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AUTOMOBILE AIR-CONDITIONING UNIT

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II. SUMMARY

In this study the refrigerant in the automobile air-conditioner is compressed by thermal energy in a unique compression system rather than by work in a standard compressor. The compression uses an intermittent compression process with a solid absorbent. The vapor is absorbed by an absorbent at relatively low temperature and ejected as the absorbent temperature is raised. A set of one way valves limits flow to one direction. Major contributions are heat transfer requirements, molecular sieve-refrigerant matching, minimizing non-producing mass, solving thermal fatigue and shock problems, and applying this to automobile air-conditioning.

The performance study shows energy savings up to fifty percent are possible, depending on engine load. A twenty percent energy savings with the vehicle tested with the air-conditioner in operation is average. The study also showed that less fuel is used with the windows open than with the air-conditioner operating.

The air-conditioner was not completely operational at the writing of this report (main oscillating controls), but the technical feasibility has been demonstrated.

A patent application initiated before the award of this contract has been completed and is presently pending.

III. BACKGROUND AND GENERAL STUDY RESULTS

This project studies the unique utilization of energy sources, uses unique applications of high quality molecular sieve manufacturing applications, uses economical simple configurations for refrigerant compression chambers and controls, uses practical and economical cooling systems for the refrigeration compression process, solves thermal fatigue and shock problems, provides some thermal storage and devises a parallel method of refrigerant compression to standard refrigerant compressor in heat pumps, air-conditioning, bus and truck air-conditioning, truck refrigeration, building heating and cooling, commercial refrigeration systems, and solar heating and cooling. The compression system is devised to be simple, economical, and conveniently installed in present heat pump, air-conditioning and refrigeration systems. The conversion of present heat pump, air-conditioning and refrigeration systems to this compression system is technically convenient in many applications.

Through this compression system, the converted cycle will replace power, an ideal expensive energy, with low grade thermal energy as the energy input to provide refrigerant cycle compression. Using the compression system in parallel to a standard compressor allows the use of thermal energy for compression when available and the standard compression otherwise. This is particularly useful for a solar application.

Other than the compression system, the refrigeration cycles are not altered. The identical components including evaporator, condenser, throttling valve, etc. and the identical refrigerant can be utilized. This makes present heat pump (air-conditioning and refrigeration) systems convenient to convert to the new compression system.

By shutting off the refrigerant lines to the absorption chamber, a residual quantity of thermal (cooling) storage can be made available. This

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allows immediate cooling upon start-up, for example an automobile air-conditioning system.

The workings of this system will become apparent in the following detailed description.

Figure 1 shows the basic components of the standard liquid-vapor heat pump (air-conditioning) cycle. The refrigerant vapor is compressed by the compression system. This compression system can be either the standard compressor presently used in heat pumps (air-conditioners) or the unique compression system disclosed in this report. The new compression system is defined in the following figures. In Figure 1 the refrigerant leaves the compressor and enters the condenser where energy is removed and the vapor liquifies. If the purpose of the cycle is heating, the energy given off by the refrigerant at this location provides the heating and the system is in the heat pump mode. The liquid leaves the condenser and passes through a throttling valve to the evaporator. In the evaporator, the refrigerant absorbs energy and provides cooling. This mode provides the air-conditioning and refrigeration utilization of the cycle. This cycle is the standard liquid-vapor heat pump cycle. This cycle does not change with the addition of the new compression system. The controls for the new system are basically identical, however, a few control changes are possible. One such change could be a refrigerant capacitance system to smooth the intermittent flow.

