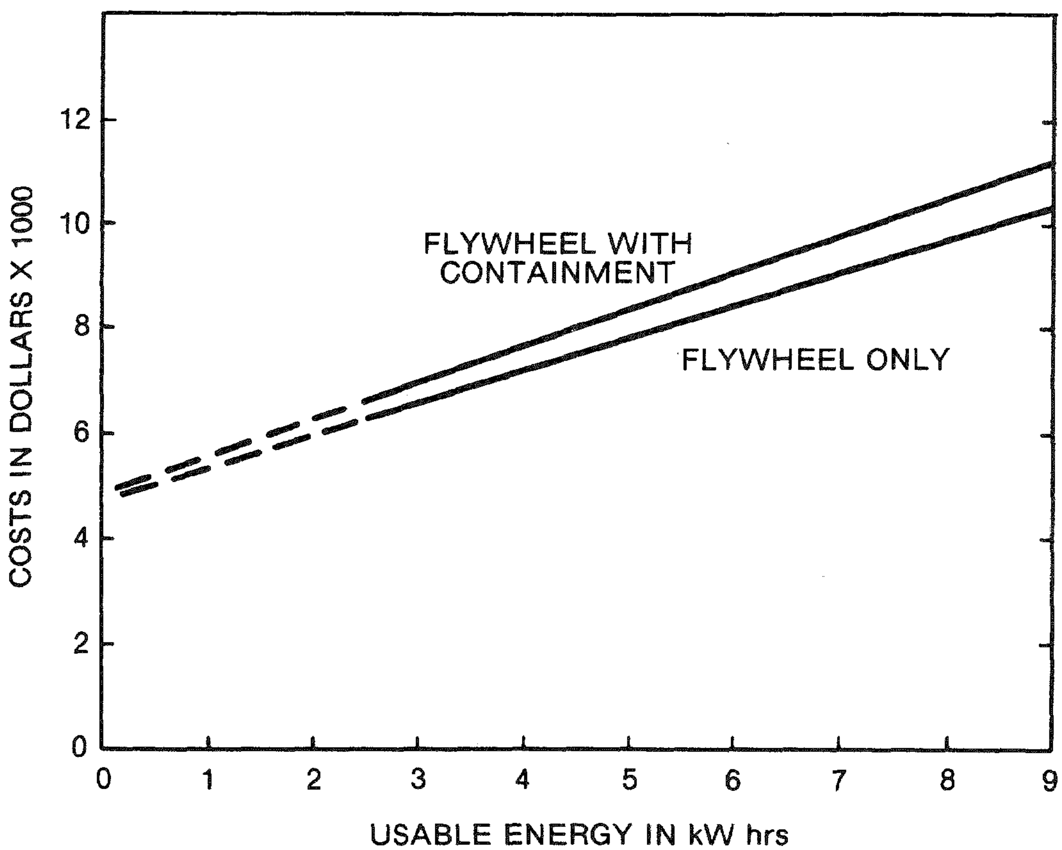
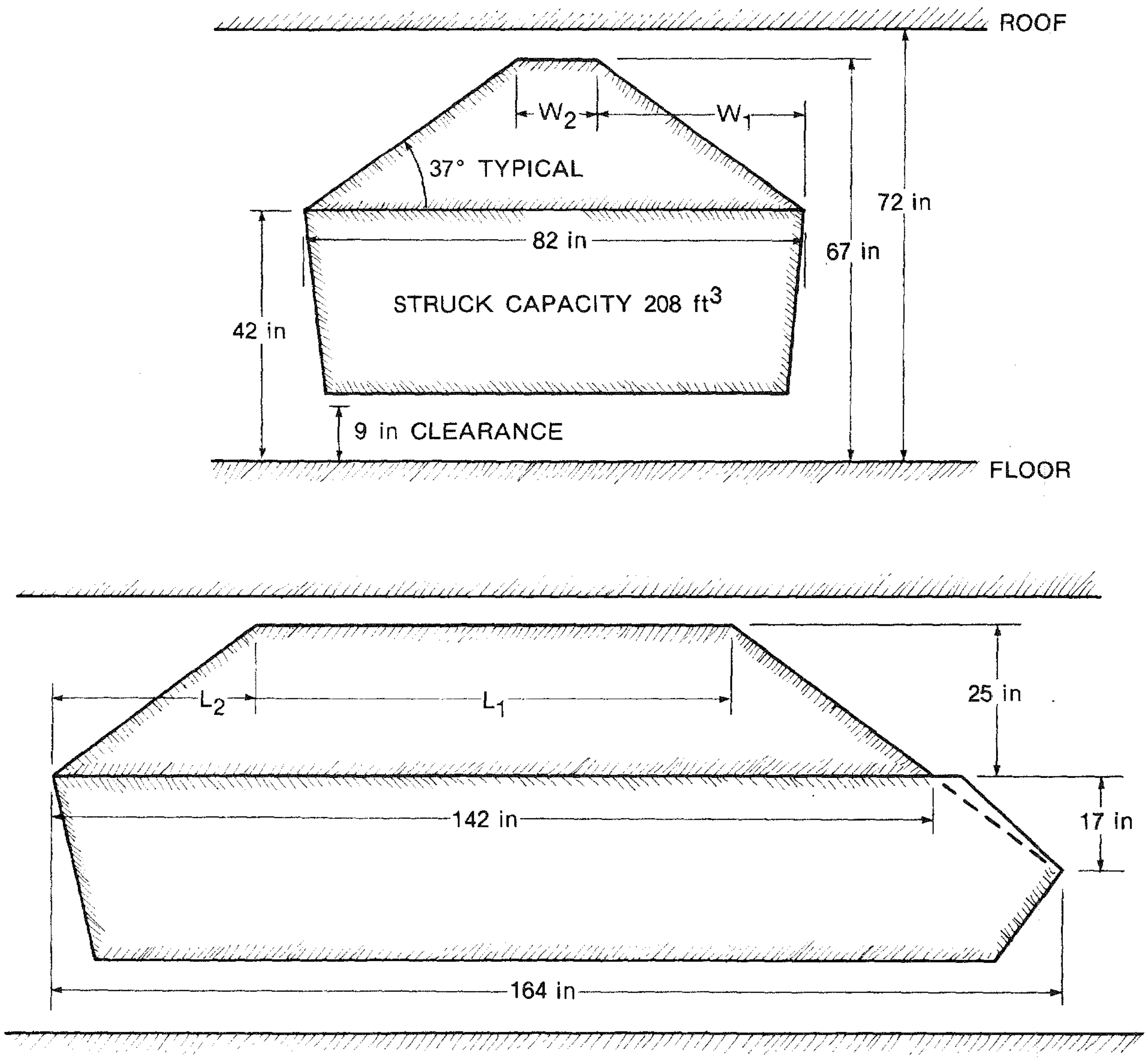


Figure 87. Flywheel and Containment Cost



Pile Capacity $= HW_2L_1 + 2 \times \frac{1}{2}\, HW_1L_1 + 2 \times \frac{1}{2} \times HL_2W_2 + 4 \times \frac{1}{3}\, HW_1L_2$

Where: $H = 25$ in (5 in roof clearance assumed)
$W_1 = L_2 = 25$ in cot. $37° = 33$ in
$L_1 = 142$ in $- 2L_2 = 76$ in
$W_2 = 82$ in $- 2W_1 = 16$ in

Pile Capacity $= 30{,}400 + 62{,}700 + 13{,}200 + 36{,}300 = 142{,}600$ in^3 $= 83$ ft^3

Figure 88. Jeffrey Model 404 Battery Ramcar Capacity Calculations

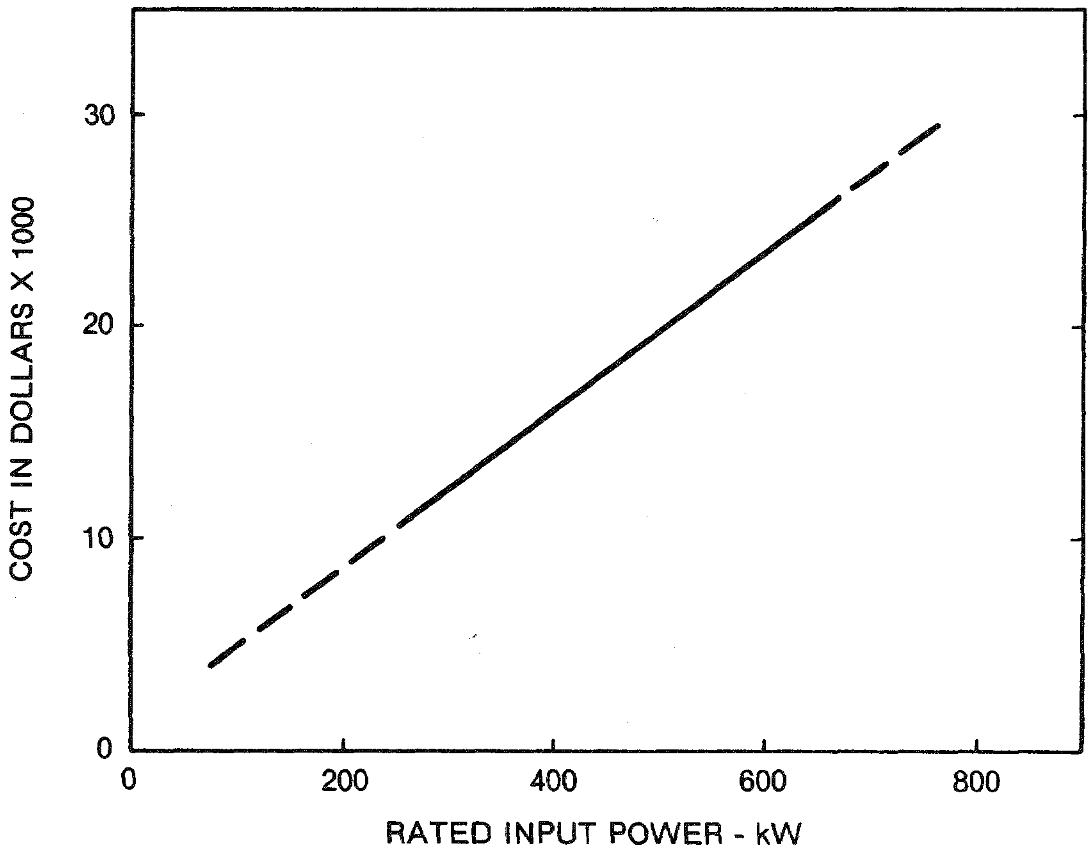


Figure 89. Estimated Inverter Costs

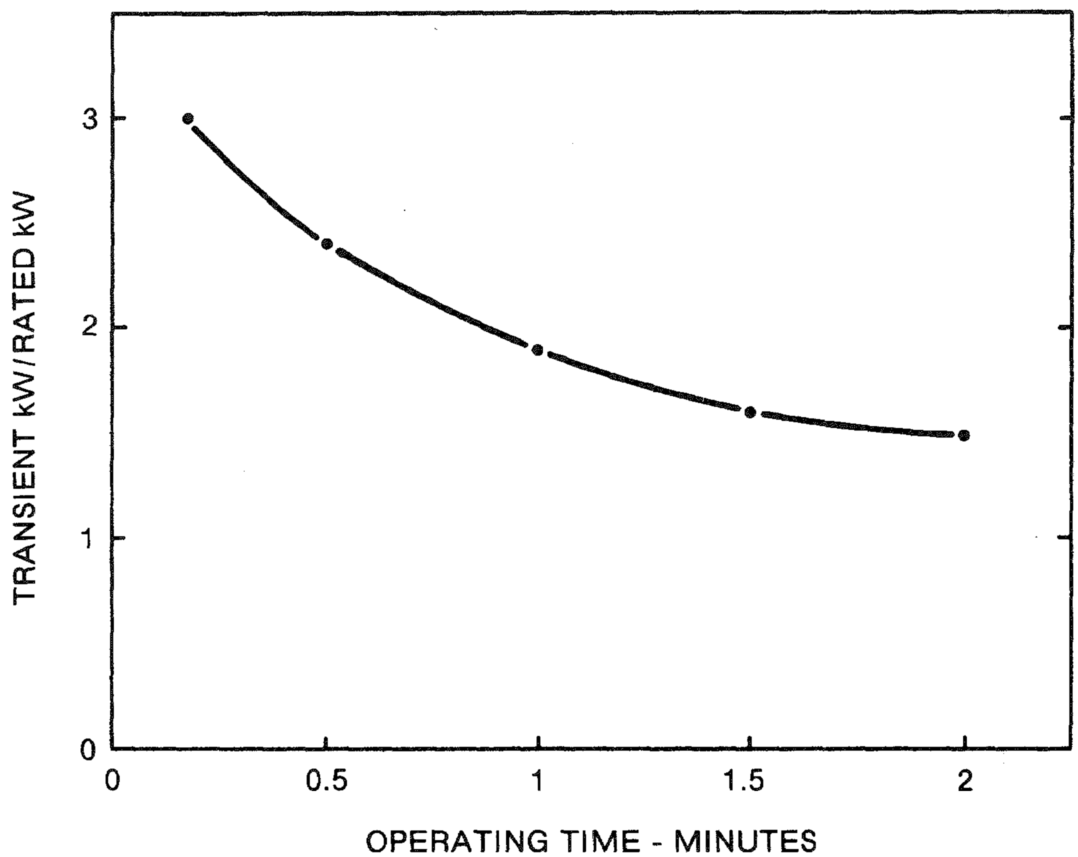


Figure 90. Transient Capability of Inverter

tethered shuttle cars used in the PSU/USBM Simulator have a rated power input of 60 kW each. The "base case" model (6 entry cut plan, 6-foot seam) run was examined to determine power requirements. The simulator showed the following averages per car per shift: 78 kW-hrs, 30 trips and 2.55 mins tram time per trip whereby

$$\frac{\text{kW-hrs x 60 mins/hr}}{\text{Trips/Shift x Tram Time}} = \text{kW load}$$

substituting: $$\frac{78 \times 60}{31 \times 2.55} = 59 \text{ kW running load per car}$$

The cost of a "typical" 750 KVA, ac/dc mine power center is estimated at $55,000 (Figure 91; Reference 1). Since this center provides power to other mining equipment its cost should be prorated in proportion to the load:

$$\frac{\text{Cost}}{\text{Expected Life}} \times \frac{\text{S.C. load}}{\text{Capacity}} = \frac{\$55{,}000}{20 \text{ yrs}} \times \frac{60 \text{ kW}}{750 \text{ kVA}} = \underline{\underline{\$220/\text{yr}}}$$

A battery powered shuttle car will also require the same power delivered to the traction motor(s). In this case we must consider the input/output or storage efficiency of the battery and the efficiency of the battery charger. Industrial-rated storage batteries operate with a storage efficiency of about 80% when new. This falls off with age to approximately 70% at end-of-life. Accordingly, a storage efficiency of 75% is used, with a battery charger efficiency of 90% (Reference 2). On the favorable side, the tethered shuttle car draws load only while tramming but we can assume that a battery set is on essentially constant charge for say 6 hours of an 8-hour shift.

$$\frac{\text{kW hrs/Shift}}{\text{Charge Time}} \times \frac{1}{\text{Storage Efficiency x Charge Efficiency}} = \text{kW Load}$$

$$\frac{78}{6 \text{ hrs}} \times \frac{1}{0.75 \times 0.9} = 19 \text{ kW Charging Load}$$

and: $$\frac{\text{Cost}}{\text{Expected Life}} \times \frac{\text{Charging Load}}{\text{Capacity}} = \frac{\$55{,}000}{20 \text{ yrs}} \times \frac{19 \text{ kW}}{750 \text{ kW}} = \$70/\text{yr}$$

(Also see Section 5.5-5 for a further discussion of Power Center)

A.4 FUEL STORAGE & HANDLING EQUIPMENT COSTS

For the diesel and steam powered cars, the cost of fuel handling and storage equipment, including the cost of any special fire prevention, control and safety equipment, is estimated at $10,000 -- with a useful life of 10 years, or an annual cost of $500 -- for each of 2 cars in a mine section.