As shown in Figure 2, the new compression system can be used independently or in parallel with a conventional compressor. In this case, the new compression system and the conventional compressor comprise the total compression system. By using the new compression system in parallel with a standard compressor, the system can be used when thermal energy is available part of the time but not continuously. A prime example is a solar energy air-conditioning system. When solar energy is available the compression system is utilized and when solar energy is not available the standard compressor is utilized. The addition of the new

Figure 1. Basic Refrigeration Cycle Schematic

Figure 2. Parallel Compression Systems

system to a standard heat pump or air-conditioner is simple and refrigerant flow can be easily controlled with a sequence of one way valves as shown in Figure 2. An active control system with positive on-off valves can be utilized to replace the one way valves. Parallel compression applications are many. Industrial applications where waste thermal energy is available part time is a prime application. In supermarkets waste heat from refrigeration compressors can be used directly for heating in winter periods and used for air-conditioning during summer periods. Parallel applications are numerous and can be applied with small or large reciprocating, centrifugal, or screw type compressors.

Figure 3 shows the basic element schematic configuration for the refrigerant flow and the heating and cooling flow. The molecular sieve absorbs refrigerant at relatively low temperatures. The traditional example is silica-gel crystals absorbing water vapor. At relatively low temperatures, silica-gel absorbs water vapor until the vapor pressure is extremely low. When the crystals are heated to a relatively high temperature, in an oven for example, the water vapor is forced out of the pores and exits at a relatively high pressure. In this case, refrigerants are absorbed by molecular sieves at relatively low temperatures, for example 100°F to 200°F, and rejected at relatively high temperatures, for example 150°F to 400°F. A sealed container with inlet check valve and outlet check valve holds the molecular sieve and refrigerant. A flow passage allows hot or cold fluids to transfer heat from and to the container. By using a system of check valves the refrigerant flow is allowed to flow in only one direction. As cooling occurs the temperature decreases, the molecular sieve absorbs refrigerant and produces flow into the low pressure area in the container. Due to the low pressure in the container the outlet check valve closes and prevents reverse flow. When the compression system and molecular sieve is heated the absorbed refrigerant is rejected, the pressure increases, inlet check valve closes, and the outlet check valve opens, and the refrigerant flows from the container at a

Figure 3. Refrigerant and Gas Flow

higher pressure. The system with the molecular sieve performs as a compressor to compress the refrigerant. A relatively warm fluid and a relatively cold fluid must flow intermittently past the chamber to alternate the heating and cooling of the molecular sieve. No fixed shape is required for the chamber and the inlet valve and outlet valve can be essentially at any location. Screens are provided at the inlet valve and outlet valve to contain the molecular sieve particles. Molecular sieve comes as powder, small spheres or small cylinders. These sizes usually vary from 1/16 inch to 1/4 inch diameters. Other configurations are possible.

The critical requirements to make the compression system work economically are the matching of the molecular sieve and refrigerant, minimizing system mass, providing an adequate heat transfer system and designing for thermal fatigue and shock.

The molecular sieve must be dense and absorb a relatively high mass of refrigerant per unit mass of molecular sieve. The molecular sieve pore size must be matched to the refrigerant molecule size. Meeting these requirements minimizes molecular sieve mass and volume. The small molecular sieve volume decreases heat transfer lengths and resistance and minimizes container size and mass. It maximizes refrigerant mass absorbed per unit of molecular sieve mass and per unit of container mass. For example, ordinary silica-gel will not be an economical absorbent for refrigerants. Designing the compression system with the proper molecular sieve is a system requirement. The energy required to heat and cool the molecular sieve mass and container mass is lost. Only the energy removal for absorbing the refrigerant and the energy for regenerating the refrigerant are useful. The thermal energy into the system must be fully utilized. For example, the available energy for automobile air-conditioning or truck refrigeration systems using exhaust gases at idle is critical. If one of these systems fails to use most of the available energy there will not be sufficient energy to provide

the required cooling or refrigeration. Using the vehicle exhaust after passage through a catalytic converter helps both in energy quantity and quality (higher temperatures for heat transfer purposes). However, this study indicates this is not required.