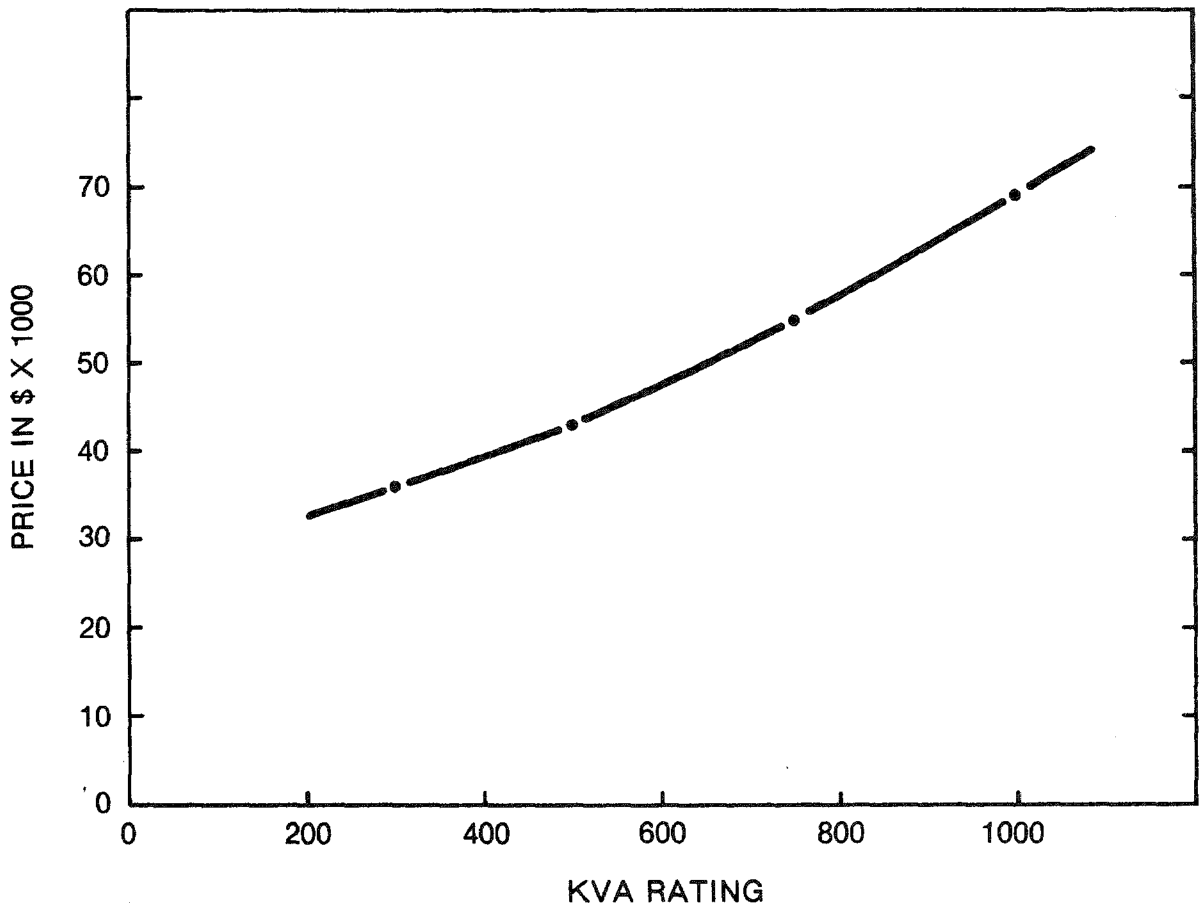


Figure 91. Mine Power Center Price
ac Input - dc Output

A.5 BASIC CAR MAINTENANCE COSTS

Appendix B includes capital cost details from C.B. Manula. Other Repair and Supply Items were reported to cost $18.40 per shift or $0.056 per ton. Based on 2 shifts per day, 328 tons per shift, 220 working days per year, and 2 shuttle cars per section, this works out to approximately $4030.00 per car per year. Shuttle car supplies include tires, motors, electrical equipment (other than cables), drive train components, brakes, lubricants, hydraulic parts, etc. For the purposes of this study, it was assumed that the cost of these basic supplies would be essentially the same for all car types.

A.6 CABLE MAINTENANCE AND REPAIR COSTS

C.B. Manula of Penn State University has obtained data on the costs of shuttle car cable repair, replacement, and down time. These data were taken from a nine months study of a central Pennsylvania underground coal mine. This particular mine had a seam height of from 42 to 48 inches. The principal implications of the low seam reflects in fewer tons of coal produced per shift and a lower load carrying capacity of the shuttle car. It seems reasonable to assume that cable failures are directly related to the number of trips. The significant information from the study is shown in Table LVIII below.

Table LVIII

Shuttle Car Cable Costs

	PER SHIFT	PER CAR PER SHIFT	PER CAR PER YEAR
Average Clean Tons	125	62.5*	31.250
Cable Splices	0.43	0.22	109.0
Splice Costs (Labor & Mat'l)	$17.30	$8.65	$4325.00
Cable Change Outs	0.0125	0.00625	
Change Costs (Labor & Mat'l) 50% New	-	-	1344.00
Change Costs (Labor & Mat'l) 50% Reconditioned			406.00
Total Cable Costs Per Car Per Year			$6075.00

*Note: At 3 clean tons per trip, this implies 21 trips per car per shift compared with an average of 31 trips in the Simulator 6 ft Seam "base case."

C.B. Manula's analysis of the study data also divulges an average of 5.8 minutes of production delay per car per shift attributable to cable related problems (48.3 hrs per year). Total shuttle car production delays amounted to 18.38 minutes per car per shift (153 hours per year).

A.7 BATTERY MAINTENANCE AND REPLACEMENT COSTS

Exide Power Systems Division, Philadelphia, Pennsylvania, was contacted with regard to battery costs and life expectancy in underground mine service. Two battery sets are offered for the Jeffrey 404L Battery Powered Car; the significant characteristics of these batteries are tabulated below.

	Volts	hrs	kWh	Wt-lbs	Cost $'s
Standard (32" coal)	128	550	68.5	6,100	10,000
Heavy Duty (38" coal)	128	850	105.4	8,500	12,000

The overall energy storage efficiency of the battery (when new) is quoted at kW-hrs output/kW-hrs input = 83%. From other sources (Reference 3) it is known that this efficiency gradually deteriorates over the life of the battery to something like 70% at the end of life. For the purposes of this study an average energy storage efficiency of 75% over the life of the batteries is assumed.

Exide reports that the life of battery sets varies greatly depending on the quality of the maintenance they receive, e.g., maintaining electrolyte level, clean terminals, etc. However, with a good maintenance program batteries survive 1,000 to 1,500 discharge cycles over a 4-year period in mine service conditions, each battery for one shift. (It is interesting to note that this type of battery simply kept on trickle charge and well maintained also has a life expectancy of 4 years.) Taking a 4-year life with 2 sets of batteries per car (one on charge while the other is in use) it will be necessary to replace a set of batteries every two years, or the annual cost of battery replacement will be $12,000 ÷ 2 = $6,000. Assuming 2 shifts per day, 2 sets of batteries, and 250 days of mining per year, 4 years of life would represent about 1,000 discharge cycles. The $12,000 cost of the larger "heavy duty" battery with its 105 kW-hr rating was used for this calculation since the 68.5 kW-hr rating of the "standard" battery does not equal the average tramming energy of 78 kW-hrs derived from the base case simulator run. Battery maintenance labor costs are neglected on the assumption that they represent negligible incremental work.

The cost of a magnetic amplifier controlled battery charger is approximately $3,000. This cost along with the cost of two initial sets of batteries is included in the cost of the basic car.

A.8 ENGINE MAINTENANCE AND REPLACEMENT COSTS

Jeffrey Mining Machinery Division, Dresser Industries, Inc. was contacted regarding its experience with maintenance, overhaul, and replacement of diesel engines in the 410 Series Ramcars. They recommend a brief standard maintenance routine, carried out at the start of each shift, which consists of inspecting and cleaning the exhaust scrubber, flame arrestor, air intake cleaner, etc. With this routine maintenance schedule Jeffrey reports about 5,000 hours, or 1 year of 3 shift operation between major engine overhauls. Jeffrey sells a complete spare engine power package for the 410H Ramcar, which may be changed out in one shift for approximately \$28,000. Jeffrey will perform a complete overhaul of the engine for approximately \$10,000.

C.B. Manula studied the experience of the Martin County Coal Company's 1C Mine operating in a 7' seam. Using a Wagner Teletram equipped with a Caterpillar DC-333C (NA) diesel engine, they have experienced 600 hours engine life. This works out to 2.6 major engine overhauls per year, or something like 8 times the frequency which Jeffrey reports. A literature search divulged the information that London, England transit buses average 15 months or 15,000 miles between major engine overhauls (Reference 4). But of course their duty is not restricted by the requirements of Schedule 31. Another source (Reference 5) reports 500 hours mean time between failures and a mean time to repair of 2.3 hours for large stationary diesel engines. Of course, an MTTR of 2.3 hours does not represent a complete engine overhaul, nor is the type of service comparable.

Assuming that Jeffrey's report may be a little optimistic and that Martin County's conditions may represent rather severe service, for the purposes of this study we will adopt a mean time between engine overhauls of 1,000 hours and an average cost of \$8,000 per overhaul. This works out to:

$$\frac{3.2 \text{ hrs/shift} \times 500 \text{ shifts/yr}}{1{,}000 \text{ hrs/overhaul}} \times \$8{,}000\text{/overhaul} = \underline{\$12{,}800\text{/yr}}$$

At this time no figures are available on engine maintenance costs for the Jeffrey steam-car since the car has just been built and there is no operating experience. However, to be cost competitive with tethered and battery powered cars, steam-car engine maintenance should be in the \$1,000 to \$2,000 per year bracket. For the purposes of this study we will assume a figure of \$1,500 annually for engine and boiler maintenance.

A.9 FLYWHEEL & WAYSIDE POWER MAINTENANCE COSTS

A design life of 10^6 cycles seems to be desirable. A flywheel cycle is defined as the mission profile energy extraction of a fully charged flywheel to the lowest usable energy level, nominally half speed or 25% of full charge. At an average of 31 trips per shift and 500 shifts per year, 10^6 cycles represents a design life of 64 years for fatigue of the flywheel alone. However, the failure rate of bearings, the inductor machine, and any auxiliaries are not included. It should also be noted that cyclic operation of flywheels at the desired stress levels is not an area where very much experience is available. In short, there is little precedent in terms of establishing an MTBF or maintenance prediction. Since the flywheel-motor-alternator is a relatively simple piece of rotating machinery operating in a protected environment, it is anticipated that it will have a high MTBF (Mean Time Between Failure) of the order of 5,000 hours. For the purposes of estimating, one complete disassembly-inspection-preventative maintenance task of 40 hours at $10.00 per hour is scheduled once a year.

An unknown source of potential maintenance costs is the connector through which spin-up energy will be transmitted from the wayside station to the car. At this point it does not seem unreasonable to allocate 1 hour per week per car, or $500 per year labor, plus $500 per year for parts to connector maintenance. Added to the $400 per year for routine flywheel preventative maintenance gives a total of $1,400.

The wayside spin-up equipment (inverter) will be all solid-state high-reliability design. That does not imply that it will be completely maintenance free; however, it is not anticipated that the maintenance costs of the inverter will amount to any significant cost.

A.10 ENERGY COSTS

Electrical energy costs, based on a steady load of 5,000 kW, range from 2 to 3.5 cents per kW-hr (Reference 6). For the purposes of this study, we will adopt a commonly used figure of 3 cents per kW-hr.

Conventional Shuttle Car

As previously noted, the "base case" simulated operation shows an average power consumption of 78 kW-hrs per shift per tethered shuttle car. Assuming 2-shift operation, 5 days a week and 50 weeks a year:

$$78 \text{ kW-hrs} \times 500 \text{ shifts} \times \$.03 = \underline{\underline{\$1,170}}$$

This figure represents the cost of electrical energy to operate one tethered shuttle car for one year.

Battery Powered Car

For a battery powered car, we must take into account the efficiencies, or more properly, the losses of the battery and the battery charger.

$$\frac{78 \text{ kW-hrs} \times 500 \text{ Shifts} \times \$.03}{.75 \text{ Storage Efficiency} \times .9 \text{ Charger Efficiency}} = \underline{\underline{\$1,733}}$$

which gives us the energy cost for one battery car for one year.

Flywheel-Powered Car

In like manner, the cost of energy for a flywheel powered car for one year will be as follows with the efficiency assumptions below.

$$\frac{78 \text{ kW-hrs} \times 500 \text{ Shifts} \times \$.03}{\eta 1 \times \eta 2 \times \eta 3} = \underline{\$1,536}$$

where: $\eta 1$ = Generating Efficiency = 92.1%
$\eta 2$ = Spin-Up Efficiency = 91.9%
$\eta 3$ = Inverter Efficiency = 90.0%

Diesel-Powered Car

For diesel powered cars, C.B. Manula reports an average fuel consumption of 2.44 gallons per hour and a mean working time of 3.2 hours per shift (Figures 92 and 93).

2.44 gals/hr x 3.2 hrs/shift - 7.8 gallons/shift

7.8 gals/shift x 500 shifts/yr x \$.36/gal = <u>\$1,404/yr</u>

The Wagner Teletram MTT-F20-18S diesel ram car is somewhat larger and heavier than the Joy 18SC13DC shuttle car used in the simulator; 38,750 lbs net vs. 26,700 lbs net weight. Straight energy calculations based on the Joy Car's energy requirements in the simulator "base case" show:

78 kW-hrs x 1.341 hp/kW = 105 hp hrs per shift (tramming)

105 hp hrs x 0.4 lbs/hp hr/7 lbs/gal = 6 gals/shift

Obviously there is a disparity since the Teletram weighs 6 tons more, and no allowance has been made for load factor (fuel consumed while the diesel is idling). The most logical explanation is that the actual coefficient of rolling resistance experienced in the Martin Coal Company's Mine is considerably less than the 0.15 figure used in the PSU/USBM Simulator.

Steam-Powered Car

Jeffrey Manufacturing Company has reported a fuel consumption rate of 3.4 gallons per hour at continuous duty for the steam powered car. Compared with the Distribution of Fuel Consumption for the Wagner MTT-F20-18S Teletram shown in Figure 93, the 3.4 gallons per hour continuous duty fuel consumption compares reasonably with the 2.44 gallons per hour mean fuel consumption shown. One would not expect as high a fuel efficiency from an external combustion Rankine cycle engine as from the internal combustion Otto cycle diesel engine. Assuming the

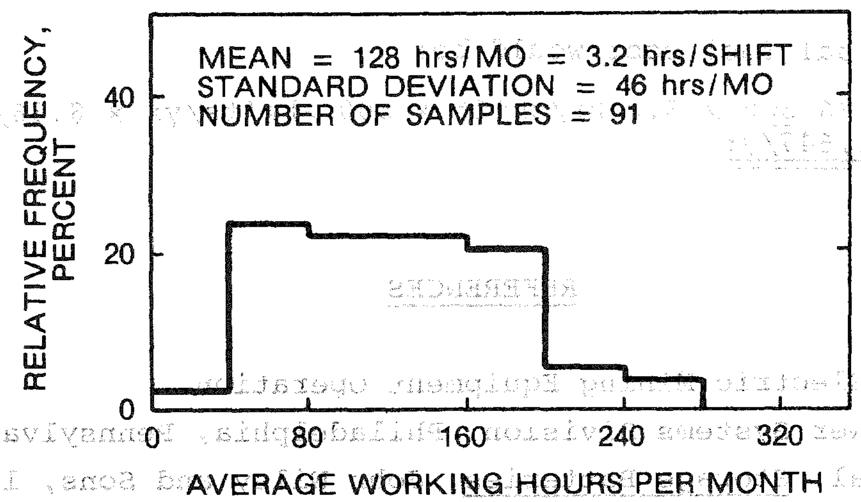


Figure 92. Distribution of Working Hours per Month for MIT-F20-18S Teletrams (NTIS Publication, PB 246 299/AS)

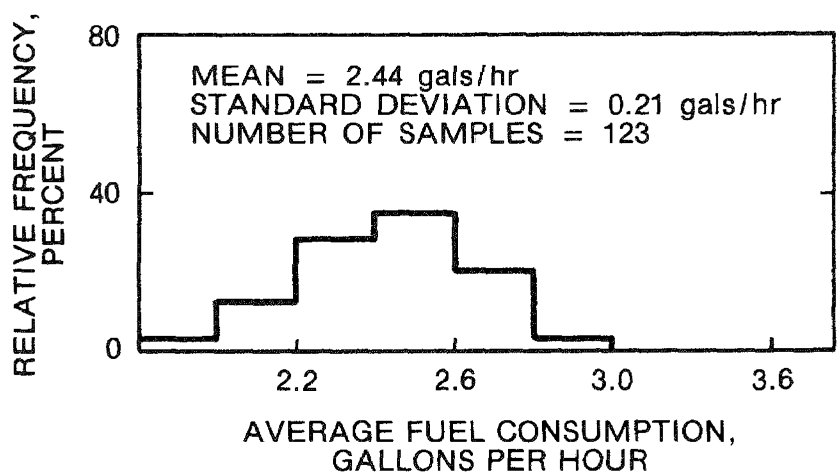


Figure 93. Distribution of Consumption of Fuel by MIT-F20-18S Teletrams*

3.4 gallons per hour continuous duty rate compares with the 2.9 maximum rate shown for the Teletram, the mean rate for the steam-car would be:

$$\frac{2.44}{2.9} \times 3.4 = 2.86 \text{ gallons per hour}$$

*NTIS Publication, PB 246 299/AS

and the annual fuel cost would be:

2.86 gph x 3.2 hr/shift x 500 shifts/yr x \$.36/gal = \$1,647/yr

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6. Internal General Electric Sources.

Appendix B

CAPITAL COST DETAILS AND AVERAGE COST CALCULATIONS

The following capital cost details and computer simulation results were reported by C.B. Manula as the result of his work on the contract.

B.1 COST ANALYSIS

The simulation results for the base case are summarized in Table LIX. Here, an average production of 327.1 tons per shift and 48.6 minutes of miner wait on shuttle car were obtained for a cable related delay of one minute at a 40 percent frequency of occurrence per trip. Table LX lists the estimated costs for this production level using an average depreciated life of 12 years for all items. Tables LXI, LXII, and LXIII provide the capital cost summary, capital cost detail, and average cost calculations for labor and supervision, respectively. The supply costs include rebuilding equipment and mine extension capital.

Estimate of Shuttle Car Costs

Power cost is estimated based on a duty cycle of 243 seconds, i.e., 30 seconds to discharge, 60 seconds loading, and 153 seconds traveling. Hence, for (2) 11,800 lb (5.9 tons) shuttle cars with 47.5 connected horsepower moving 327.1 tons per shift (Table LIX), 55 trips per shift or 3.7125 hours of shuttle car operation are required. For a load factor of 60 percent the power costs are obtained as follows:

Power Cost = 0.6 LF x 35.44 kW x 3.7125 hrs x $0.03/kWhr = $2.37/shift

= $2.37/327.1 TPUS = $0.0072/ton

Supply costs include cable repair and cable replacement costs and other supply and repair items listed below:

1. Cable repair - $12.30/2.3 shifts = $5.35/shift = $0.016/ton
2. Cable replacement - $500/80 shifts = $6.25/shift = $0.019/ton
3. Other repair and supply items - $18.40/shift = $0.056/ton

Total shuttle car costs = $0.091

Table LIX

Simulation Summary (Base Plan)

Shift	1	2	3	4	5	6
Production	361.9	327.2	361.9	326.9	323.1	352.5
Wait on Shuttle Car	40.49	47.64	35.69	36.14	53.94	41.34

Shift	7	8	9	10	11	12
Production	328.7	300.6	309.4	338.7	293.7	345.6
Wait on Shuttle Car	49.99	46.54	44.99	57.69	52.79	57.24

Shift	13	14	15	16	17	18	19*	Average	
Production	339.4	330.0	329.6	327.9	290.0	308.1	199.4	327.1	Tons
Wait on Shuttle Car	42.09	45.79	51.39	57.79	51.09	61.89	39.29	48.6	Minutes

*Shift 19 worked 316.5 minutes

B.2 SIMULATION

A delay of 0.25 minute was applied before loading and dumping to simulate ram car change-out times. These results are compared with the base plan output. Here, the 2 untethered car plans are less productive than the base plan and, hence, more costly because of the added capital investment. This is the result of the added delay for car change-out times, i.e., 0.25 minute before loading and dumping.

Table LX

Estimated Costs for the Base Case

Simulated tons/shift	327
Manning	21/unit
Cost/man-day	$90.00
Tons/man-day	15.57

Item	Cost Per Ton
Labor	$ 5.780
Supplies	3.997
Power	0.482
Tonnage Part of Health and Welfare	0.820
Compensation and Black Lung	1.156
Administration	0.750
Insurance and Local Taxes	0.376
Depreciation	3.062
Royalty/Depletion Allowance	0.50
Estimated Cost per Ton	$16.923

Table LXI

Capital Cost Summary ($ in 1000s)

Section-Related (Per Section)	
Face	677
Haulage	319
Electrical	60
Other	28
General Haulage	162
General Electric	21
	$1,267 Per Section
General Inside	
Mobile	656
Other	215
	$ 871
Shaft and Slope	305 + (8.0 x VF)
Surface	10,695
Mobile Surface Equipment	415
Initial	800
Development Cost	5,000

Σ = [18,086 + (1,267)(No. Sections) + (8.0)(V.Ft.)] x 110% for Contingencies

Table LXII

Capital Cost Detail ($ in 1000s)

Section Equipment		
Continuous Miner	300	
Shuttle Cars (2)*	160	
Roof Bolters (2)	45	
Feeder Breaker	50	
Scoop w/Battery	55	
Rock Duster	5	
Spare Parts @ 10%	62	677
Section Haulage		
Belt Complete (3000' x 36")	150	
Rail - 60# x 3000'	52	
Parts Car	10	
Trolly and Feeder Wire	18	
Supply Cars (3)	15	
Portal Bus	20	
Supply Loco	25	
Spare Parts @ 10%	29	319
Section - Other		
Face Pumps	5	
3000' Pipe	10	
Auxiliary Fan	10	
Welder and Tools	3	28
Section - Electrical		
Power Center	25	
High-Voltage Cable (3000 ft)	18	
Belt Transfer	10	
Trailing Cables	10	60
	Total/Section	$1,063

*Add: $220 for two flywheel cars plus wayside inverter
-150 for two standard cars
$ 70 added
$70,000/(10 yrs x 440 shifts/yr) = $15.91/shift extra

For third car
($70,000 + 100,000)/(10 x 440) = $38.64/shift extra

Table LXII (continued)

Capital Cost Detail ($ in 1000s)

General Inside - Other

Shop Tools	40	
Belt Splicing Kit	30	
Fire Cars (4)	20	
Gas, Dust, Noise Detectors	25	
First Aid and Mine Rescue	25	
Miscellaneous and Training	25	
Mine and Trolley Phones and Wire	50	215

Bottom Area and Shaft, Slope

Site Preparation for Shafts	100	
Slope Bottom Preparation	50	
Shaft Bottom Preparation	100	
Slope ($3,000/Vft)		
Shaft ($25,000/Vft)		
Vent Shaft (1,500/Vft)		
Emergency Hoist	15	
Slope Belt Term	40	
Slope Belt ($40/ft) x 3 = $120/Vft		

Shop and Mobile Equipment

Shop Tools	100	
Front-End Loader	80	
Crane	75	
Trucks (2)	80	
Auto and Jeep	15	
Pick-up Trucks	15	
Fencing, Shrubs	50	415

Table LXII (continued)

Capital Cost Detail ($ in 1000s)

Initial		
Drilling	500	
Mapping	200	
Study	100	800
Development Cost	5000	5000
Initial - General Underground (@ 1500'/Section) Per Section		
Haulage		
60# Rail (1500')		
(Trolley and Feeder)	33	
Sectionalizing Switch	7	
Rectifier	25	
Belt Conveyor (1/2 x 3000' x 48")	78	
Gathering Pump	6	
Steel Pipe (6" x 1500')	8	
PVC Pipe (6" x 1500')	5	162
General Inside - Mobile Equipment		
Cutting Machine	120	
Loading Machine	90	
Drill	75	
Track - Roof Bolter	55	
Compressor	28	
Rock Dust Locomotive	20	
Bulk Rock Duster and Tank Car	58	
Supply Cars (10)	50	
Stoper	1	
Service Loco and Oil/ Grease Cars (2)	48	
Personnel Jeeps (10)	80	
Ballast Car	11	
Flat Car	8	
Main Pump	8	
Powder Car	4	656
Top Area		
Temp Power	125	
Main Power Line	500	
Surface Substation (Complete)	100	
Site Preparation	200	
Coal Silos - Clean and Raw	750	
Preparation Plant (350 tph)	7000	
Fresh Water Supply	100	

Table LXII (continued)

Capital Cost Detail ($ in 1000s)

Top Area (Continued)

Rail Extension	250	
Refuse Ponds	100	
Refuse Bins	100	
Refuse Truck	100	
Refuse Site	75	
Refuse Dozer	100	
Supply Locomotive	25	
Storage Yard	50	
Powder Mag	10	
Office and Bathhouse	300	
Warehouse and Shop	300	
Training Site	150	
Roads	100	
Fan and Housing	200	
Office Equipment and Furniture	30	
Temporary Office and Bathhouse	30	10,695

Table LXIII

Calculation of Average Cost per Man Day

	COST/DAY	COST/YEAR	COST/YEAR @ 220 DAYS
Direct Cost	$55.00	$ --	$12,100.00
Overtime at 6%	3.30	--	726.00
Vacation at 12-1/2 Days*	--	688.00	688.00
Sick Pay - 5 Days*	--	275.00	275.00
Holidays - 10 Days*	--	550.00	550.00
Birthday - Triple Pay*	--	110.00	110.00
Jury, Bereavement, etc.*	--	55.00	55.00
Hourly Health and Welfare	12.32	--	2,710.00
Social Sec., Unemployment*	--	1,250.00	1,250.00
Total of * Items		$2,928.00	
Total			$18,464.00

Note: Items with an * are those which must be paid to additional workers hired to compensate for absenteeism. Or at 10% absenteeism the annual cost for 220 work days is $18,464 + ($2928)(10%) = $18,757.

Or, letting n = number of work days, a = percent absenteeism, then annual cost is:

$$C = (55 \times 1.06 + 12.32)\ n + 2928 + 2928\ (a)$$

$$C = (70.62)\ (220) + 2928 + (2928)\ (0.1) = \$18,757$$

$$C = (18,757)/(220) = \$85.26$$

SALARIED = $20,000 Average + 33% Benefits
6,600

$26,600 ÷ 220 = $120.91

AVERAGE COST = (120.91)(0.14) = 16.93

(85.26)(0.86) = 73.32

$90.25 or $90.00

Appendix C

DRIVE SYSTEMS SIZING

C.1 VEHICLE AND LOAD CRITERIA

In order to determine drive system sizing, a typical vehicle was selected. The general vehicle and load criteria which are employed for tractive effort, torque, and rpm values are as follows:

Vehicle weight	20,000 lbs
Trailer weight	6,000 lbs
Load capacity	16,000 lbs
Shuttle speed	5 mph empty
	4.5 mph full load
Rolling resistance	200 lbs per ton (worst case)
	100 lbs per ton (normal)
Coefficient of traction	0.5
Differential gear reduction	7:1
Planetary gear reduction (in wheel hub)	3.1:1
Radius of wheel	14.9 inches

C.2 CALCULATIONS OF HORSEPOWER, TORQUE, TRACTION, AND ACCELERATION

Horsepower

The horsepower requirement at the wheels is established through calculation of the rolling resistance. Accepted values of rolling resistance are:

200 lbs per ton	maximum
150 lbs per ton	moderate
100 lbs per ton	medium

Total vehicle weight = 42,000 lbs or 21 tons; therefore, the range of rolling resistance to be considered is:

200 lbs/ton x 21 tons = 4,200 lbs
150 lbs/ton x 21 tons = 3,150 lbs
100 lbs/ton x 21 tons = 2,100 lbs

A selection of a horsepower is based upon the ability to overcome a designated rolling resistance at an acceptable velocity. To do this a family of curves is plotted of rolling resistance versus miles per hour for selected constant horsepowers using the equation:

$$\text{Rolling Resistance} = \frac{\text{hp} \times 375}{\text{mph}}$$

mph	RR @ 30 hp	RR @ 40 hp
1	11,250	15,000
2	5,625	7,500
3	3,750	5,000
4	2,812	3,750
5	2,250	3,000

See Figure 94 (Rolling Resistance Versus mph)

From these values it appears logical to use 35 hp and a rolling resistance of 3,150 lbs which yields a velocity of approximately 4 mph on level ground. Efficiencies are not included, since the purpose of these calculations is to get a gross approximation of size.

Torque

The torque required to turn the wheel = rolling resistance x wheel radius or 3,150 lbs x 14.9 inches = 46,935 in/lb

From this the torque on the drive shaft is:

$$\frac{46{,}935 \text{ in/lb}}{21.7 \text{ gear reduction}} = 2{,}162 \text{ in/lb @ } 100\% \text{ efficiency}$$

(21.7 gear reduction = 3.1 planetary x 7 differential)

RPM of drive shaft =

45 RPM wheel @ 4 mph x 21.7 = 976.5

Traction

The slip point of traction at the traction wheels is based on a coefficient of traction of 0.5 and the load on the drive wheels:

Total load on drive wheels = 20,000 lbs or 10,000 lbs each wheel

Slip point = 10,000 lbs x 0.5 = 5,000 lbs

Slip point torque = 5,000 lbs x 14.9 inch wheel radius = 74,500 in/lb per wheel

Traction available exceeds the torque provided at 35 hp; therefore, the slip point should not be exceeded (46,935 in/lb versus 74,500 in/lb).

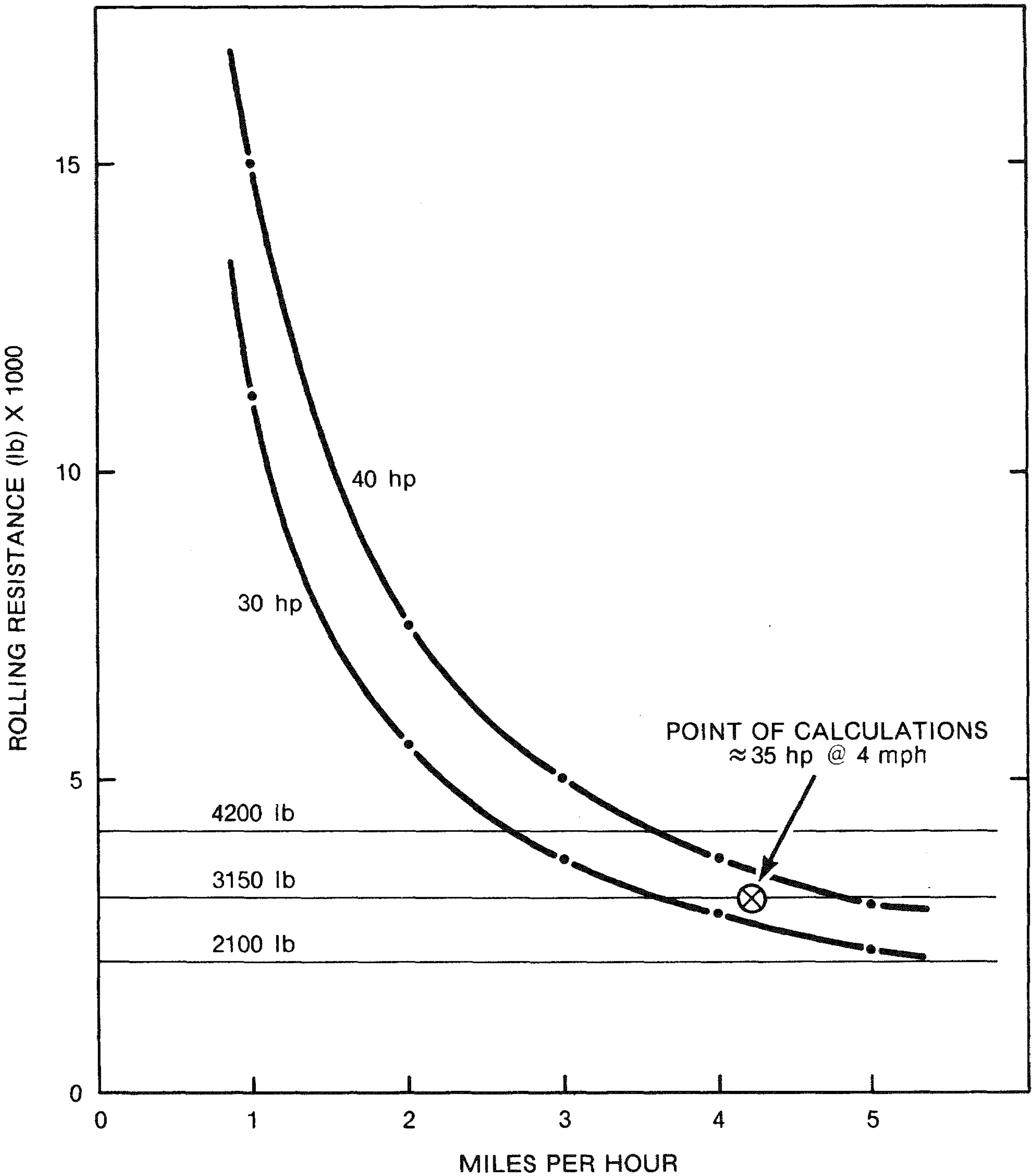


Figure 94. Rolling Resistance Versus Speed at Various hps

Acceleration

The accelerations available at 35 hp can be determined by considering the force required to overcome rolling resistance plus the force of acceleration.