The second critical requirement is to provide sufficient heat transfer for both heating and cooling in a reasonable amount of time. Cycling a compression unit every thirty minutes will not be economically feasible in most cases. To provide sufficient refrigerant flow rate for a reasonable sized compression unit, rapid cycling is required. The non-molecular sieve mass and container mass must be minimized. Heating and cooling this mass uses non-productive energy and slows heating and cooling. This means less cycles per unit of time which results in an even larger system. Heating and cooling castings is normally not feasible (castings usually contain a relatively large mass). Every component must be light weight. Thin wall tubing is essentially required. The circular form is required for high pressures within thin walls to meet pressure requirements. Non-contributing mass must be minimized for technical and economic feasibility.

In the thermal cycling the system is alternately heated and cooled. In the thermal switching process, thermal shock occurs. The change of temperatures causes changes in component size. As a result, thermal stresses occur. The tubes in the compression system must be flexible so that as they heat and cool no large stresses occur. For example, placing a U-turn in the tubes will allow expansion and contraction. Making end plates on the end of cyclinders conical rather than flat will produce slight movements rather than stresses as the piece is heated and cooled. The thermal stresses have been a major problem in the testing of prototype systems in this study. The heat transfer requirements, the design for thermal fatigue and shock, and the critical heat transfer requirements has probably caused the economic and technical failure of past application designs in the solid intermittent absorption heat pump systems.

Figures 4 and 5 show schematics of a number of possible heat flow passages. By extending the fins (conduction paths) into the flow passages, heat transfer area between the fluid and solid is increased. The continuation of the fins through and behind the packed molecular sieve provides additional conduction paths for the heat transfer. The volume of molecular sieve is minimized to minimize the length of the conduction path. Heat transfer area is maximized and conduction path length is minimized. Construction of the system with a good conducting material adds to high heat transfer rates. Some trade-off is required to minimize non-molecular sieve mass. The refrigerant flow to the molecular sieve is controlled by one way valves. The ends of the inlet and outlet are screened to contain the molecular sieve material.

Figure 4 shows a system where the fluid flow passages only pass through the center of the molecular sieve container. In this case the fluid in flow passages has to alternate between hot and cold. This heats and cools the molecular sieve material. Normally the outside of the system would be insulated. Conduction paths shown by arrows lead to both sides of the molecular sieve to maximize heat transfer. Figure 5 shows a schematic with flow passages on both sides of the molecular sieve. This allows hot and cold fluid passages on both sides of the molecular sieve to increase heat transfer area and minimize conduction path lengths. Note the fins or conduction paths extend into the outside flow passages. An alternative is for hot fluid in center passage while the outside cold passage is closed. This is the first configuration built in this study. Then the center hot flow passage is closed and cold outside passage allows cold fluid to flow. If the cold flow is at ambient temperature, the insulation can be eliminated.

To provide continuous compression, a number of basic systems can form the compression system. Figures 6 and 7 show possible groupings of the sub-systems and a basic set of controls to form a total compression system. Figure 6 shows

Figure 4. Compression Sub-System Cross-Section

Figure 5. Compression Sub-system Cross Section

Figure 6. Grouping of Sub-Systems

Figure 7. Grouping of Sub-systems

an end view of three basic sub-systems combined in a circle to form a total compression system. A rotating disk allows fluid access through an opening to one flow passage at a time. The other two systems could be cooling from natural or forced convection over external fins. Since the temperature difference for heating with automobile exhaust is higher than the temperature difference for cooling, the heating requires less time than the cooling. With three compression units, one unit can be heating and two units can be cooling. The fins could be enclosed with a secondary positive controlled cold flow. A system of valves could be used to control the cold flow. Figure 7 shows two sub-systems side by side with positive control of the heat flow through the center passage. The system could be used for car air-conditioning with exhaust gases passing through the system. A plate moves between the two sub-systems on an impulse from a vacuum solenoid. The plate flips between the sub-systems as the vacuum is oscillated between opposite sides of the solenoid. Cycles of a few minutes are a possible sequence. Hot gas is passed through one or the other sub-system flow passages from the exhaust inlet. To maximize input energy from automobile exhaust, the gases after the catalytic converter are utilized when available. The vacuum solenoid requires essentially zero energy input. Except for the shaft the system input gas path is sealed. On the outlet side, the gas moves through a venturi nozzle to pump cool air over the outer fins on the shut off system. One sub-system is heated, one sub-system is cooled to provide essentially continuous flow. A system using forced controlled flow over the external fins is a possibility and was used in the experimental work.