Force available at 35 hp = Rolling resistance$_{(terrain)}$ + ma

$$\frac{hp \times 375}{mph} = \frac{150\ lb}{ton} \times W + \frac{W}{g}\, a \quad \text{where } W = \text{total vehicle weight}$$

$$\frac{hp \times 375}{mph} = W \quad \frac{150\ lb}{ton} + \frac{1}{g}\, a$$

$$\frac{hp \times 375}{mph \times W} - \frac{150\ lb}{ton} = \frac{a}{g}$$

$$\frac{hp \times 375}{mph \times W} - \frac{150\ lb}{ton}\, g = a$$

W = total vehicle weight
= vehicle weight + load

$$\frac{35 \times 375}{mph \times (26{,}000 \text{ is} + \text{load})} - \frac{150\ lb}{2000\ lb}\ 115{,}920\ \frac{ft}{min^2} = a$$

Vehicle weight = 26,000 lbs
Load = 16,000 lbs

Therefore:

ACCELERATION ft/min^2

mph	Unloaded	Loaded
1	49,730	27,531
2	20,518	9,419
3	10,781	3,381
4	5,912	362
5	2,990	0

These values are plotted in Figure 95.

Torque Versus Speed

Drive shaft torque values versus velocity =

MPH	Torque (in/lb)
1	8,992
2	4,496
3	3,079
4	2,162
5	1,800

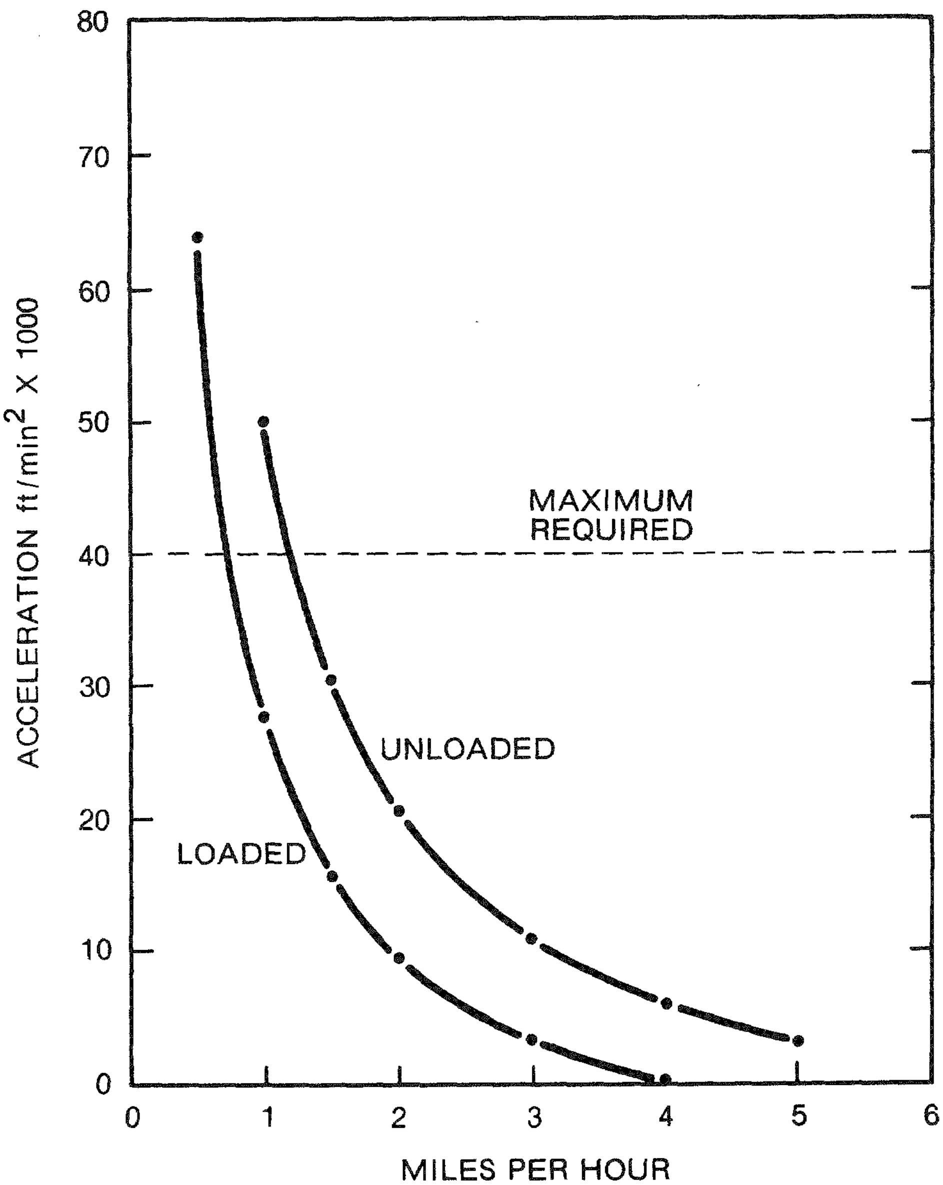


Figure 95. Acceleration Versus Speed at 35 hp

The above acceleration values can be compared to a value of 40,000 ft/min^2 which is considered as an optimum design value for shuttle cars. This indicates that a high value of acceleration is possible at 1 mph and diminishes accordingly as velocity is increased. This appears acceptable, since in a shuttle car acceleration is required at start-up but diminishes at operating velocities.

A reasonable approximation of torque and rpm values is shown in Figure 96. The drive system shown is that of the hydro-mechanical system, but the values are representative for other mechanical systems.

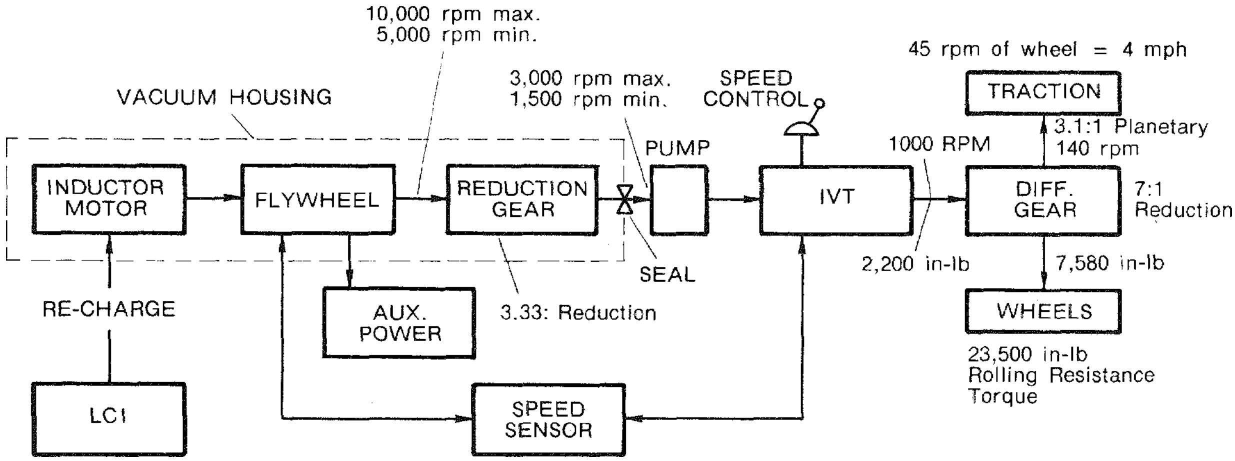


Figure 96. Hydromechanical Drive System

Appendix D

TRACTOR-TRAILER HAULAGE VEHICLE AND FLYWHEEL ENERGY STORAGE SYSTEM

The Jeffrey Mining Machinery Division, Dresser Industries Incorporated, under subcontract to the General Electric Company, investigated suitability of tractor-trailer type face haulage vehicles for a flywheel energy storage system. The scope of work included preparation of a preliminary layout with consideration of:

- Structural modifications
- Equipment layout
- Operator visibility
- Operational considerations
- Functional interface with the flywheel energy storage system
- Impact of a wayside power connection
- Other design considerations

One objective in this study is selection of a suitable vehicle which requires a minimum of design changes for interface with the flywheel system and that has proven capability. Jeffrey has been producing these type vehicles, referred to as RAMCARs®, for many years. The study was then confined to consideration of existing RAMCAR designs so that the task to demonstrate a flywheel powered haulage vehicle would not require developing a new machine.

Findings during the study were reviewed with G. E. personnel during meetings at Jeffrey and through numerous telephone contacts. Results of this study are presented in this appendix as prepared by Jeffrey. Included is a dicussion of:

- Haulage vehicle selection
- Operator visibility and component layout
- Vehicle performance and efficiency
- Integrated cooling system (Flywheel and vehicle powertrain)
- Vehicle electrical system
- Vehicle modification tasks

D.1 SUMMARY AND CONCLUSIONS

Any of the Jeffrey RAMCAR designs could be used as the vehicle for demonstrating a flywheel powered face haulage system. There is adequate space on the tractor portions, normally occupied by a battery, a steam or diesel engine, or a cable reel assembly, where the flywheel assembly and associated components could be mounted. Of these possibilities, the four wheel drive vehicle designed for the steam powered engine appears best suited to accommodate a flywheel power package. Figure 97 is an artist sketch of the flywheel powered RAMCAR.

The original vehicle selection and the preliminary configuration analysis examined the interface requirements, resulting vehicle performance and other relevent factors. Significant in this vehicle recommendation are the four wheel drive and the large payload capability. Also, efficiency in vehicle power requirements must be favorable because of flywheel capacity and recharge considerations. As shown in Section D.3, the preliminary configuration has drive line efficiencies, including the D.C. motor, typically between 50 and 65 percent. Vehicle tram speed capability remains high.

Interface with the flywheel power system should present no major problems. A constant speed D.C. motor and a matching torque converter will be used as input to the mechanical drive line. This motor can accept the 500 to 600 volt output from the flywheel package.

Other changes to the vehicle are minimal and are identified in outline form in Section D.7.

Component placement and operator visibility are also favorable. Height of the flywheel package results in a dome rising above the original top surface of the machine. By keeping the flywheel as close to the tractor bumper as possible the angle of visibility obscuration is relatively small. The mounting for the flywheel must provide for shock deflection but this can be done so that the dome height should not exceed approximately 12" above the original top surface. Photographs of a full scale mock-up are included to show these factors.

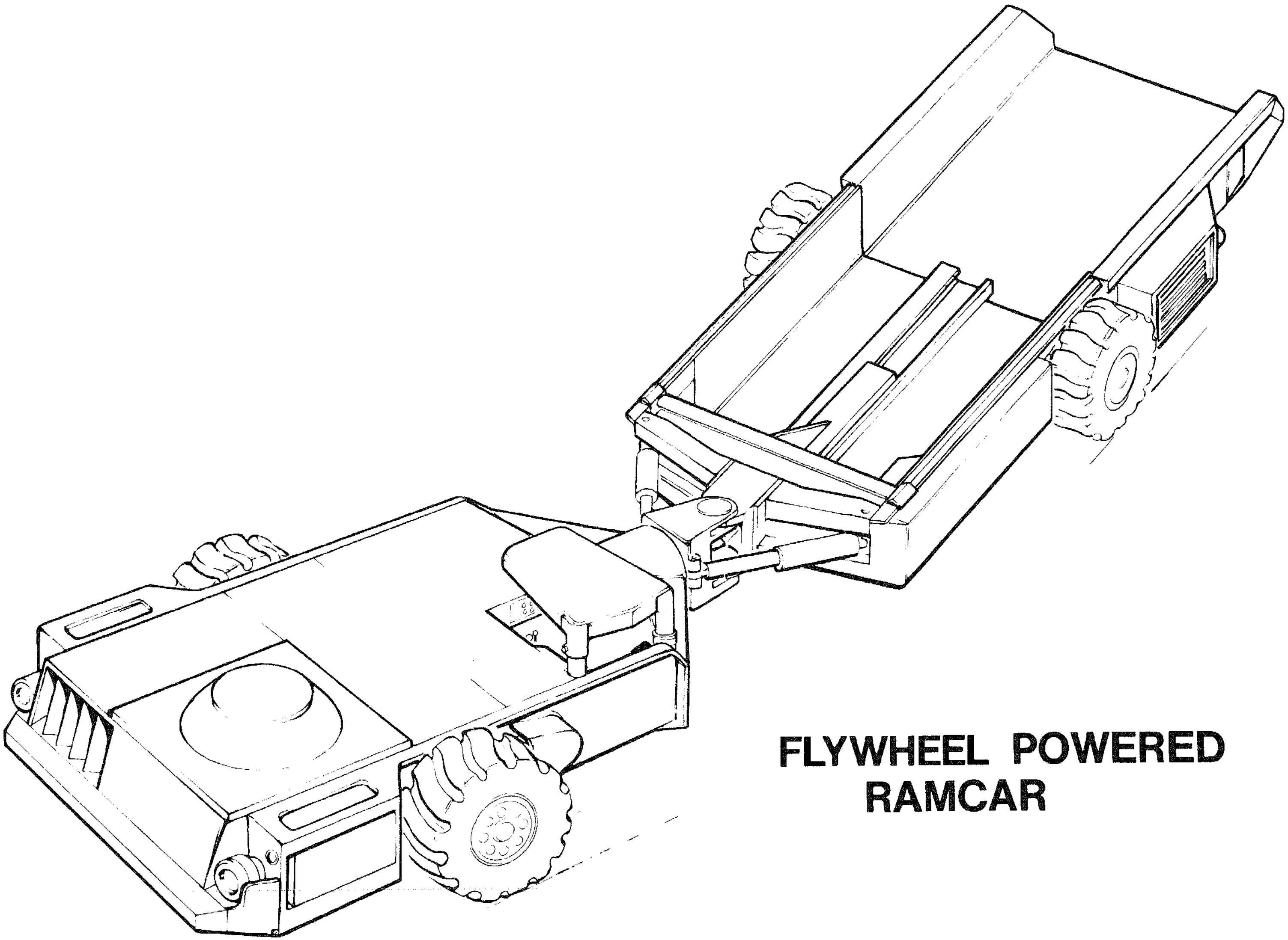


Fig. 97. Artist Sketch - Flywheel Powered Ramcar

An underground evaluation with this system would probably be in a seam height of 60" or more. This would allow raising the operator seat to improve visibility over the flywheel dome.

Cooling of the components in the flywheel power package can be accommodated in the proposed circuit including the torque converter and the drive motor cooling fan. Electrical changes are straight forward.

In conclusion, the four wheel drive RAMCAR is well suited to this application, and changes to the existing vehicle are straight forward.

D.2. VEHICLE SELECTION

In the Phase I study of this contract, a 43" diameter flywheel was selected to provide 4.5 kW hrs of useable energy (6 kW hrs total). It was established that this energy would be adequate to meet the worst mine bottom condition of 300 lbs./ton rolling resistance and 5% uphill grades with full load, in a six foot seam height. This would result in an overall diameter of 50 to 52 inches for the packaged flywheel with containment.

The original plan to install a flywheel system in a conventional shuttle car was abandoned because the only available space, the cable reel compartment, could only accommodate a disc of approximately 20" diameter. This fact led to investigating articulating tractor-trailer haulage vehicles like the Jeffrey RAMCAR. The tractor module of these vehicles has a large open area for the prime mover, which is usually batteries or a diesel engine.

Since several RAMCAR models are curently in production or in the prototype stage, a relative comparison rating was used to evaluate the merits of each for the flywheel application. Table LXIV shows the results of the analysis. The rating was 10,5 or 1 with 10 the most favorable. Studies had already progressed with the conclusion that the application would require separate motor and control voltages. This fact was assumed to apply equally to all the vehicles evaluated.

The following is a list of candidate vehicles and a brief description of each:

Four Wheel Drive RAMCAR

This is a four wheel drive vehicle with a torque converter-powershift transmission powertrain for use with either steam or diesel engines. It is a low vehicle of heavy structural design suitable for dense material (80-100 lbs./cu. ft.) haulage as well as coal. This unit was developed under the USBM contract "Demonstration of a Steam Powered Face Haulage Vehicle."

404L RAMCAR

This is a new battery powered two wheel drive vehicle with a single constant speed D.C. motor coupled to a torque converter and powershift transmission. It is a low vehicle, structurally designed for coal haulage only.

404M RAMCAR

This is a well established production vehicle. This RAMCAR is battery powered. D.C. tram motors are mechanically coupled directly to each of the two drive wheels. The structural design is heavy throughout and well proven.

410M RAMCAR

This is a new diesel powered two wheel drive vehicle with a torque converter-powershift transmission drive train. The structural design is heavy throughout and intended for dense material haulage. It is also a low height vehicle design.

Table LXV lists each consideration and a brief description of factors used in its evaluation.

A review of the data in Table LXIV shows that no one vehicle is superior in a majority of categories such that its overall rating makes it unique. Instead the results indicate several, if not all, vehicles could perform the desired task. However, in the final selection for the vehicle recommendation, two considerations must carry more weight than the others, four-wheel drive and payload capacity. Since the demonstration vehicle may operate in poor bottom conditions and a minimum average load of 269 cubic feet is required to assure cost effectiveness, the four-wheel drive RAMCAR selected.

Both the battery powered 404L and the diesel powered 410M are good alternate selections.

TABLE LXIV

RELATIVE RATINGS FOR VEHICLE SELECTION

FACTORS	4-WHEEL DRIVE	404L	404M	410M
OPERATING CONSIDERATIONS				
Proven Underground	5	10	10	5
Payload Capacity	10	5	10	10
Drive Line Efficiency	5	5	10	5
Repair Accessibility	5	10	5	10
Four Wheel Drive	10	1	1	1
Visibility	10	5	1	10
Potential Investment Cost	1	10	5	5
SubTotal	46	46	42	46
DESIGN CONSIDERATIONS				
Flywheel Package Space Available	5	5	5	5
Drive Motor Available	5	5	5	5
Electrical Controls Available	5	10	10	5
Flywheel Package Cooling	10	10	1	10
Auxlliary Packages	10	5	1	5
Subtotal	35	35	21	30
OVERALL RATING	81	81	62	76

TABLE LXV

DEFINITION OF VEHICLE SELECTION FACTORS

Operating Considerations

Proven Underground - ability of the car to operate reliably for an underground demonstration period of 3 to 4 months based upon past vehicle experience.

Payload Capacity - the vehicle rated trailer capacity adequate to assure a minimum load of 269 cu. ft. under loading conditions.

Drive Line Efficiency - the overall calculated efficiency based on component data verified by test.

Repair Accessibility - relative ease to maintain and service vehicle based on simplicity of design and space.

Four Wheel Drive - four wheel drive vehicle vs. two wheel drive. Increased ability to operate in poor bottom conditions and/or steep grades.

Visibility - ability of the operator to see beyond the periphery of vehicle.

Potential Investment Cost - cash outlay to purchase vehicle before modification to install flywheel system.

Design Considerations

Flywheel System Space Available - Volume, or more specifically plan view area, to install system without major structural or component modification.

Drive Motor Availability - based on engineering design time, if required, procurement lead time, and cost.

Electrical Control Availability - based on utilizing existing components, new design requirements, procurement lead time, and cost.

Flywheel Package Cooling - ability to design a simple and reliable system which can be integrated with the vehicle powertrain cooling requirements.

Auxiliary Packages - availability of spare power take-off and space to accommodate flywheel auxiliary systems.

D.3 VEHICLE PERFORMANCE AND EFFICIENCY

After selecting the four wheel drive RAMCAR as outlined in Section D.2, the existing powertrain was reviewed for its compatiability with the proposed flywheel system. It was decided to keep the powertrain intact and replace the Steam or Diesel engine with a constant speed D.C. Motor. This is the same concept employed in the newest Jeffrey Battery Powered RAMCAR, the 404L.

Jeffrey had first used the torque converter powershift transmission drive on a diesel powered haulage vehicle (model 410H) and its success has lead to more recent vehicles employing this concept; for example, the 404L just mentioned, the four wheel drive RAMCAR and the most recent design, the 410M Diesel RAMCAR. Jeffrey chose the mechanical drive line over individual electric motor drive for the following reasons:

- Components are mass produced on expensive tooling for reduced cost and increased quality.
- Reduced capital investment.
- High efficiency.
- One or more hydraulic pump drives available.
- Convenient component set-up for 4-wheel drive.
- Vehicle electrical system consists of only the lights and a simple motor start circuit.

By way of comparison, the tram motor system employed in the type 404M vehicle requires an additional motor to drive the hydraulic pump. That version has three motors and the power drive efficiency is only slightly different from the four-wheel drive RAMCAR. If a cable reel is employed, such as on a conventional shuttle car, there is another significant drop in power drive efficiency due to the cable.

Prior to starting the powertrain calculations a search was made for available D.C. motors in the 50-75 H.P. range. Voltages of 240 and 550 were initially explored but electrical considerations (by General Electric) of the flywheel charging dictated 550 volts as a definite preference.

Jeffrey has a fan cooled 75 H.P. motor designed for either 500 volts or 250 volts. This motor has only been manufactured in the 250V version but all patterns and tooling are available for 500 volts. Based on past experience with commutator arcing from coal mine dust contamination the 500 volt version has generally not been recommended.

Contacts with suppliers including Louis Allis and Reliance Electric Company indicated no concern over motors of this voltage. However, neither company had a readily available motor although both expressed interest in providing one.

Utilizing a typical motor speed droop curve, calculations were made to match a motor to the existing or a slightly modified powertrain. Eleven (11) combinations were investigated before the optimum efficiency, tractive effort and vehicle speed was obtained. Several early attempts with 50 H.P. revealed this to be insufficient power for bad bottoms and/or grades. The remaining combinations were then based on 75 H.P.

The motor speed torque curve was derated for the full time losses from the transmission charge pump and the vehicle hydraulic pump (referred to as parasitic losses). These losses were estimated as 8 H.P. and decrease the power input to the torque converter as shown in Figure 98.

Calculations for the final three (3) sets of combinations are summarized in Table LXVI. The performance curve of each is illustrated in Figures 99, 100 and 101 respectively. All calculations were made with the best available torque converter, the Funk Manufacturing Company 12-3/4 inch model.

The first two sets illustrate similar efficiencies; however, the first set (No. 9) requires major powertrain revisions; i.e. new ratio differential with higher torque levels, replacement of the outboard gearbox with pillow blocks, and a new chain drive ratio. All this is required for only a slight increase in efficiency of 1 to 1½%. The second set (No. 10) is identical to the present powertrain except the transmission must be changed to an optional input gear set ratio available from the manufacturer, Funk.

The last set (No. 11) illustrates the sacrifice in efficiency if the transmission ratio is not modified. The higher transmission speeds result in considerable windage losses. It also shows higher vehicle speeds at the expense of gradeability.

The summary (Table LXVI) illustrates the differences noted above and shows that the selected powertrain, set number 10, has the following performance at 165 lbs/ton rolling resistance:

Empty Vehicle Speed	4.6 MPH
Loaded Vehicle Speed	3.9 MPH
3rd Gear Gradeability	5%
Efficiency Range	57 - 63%

This preliminary selection should be reviewed in subsequent design phases to fully evaluate the impact of efficiency.

A block diagram of the vehicle powertrain (with component efficiencies) is shown in Figure 102.

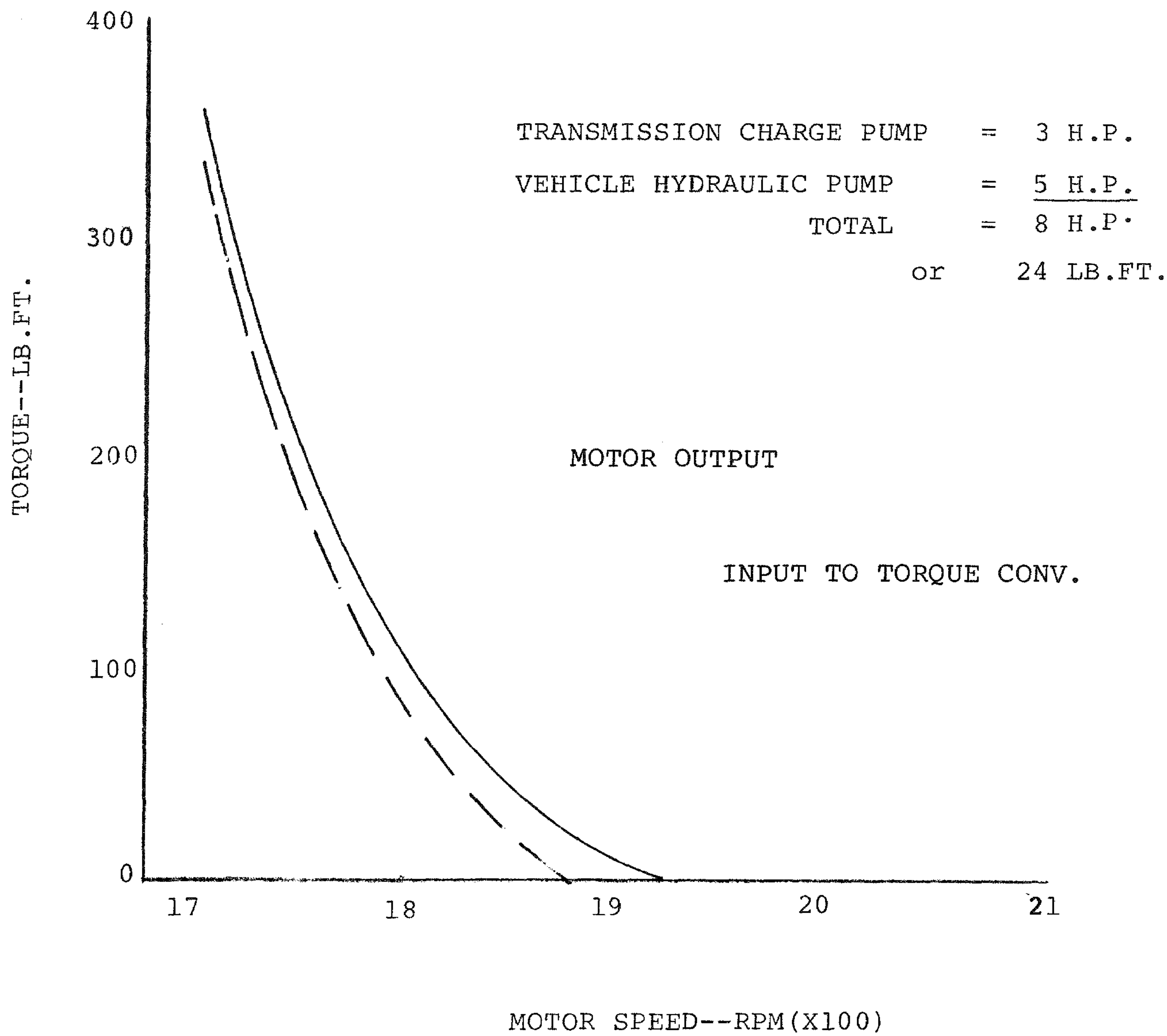


FIGURE 98 PUMP DRIVE POWER LOSSES

TABLE LXVI

EFFICIENCY CALCULATIONS FOR THREE COMBINATION SETS

Set No.	RR #/Tons	Empty MPH	Empty HP	Loaded* MPH	Loaded* HP	Empty Heat HP	Loaded Heat HP	Empty Loaded % Eff.
9	165	5.2	61	4.2	74	11	13	57/65
10	165	4.60	58	3.85	72	8	8	57/63
11	165	5.30	65	4.25	76	9	9	51/59