Figure 9 shows a group of sub-systems combined with a hot fluid manifold and a cool fluid manifold. The manifolds enter the end of the compression sub-systems through a fluid switching device. The sub-system can be switched from hot fluid to cool fluid by moving the lever up and down. The spring keeps the system in either the hot or cool phase. Each sub-system has its own switching

Figure 8. Grouping of Sub-Systems

Figure 9. Grouping of Sub-systems with Hot and Cold Manifolds

device. The systems can be controlled by external movement of any kind sequenced in the proper order. For example, one sub-system at a time could be heated for two minutes while the other sub-systems are being cooled. The heating can be moved from one sub-system to another while the other sub-systems are cooled. The type system with two sub-systems was the final system tested.

Figure 10 shows a cross section of the switching device and the end of the sub-system. The switching device has a valve which either closes the hot manifold or cool manifold. Only one manifold or the other can be closed.

Figure 11 is an alternate cross section of the sub-system. The molecular sieve is encased in a thin walled tube. The heat transfer is accomplished by providing numerous flow passages for hot or cool fluid. The numerous passages minimize the thermal conduction path lengths and maximize the area for convection heat transfer. By using all thin wall tubing, the non-productive mass which must be heated and cooled is minimized. The configuration is an alternative to the cooling paths in Figures 4 and 5. The plate must be made conically to eliminate thermal fatigue. The tubes must be curved in the container to prevent large thermal stresses.

Figure 12 shows a schematic of the system in an automobile. The condenser is still in front of the radiator and the evaporator is still located behind the engine at the front of the passenger compartment. Except for the compressor the system is identical to a conventional automobile air-conditioning system. Even Refrigerant 12 can still be used. The compression system behind the catalytic converter could be similar to the systems in Figures 6 to 11. It must be noted that the standard compressor has been eliminated.

An additional possibility is to provide cold storage when the system is off. By closing valves A and B when the air-conditioner is off, the intermittent compression system will cool and be ready to absorb refrigerant instantly as the air-conditioner is started. As a result, instant response to air-conditioning

Figure 10. Switch Device on Sub-system.

Figure 11. Compression Sub-system Cross-Section

Figure 12. Automobile Air-Conditioning System

will result upon turning on the system. Air-conditioning is available before the engine is started. In addition to valves A and B, a small liquid refrigerant reservoir is a requirement. The reservoir is required to hold the liquid refrigerant which is normally absorbed by the molecular sieve material. Longer period and larger quantity cold thermal storage, using the same technique, is possible in larger systems using this compression technique. Sufficient absorbent capacity and sufficient liquid refrigerant storage must be available. Even though this has been determined as a requirement, it was not installed on one of our experimental systems.

Internal automobile heating using the system is a possibility. By opening valve A and closing valve B the hot refrigerant vapor moves directly from the compression system to the evaporator (used as a heat exchanger) and thereby heats the automobile. The response time to heat output when starting a cold car will be much faster with this system using exhaust gas thermal energy than when using engine water. The exhaust gases are hot immediately while the water system temperature lags. This also eliminates the second heat exchanger used for heating in the automobile environmental control system and thereby helps automobile capital economics.

Mounting a refrigeration unit using this compression system and the exhaust gases on a refrigeration truck is a feasible application. Normal truck exhaust gases, especially on a diesel, are at a higher temperature and much more consistent than the exhaust gases from an automobile. This is probably the simplest application from a technical standpoint. For periods when exhaust gases are not available, a plain burner in the exhaust line can provide the thermal energy. An alternative, Figure 2, is to use a standard type compressor in parallel with the compression system during these periods. The new refrigeration system is more economical from a capital and operating standpoint than the present refrigeration system using a small diesel engine.