* ASSUMES 9 TON LOAD

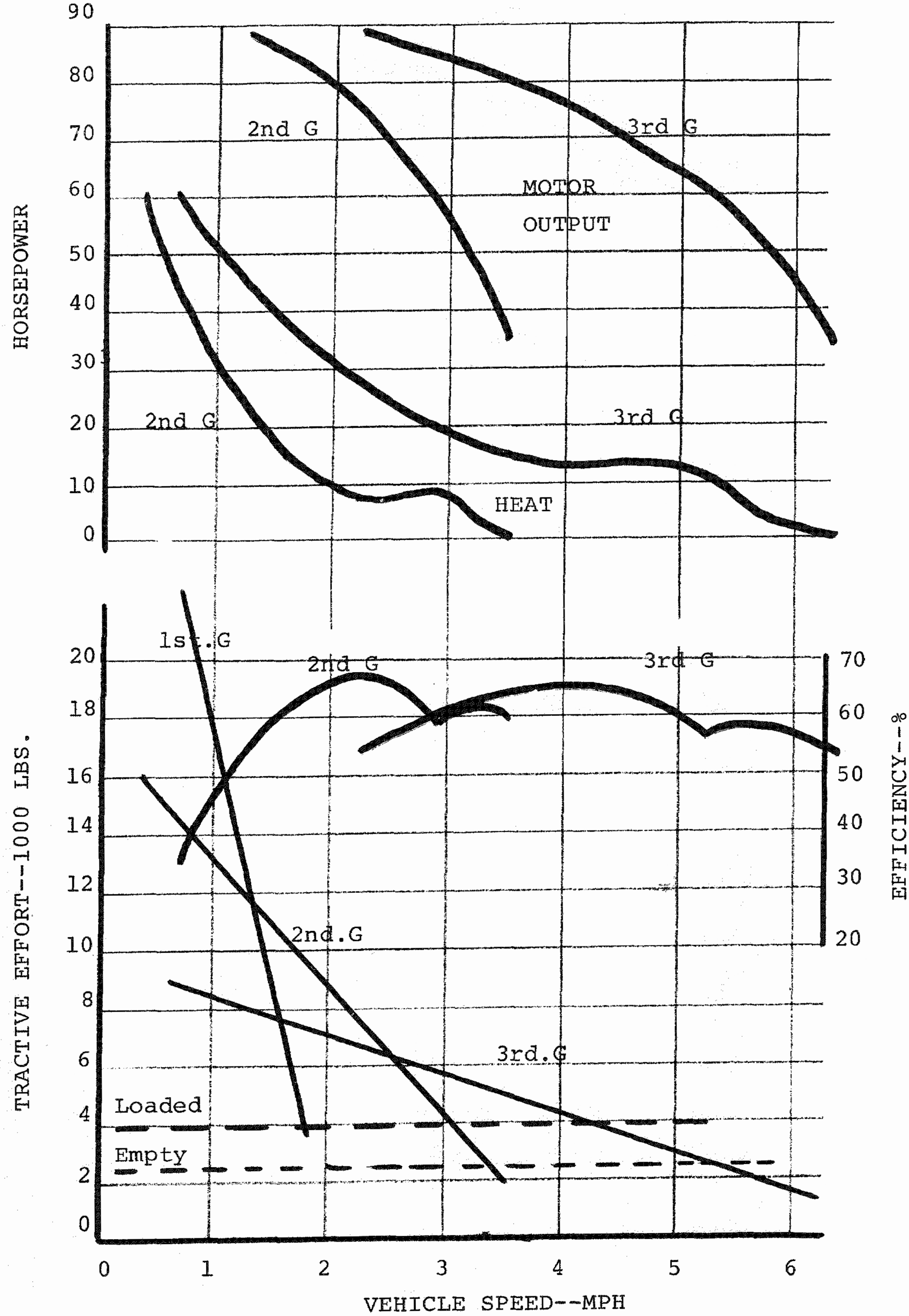
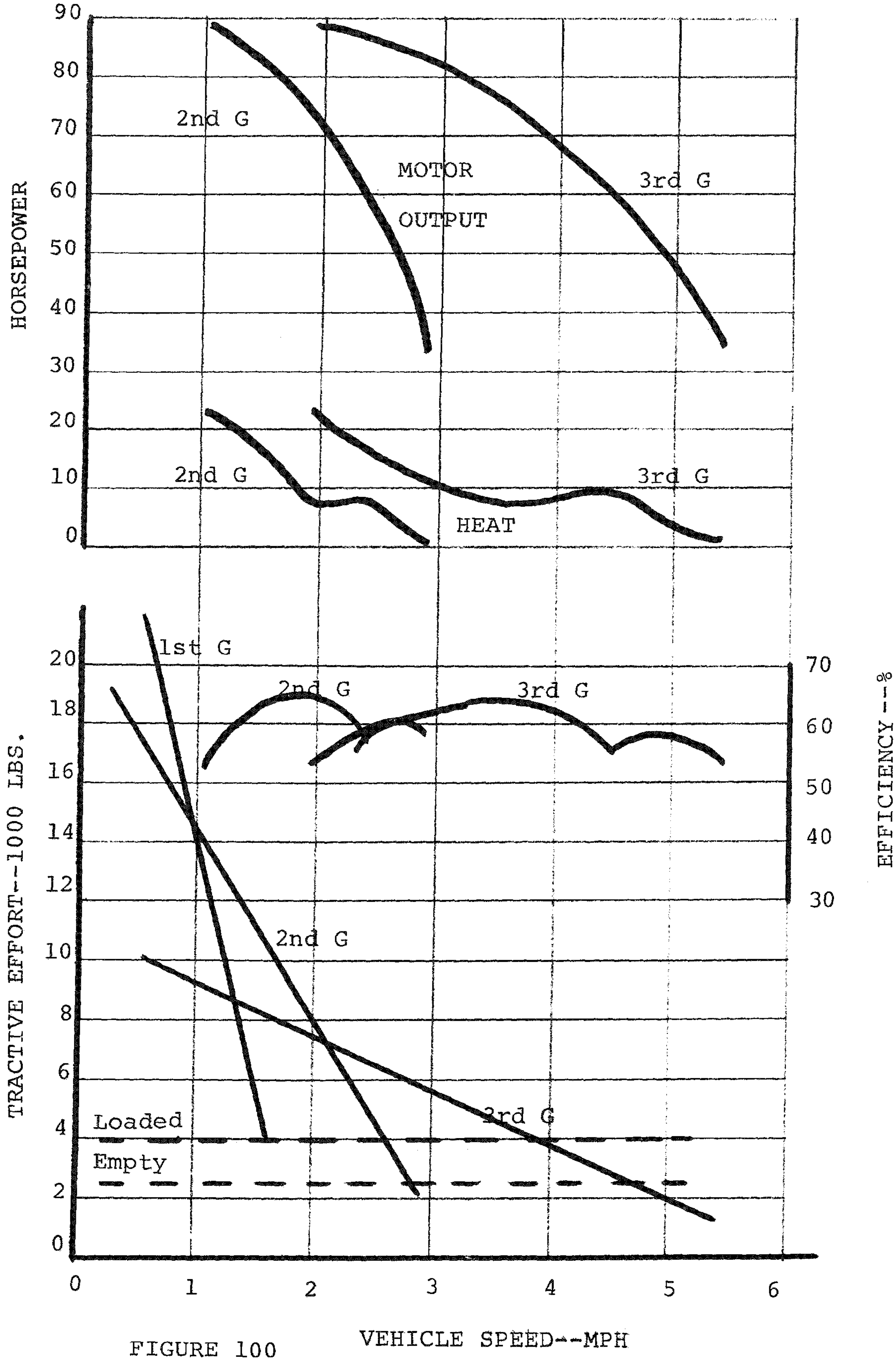


FIGURE 99 No. 9



90
80
70
60
50
40
30
20
10
0
HORSEPOWER
2nd G
MOTOR
OUTPUT
3rd G
2nd G
3rd G
HEAT
20
18
16
14
12
10
8
6
4
2
0
TRACTIVE EFFORT--1000 LBS.
1st G
2nd G
3rd G
2nd G
3rd G
Loaded
Empty
70
60
50
40
30
EFFICIENCY--%
0
1
2
3
4
5
6
VEHICLE SPEED--MPH

FIGURE 100

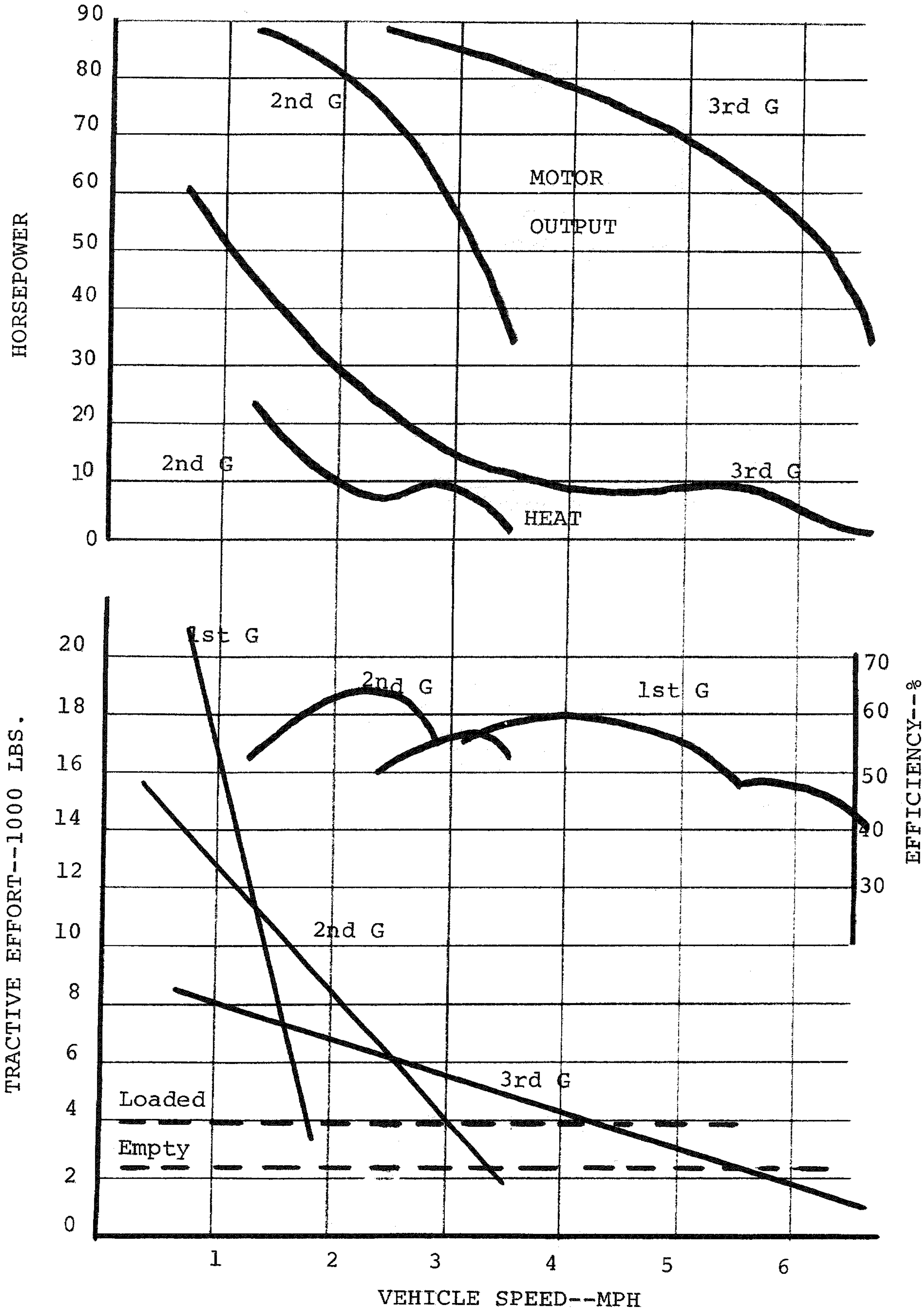
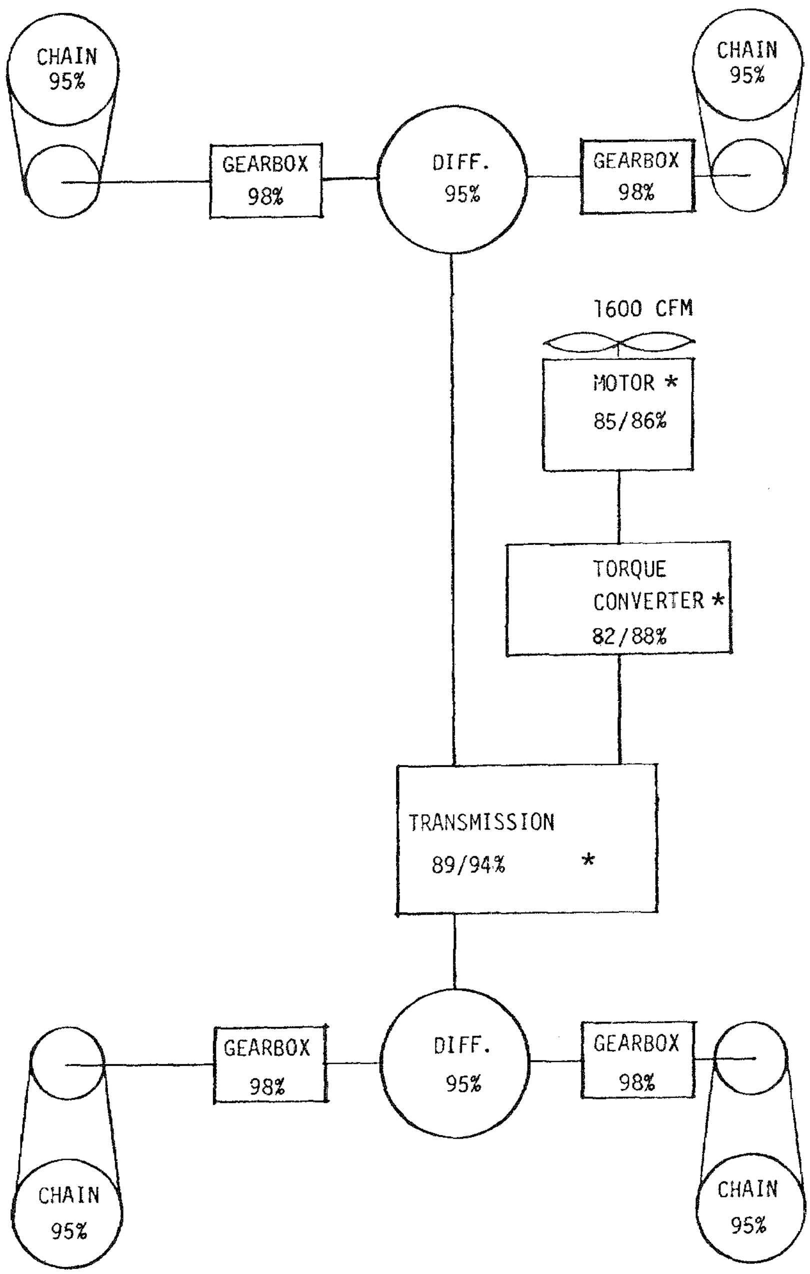

HORSEPOWER
90
80
70
60
50
40
30
20
10
0
2nd G
3rd G
MOTOR
OUTPUT
2nd G
3rd G
HEAT
TRACTIVE EFFORT--1000 LBS.
20
18
16
14
12
10
8
6
4
2
0
1st G
2nd G
1st G
EFFICIENCY--%
70
60
50
40
30
2nd G
3rd G
Loaded
Empty
1
2
3
4
5
6
VEHICLE SPEED--MPH

FIGURE 101



* EFFICIENCY UNLOADED/LOADED
165 LB/TON ROLLING RESISTANCE

FIGURE 102 POWER TRAIN COMPONENT EFFICIENCY DIAGRAM FOR SET NO. 10

Additional work investigated the effect of changing the flywheel and operator seat heights. Figure 106 illustrates improved visibility when the flywheel is lowered 1-1/2" to simulate a smaller capacity unit, such as in Figure 20. It should be noted that the package for the smaller unit is actually 3-3/4" lower. Figures 107 and 108 show improved visibility with both the smaller 4.5 KWHr. and 6.0 KWHr. flywheels with the operator raised 4". The figures illustrate the flywheel should be made and installed as low as possible and the operator should be raised.

Figure 109 shows the overall vehicle configuration, dimensions and load capacity. The vehicle specifications are contained in Table LXVII.

D.4 OPERATOR VISIBILITY AND COMPONENT LAYOUT

As stated previously, the four wheel drive RAMCAR was recommended as the test vehicle for the flywheel system. Figure 103 shows a preliminary tractor general arrangement layout of the major flywheel, powertrain, and electrical components. The flywheel package was placed adjacent to the tractor bumper so that obstruction to operator visibility by the flywheel dome would be minimized. Figures 104 and 105 show the difference in visibility when the flywheel is placed near the front bumper and near the operator respectively. The flywheel used as a base line is the double disk version (shown in Figure 38) which has 6.0 kWhrs useable energy. For Figure 104 through 108, the camera position was fixed laterally.

The flywheel control case is placed between the flywheel and the bulkhead separating the operator area. This simplifies the cable mounting to the circuit breaker box on the operator side of the bulkhead.

The 500V D.C. motor (75 HP at 1750 RPM) is placed on the side opposite the operator's compartment. The torque converter with two auxiliary pump drives is mounted on the shaft end of the motor by means of an adapter housing. A two pass air to oil cooler and fan shroud arrangement are mounted on the rear of the motor to utilize the motor's external fan to cool the torque converter and flywheel system. The powertrain components on the operator side of the bulkhead are the same as original in the RAMCAR.

Adequate space is available in the operator side of the bulkhead for the 500V D.C. motor circuit breaker and 120V D.C. control circuit breaker box. The switch box (adjacent to the operator's left hand and the transmission control) houses all the operator electrical controls; separate tractor and trailer lights and the motor start and stop push buttons. The start button is interlocked with the transmission controls so that the motor can only start with the transmission control lever in the neutral safety start position. The motor relays and starters are housed in the large control case shown on the left hand fender. Several other locations are also available for the large control case, the final selection probably being determined by the simplest cable routing.