To summarize, a unique thermal compression system is added to standard liquid-vapor heat pump (air-conditioning/refrigeration) to allow thermal energy use rather than power. The study emphasizes the molecular sieve-refrigerant match, minimum non-producing mass, thermal fatigue and shock, and heat transfer requirements to make the system work technically and economically.

IV. VEHICLE PERFORMANCE WITH AND WITHOUT AIR-CONDITIONING

To evaluate the energy savings with this compression system, performance tests were made with the conventional air-conditioning system. Performance data, miles per gallon, were collected for three different combinations. These were:

1. Air-conditioning on with windows closed,
2. Air-conditioning off with windows closed, and
3. Air-conditioning off with windows open.

These conditions were used for runs for an unloaded vehicle and for a loaded vehicle. The data was taken primarily on interstate highways which are relatively flat but have minor rolling hills. Speeds varied from 45 to 65 mph. Performance was measured by an electronic computer integrated into the automobile system. The energy penalty for air-conditioning is greatest at high speeds and heavy loads. It can reach 100 percent and 20 to 30 percent penalties are not unusual. The data represents comparisons rather than fixed performance numbers.

The normal sequence for taking data consisted of:

- a. Setting the speed control on the truck to a predetermined speed.

Once steady state conditions were achieved, data was taken.

- b. A ten mile stretch was used for each of the conditions. Over this distance, data was taken in approximately 15 second increments. At each time increment, mpg was recorded. The actual speed, distance covered, and temperatures were recorded. Between conditions, for example changing the air-conditioning from on to off, a few miles were allowed for the system to settle into steady state. At the end of the run an average performance (mpg) was calculated for each of the conditions.

To assure reasonable comparisons, continuous runs (normally around 50 miles) were made for each set of data. During this period, no changes were made in the speed control and the entire run was made in one direction. The identical direction was used to minimize the effects of wind. To try and match terrain for each run, relatively level stretches were used for the tests. Most runs were made on I-59 between Tuscaloosa, Alabama and the Mississippi border and traveling east and west on I-75 in the southern part of Alabama and the northern part of Florida. A few other interstates and even some little used minor highways served as test tracks. Even under these conditions rolling hills were involved in all the data. Fifty mile continuous level stretches do not exist. If sufficient interference from other traffic occurred, runs were aborted or cut short. A number of runs have less than ten miles and others have only the first conditions, air-conditioning on and off with windows closed.

With the new system, configurations were kept as close to conditions as possible. With air-conditioning on, the system was at maximum cooling with maximum blower speed. With the air-conditioning off, the system used outside air at maximum blower speed with windows open and with windows closed. The major difference is the work to run the compressor. The energy saved with the new system is the energy saved by removing the conventional compressor.

The computer used to acquire data is Electronic Computer, Model No. 15-7260W from J. C. Whitney and Company in Chicago. The various data are acquired and calibrated as follows:

- a. Speed is measured by placing two magnets on the drive shaft and sensing pulses. The system is calibrated by stopping at a mileage marker on an interstate, setting the unit for calibration, driving ten miles and inserting the ten miles. This was done at the beginning of each series of runs. The system held the calibration accurately and, on continuous checks with mileage markers, the error was always less than two percent.

b. The fuel flow calibration is made by filling the tank, driving 125 to 175 miles, filling the tank again and placing the number in the computer. The primary problem is filling the tank to exactly the same level and variations in gasoline density due to temperature. Accuracy in calibration is about 4 parts (4/10 of a gallon) in 150 parts (150 tenths in fifteen gallons). This is less than a three percent error. The major problem was the calibration usually was lost after a few hundred miles and 125 plus miles were required to recalibrate. The loss normally occurred as a step function so it could easily be detected.