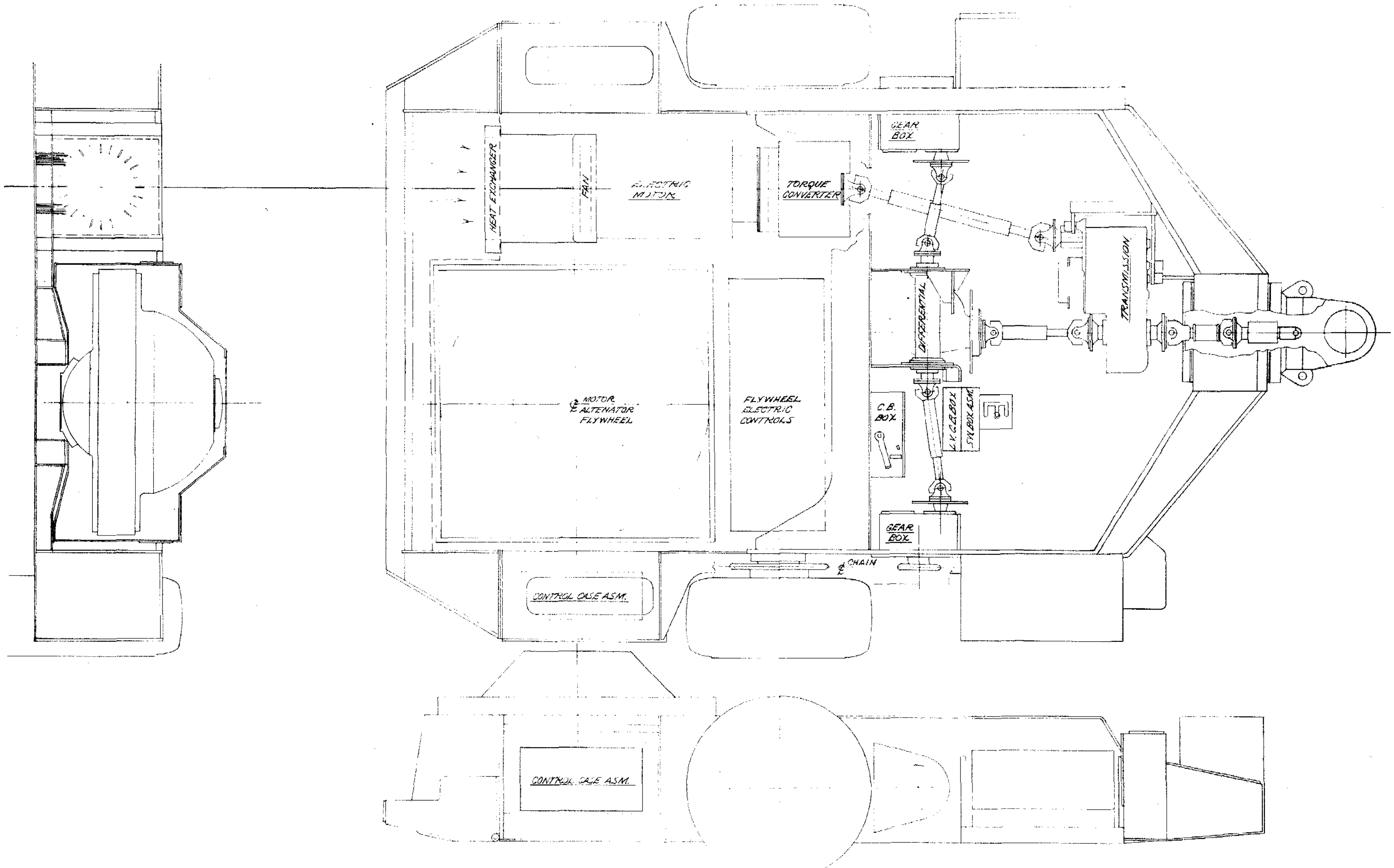


FIG. 103 Preliminary Layout of Tractor With Flywheel Power

TABLE LXVII

FLYWHEEL POWERED RAMCAR PRELIMINARY SPECIFICATIONS

Electric Motors	
Make	Jeffrey
Model	154E914
Rated HP	75 HP at 1750 RPM
High Idle	1880 RPM
Voltage	550 or 500 D.C.
Transmission	
Make	Funk
Model	1000 Series
Type	Powershift
Gears	3 forward, 3 reverse
Ratios	3.34:1, 1.82:1, 0.97:1
Torque Converter	12-3/4 inch
Stall Ratio	2.75:1
Differential	
Make	Spicer-Heavy Duty
Type	Hypoid
Ratio	5.43:1
Outboard Gearbox	
Make	Jeffrey
Type	Parallel shaft
Gears	Spur
Ratio	2.78:1
Chain Drive	
Make	Jeffrey
Chain Pitch	2"
Reduction	2.15:1
Tensile Load	60,000 lbs.
Suspension	
Type	Rotating pivot between tractor/ trailer

TABLE LXVII
(con't)

Steering	
Type	Articulation
Actuation	Two hydraulic cylinders
Steering Angle	60° left & 60° right
Control	Hard lever/hydraulic power
Service Brakes	
Type	Power disc/caliper
Location	4 outboard gear boxes
Control	Foot pedal/hydraulic power
Caliper	Mico
Parking/Emergency Brakes	
Type	Disc/caliper
Application	Spring applied
Release	Hydraulic power
Caliper	Mico
Control	Hand lever
Location	Input tractor differential
Hydraulic System	
Pump Type	Gear, double
Pump Rating at 1800 RPM	14, 28 GPM
Pump Drive	Torque Converter 1:1
Steering	Priority/demand valve
Max. Pressure	1800 PSI
Brakes	Priority/accumulator
Brake Pressure	1800 PSI
System Type	Accumulators with Unloading valves.
Electrical System	
Approval	MESA
Motor Protection, 500V. D.C.	Circuit breaker
Control Protection, 120V. D.C.	Circuit breaker
Motor Control	Constant speed, start and stop
Lights	Separate tractor and trailer, ON & OFF

TABLE LXVII
(con't)

Weight and Performance	
Unloaded Weight	Front axle 15,000 lbs. Rear axle 15,000 lbs. EVW 30,000 lbs.
Loaded Weight	Front axle 15,000 lbs. Rear axle 43,000 lbs. GVW 58,000 lbs.
Max. Safe Load	14 tons
Vehicle Performance	D.3-3
Tires	14.5 X 15
Dump Time	20 seconds
Trailer Capacity-Struck W/12" sideboards	300 cu. ft.
Dimensions	
Overall Length	34'
Overall Width	10' 4"
Wheel Base	18'
Overall Height	42"
Ground Clearance	6" tractor 8" trailer
Turning Circle Outside	21' 6"
Inside	10' 4"
Canopy	
Type	Cantilever
Height	42' 54" adjustable
Features	Swing out for access/ egress
Operator Compartment	
Seat Position	Central - transverse
Seat	Padded seat/back rest
Pedals	Tram/brake
Levers (Left Hand)	Forward/reverse Speed 1st, 2nd, 3rd gears Parking brake Start motor Stop motor Lights
Levers (Right Hand)	Steering Ramplate Telescopic body

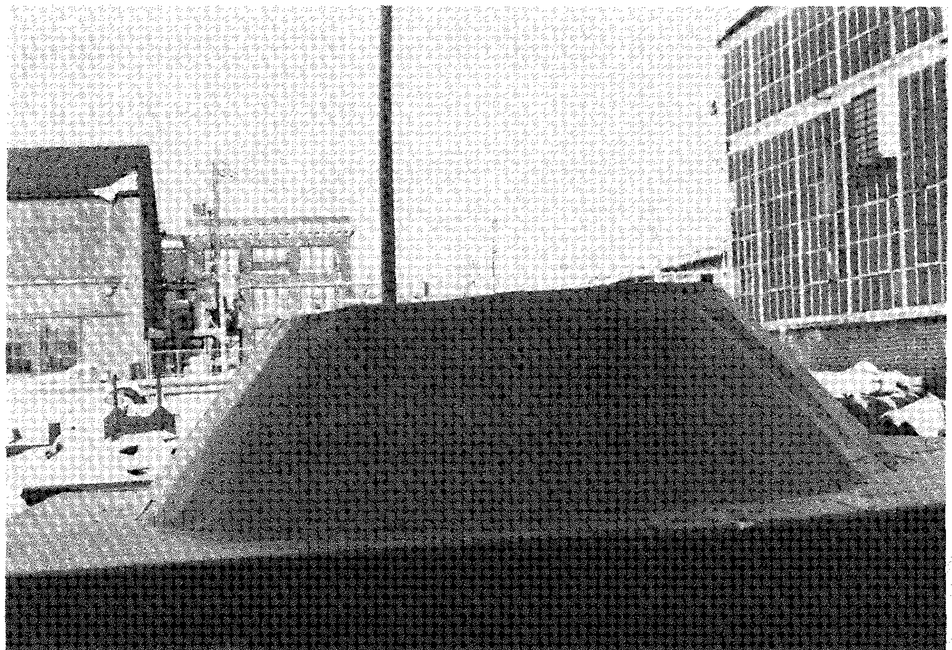


FIGURE 104

FULL SCALE MOCKUP WITH FLYWHEEL NEAR BUMPER

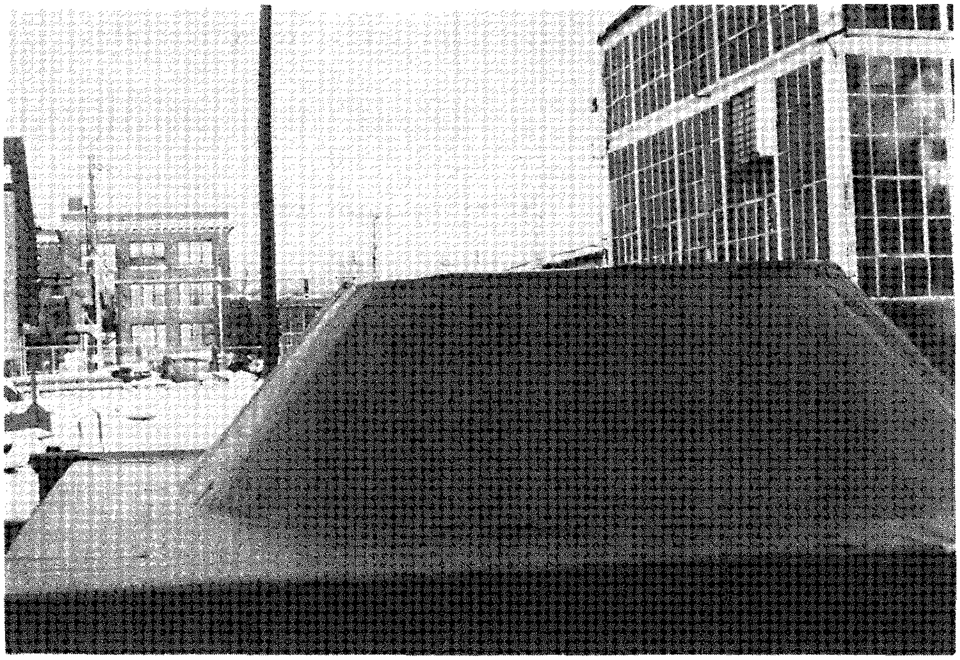


FIGURE 105

FULL SCALE MOCKUP WITH FLYWHEEL CLOSER TO OPERATOR

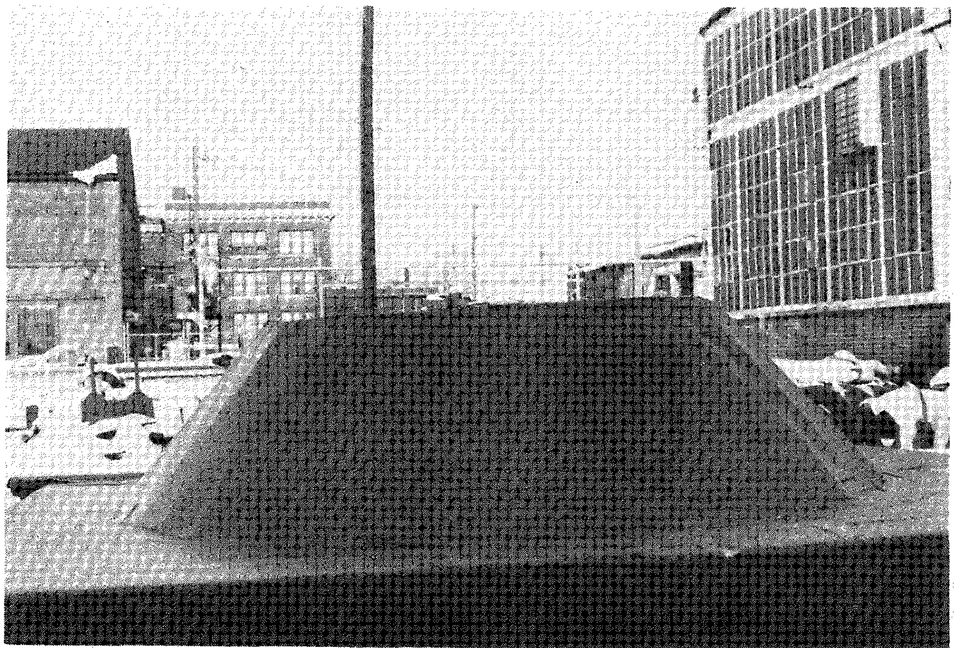


FIGURE 106

FULL SCALE MOCKUP WITH LOWER (1-1/2") FLYWHEEL

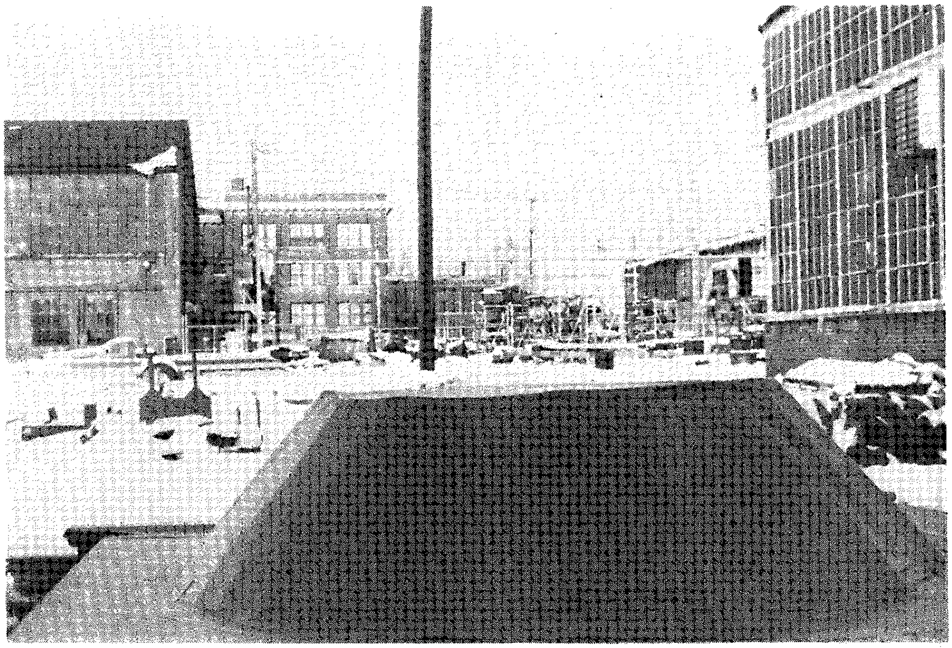


FIGURE 107

FULL SCALE MOCKUP WITH 4.5 KWHR. FLYWHEEL
(operator's seat raised 4")

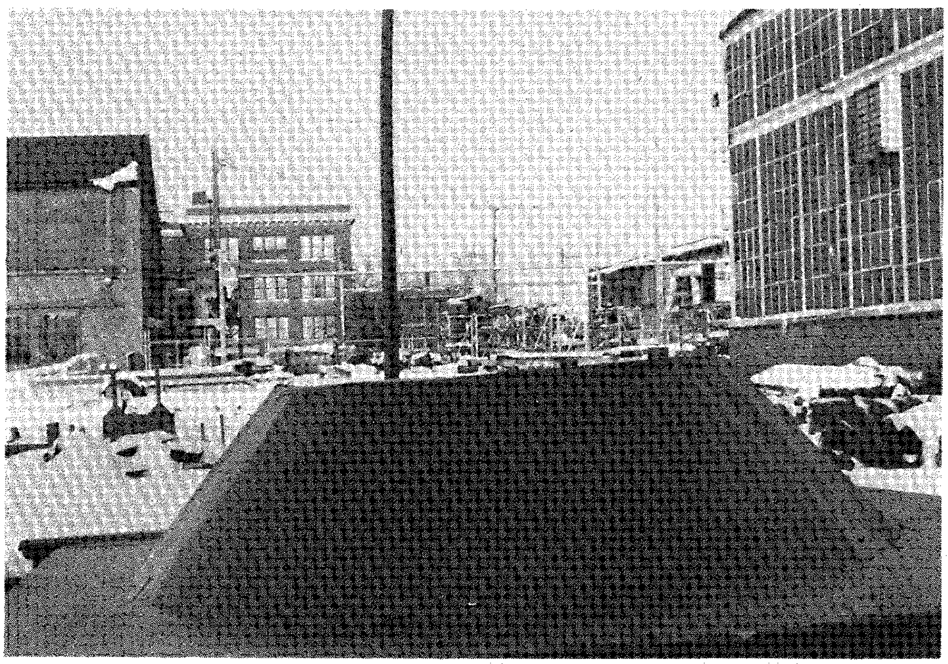
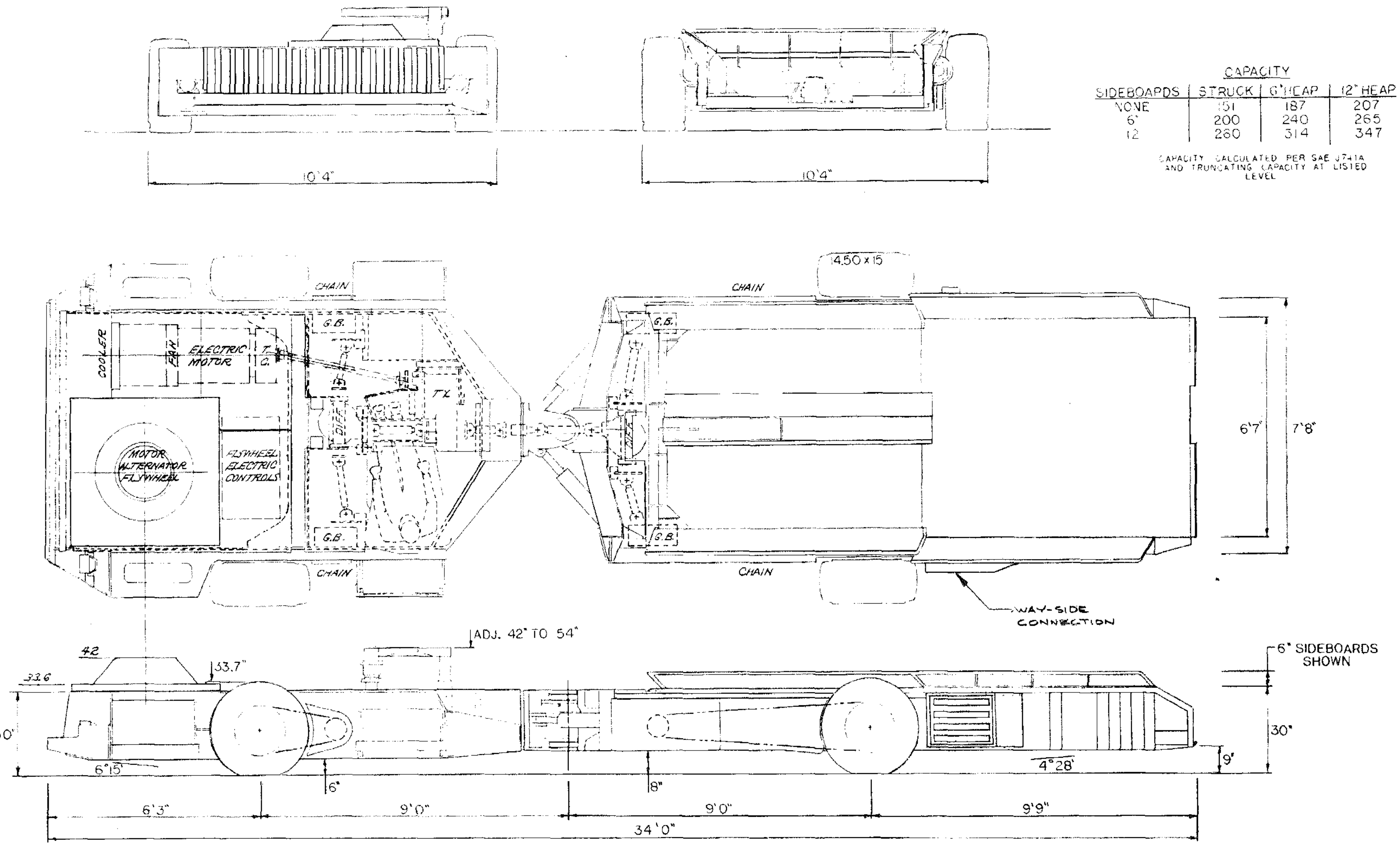


FIGURE 108

FULL SCALE MOCKUP WITH 6.0 KWHR. FLYWHEEL
(operator's seat raised 4")



CAPACITY

SIDEBOAPDS	STRUCK	6" HEAP	12" HEAP
NONE	151	187	207
6"	200	240	265
12	260	314	347

CAPACITY CALCULATED PER SAE J741A AND TRUNCATING CAPACITY AT LISTED LEVEL

Figure 109 Overall Vehicle Configuration for Flywheel-Powered RAMCAR

D.5 FLYWHEEL AND VEHICLE COOLING SYSTEM

When a torque converter drive is used a cooling system must be provided to cool the torque converter fluid. Normally these powertrains are used with gas or diesel engines and the engine coolant or fan airflow is used as a heat transfer medium. On underground mining equipment without engines another cooling method must be found. Since water is not available on haulage vehicles, Jeffrey has successfully used airflow from externally fan cooled electric motors. This is the method proposed and investigated here.

Figure 109 shows a block diagram of the proposed system. This system integrates the normal torque converter system with the cooling requirements of the flywheel and its control case, thereby eliminating the need for a seperate system for the flywheel components. Oil flow is provided by the transmission charge pump mounted on the torque converter. After the oil exits the torque converter it enters a two-pass air to oil cooler where the motor fan draws mine air at typically 55° F. through the cooler lowering the oil temperature from 185° to 162° F. The exhaust mine air is raised to 105° F. The cooled oil is then routed to the flywheel control case to take advantage of the lower oil temperature to cool the electrical components. The components being cooled are placed in series and the temperature rise across the flywheel components is shown for average heat loads during the maximum energy use conditions anticipated. The design heat load for the torque converter allows continuous operation at the shift point between 2nd. and 3rd. gears.

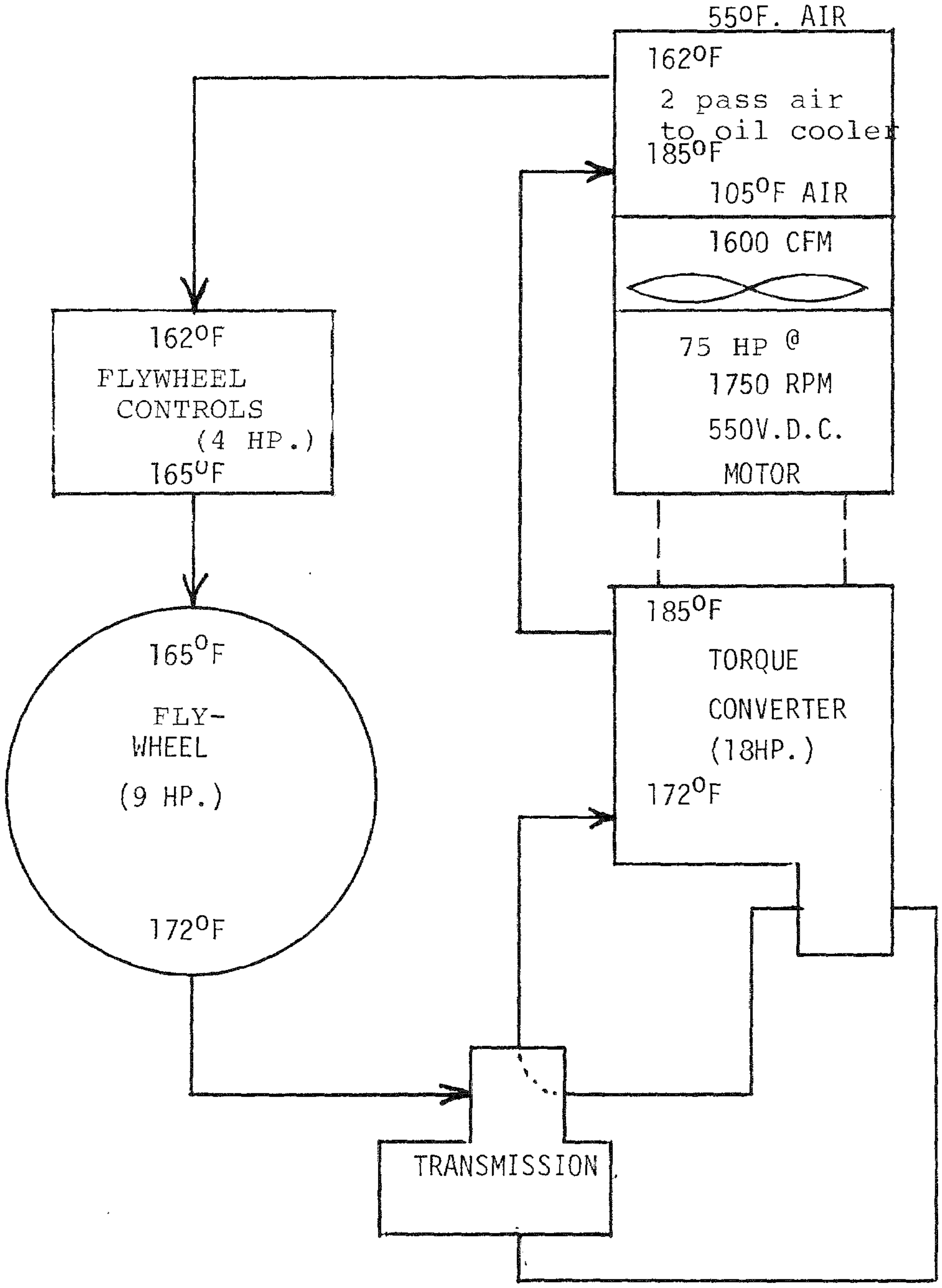


FIGURE 110 HYDRAULIC FLUID FLOW DIAGRAM AND TEMPERATURE DIFFERENTIALS

D.6 VEHICLE ELECTRICAL SYSTEM

As outlined in Section D.3 on Vehicle Performance and Efficiency one advantage of using a constant speed D.C. motor with a torque converter drive was the simplified electrical system. Circuits for seperate tractor and trailer lights and D.C. motor start/stop are the only requirements.

Initially 120, 240 and 500 volts were considered for controls. The motor voltage, 500V, would have resulted in the simplest and cleanest design from a vehicle standpoint but was ruled out because a completely new system was required. 120 and 240 volts were both practical but the 120V was selected because more existing hardware could be utilized. The components proposed are from the Battery Powered (120V) 404L RAMCAR. General Electric personnel agreed that a separate power supply of approximately 500 watts could be readily provided.

A block diagram of the vehicle electrical system is shown in Figure 111. The motor circuit breaker box and the switch box can be existing designs. The main control case is somewhat questionable and may need enlarging to accomodate increased contactor load because of the higher voltage motor, 500 volts, (404L has a 120V 30 H.P. motor). The control circuit breaker box is new. The vehicle lights are presently 24V D.C. but 120V components are readily available.

In addition to the above, provision for wayside power connection must be included. This is assumed to be located on the side of the trailer between the wheel and the discharge end. As shown in Figure 97, it is located on the operator side. This subject was not investigated further in the Jeffrey Part of the study. A method for routing the high amperage rated cables through the pivot to the flywheel control case must be determined.

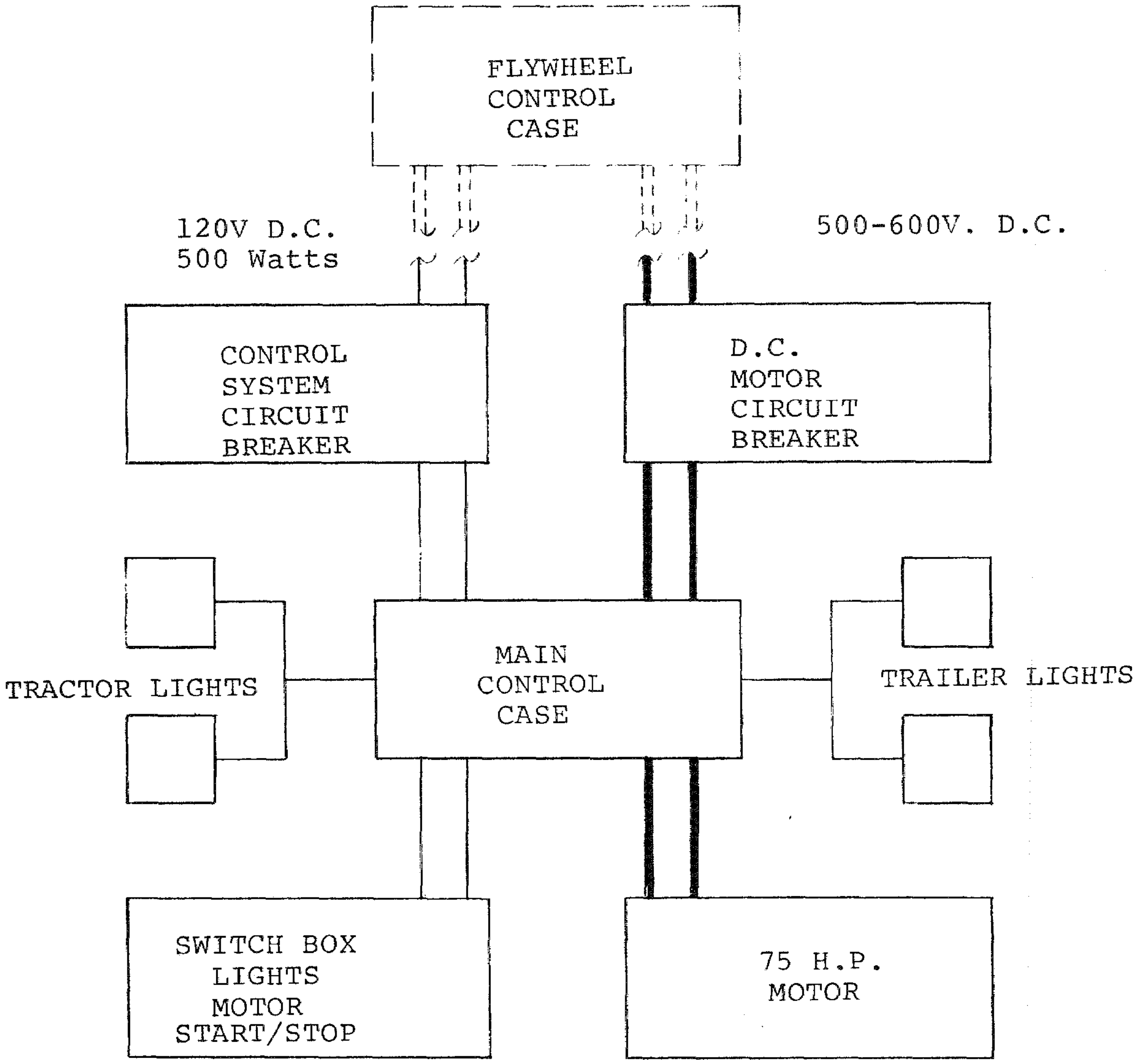


FIGURE 111 PRELIMINARY BLOCK DIAGRAM FOR VEHICLE ELECTRICAL SYSTEM

D.7 VEHICLE MODIFICATION TASKS

The purpose of this section is to document tasks, decesions, alternatives, problems, etc. which collectively will comprise the overall scope of modifing the selected vehicle for flywheel power. The following list will undoubtedly overlook some tasks, but tabulating the items throughout the prelimenary study will be helpful in estimating and planning a follow-up program.

Tractor Frame

- Design motor mounting plate with alignment for proper drive shaft installation.
- Provide structure across deck plate to support isolation mounts of flywheel.
- Rework bumper cover plate to fit around flywheel containment housing.
- Provide mounting for flywheel electrical control case.
- Design new tractor covers.
- Provide for vehicle main control case mounting.
- Mount circuit breaker boxes (2) and switch box.
- Add access holes for cable routing.
- Provide bulkhead mounting for emergency spin-up connectors.
- Revise grill design.

Operator Compartment

- Design transmission control for neutral safety start with switch box.
- Raise operator seat for improved visibility.