The gas flow is measured by a flow meter in the fuel line to the carburetor. This means that to get accurate instant mpg, the entire system must be at steady state. The recorders were very conscious of this and did not take data during fluctuations. The average gpm are not effected by these variations as they are based on gas used in the run distance.

The calibrations are lost when power from the battery is interrupted. This did occur a few times and losses amount to about 150 miles of test data. Altogether good data is available on only 30 percent of the miles driven. Downhill gives very high mileage while uphill gives extremely low mileage. This occurs when the second two barrels in the carburetor open. Normally only two of the four barrels are utilized.

Summaries of the results are presented in Tables 1 through 3. All data is for relatively high speeds on the open highway. The data in Table 1 and Table 2 are for a heavily loaded truck. An eleven foot slide-in camper was on the vehicle. As a result, the data is for the truck with the engine loaded. The data in Table 3 is for an empty truck.

The ratios given are the mpg for the air-conditioner operating divided into the mpg with the windows closed and the air-conditioner off. The numbers note that over 25 percent in energy can be saved on the open road when air-conditioning

TABLE 1. Performance Run Summary*

Run No.	Miles	MPH	A/C	Avg. MPG	Off/On Ratio	T _{outside} °F	T _{AC} °F	Comments
1.	10	52	On	8.5	1.38	79	43	
1a.	10	52	Off	11.7		81		
2.	10	46	Off	13.9		81		
2a.	10	46	On	14.3	0.97	79	41	
3.	10	55	Off	7.8		79		
3a.	10	55	On	6.9	1.13	81	43	
4.	10	59	Off	5.2		82		
4a.	10	59	On	3.8	1.37	81	42	
5.	10	55	Off	9.2		84		Carburetor Leaned
5a.	10	55	On	5.3	1.74	83	43	
6.	10	59	Off	4.9		84		
6a.	10	59	On	3.5	1.40	84	43	
7.	10	50	Off	12.7		86		
7a.	10	50	On	7.8	1.63	86	41	
8.	10	51	Off	12.5		84		
8a.	10	51	On	9.5	1.32	82	43	
9.	10	55	Off	10.8		82		
9a.	10	55	On	6.1	1.77	84	42	
10.	5	58	Off	7.7				
10a.	6	58	On	6.1	1.26			

*March 12, 1981, running at 8600#, camper on truck

TABLE 2. Performance Run Summary*

Run No.	Miles	MPH	A/C	Avg. MPG	Off/On Ratio	T _{outside} °F	T _{AC} °F	Comments
11.	10	54	On	3.8	0.95	79		Head Wind
11a.	10	54	Off	3.6		79		Head Wind
12a.	10	55.6	Off	5.5		79		Windows Open
12b.	10	55.5	Off	7.4		81		Windows Closed
12c.	10	54.3	On	4.1	1.8	81	43	
13a.	10	51.4	Off	10.4		84		Windows Open
13b.	10	51.5	Off	11.7		84		Windows Closed
13c.	10	51.1	On	7.6	1.54	81	44	
14a.	10	50.1	Off	9.9		86		Rough Asphalt Road
14b.	10	52.4	Off	12.0		88		Back Window Open
14c.	5	52.3	Off	9.5		86		
14d.	10	51.3	Off	10.6		83		Windows Closed, Asphalt
14e.	10	51.8	Off	10.2		76		Windows Closed
14f.	8.5	51.8	Off	9.2		84		Windows Open
14g.	10	49.8	On	7.5	1.36	84	43	

*March 14, 1981, running at 8600#, Camper on Truck

TABLE 3. Performance Run Summary*

Run No.	Miles	Avg. MPH	A/C	Avg. MPG	Off/On Ratio	T _{outside} °F	T _{AC} °F	Comments
16a.	10	45.3	Off	16.1		77		Windows Open
16b.	10	45.2	Off	15.1		77		Windows Closed
16c.	10	45.0	On	14.5	1.04	77		
17a.	10	56.0	Off	11.3		77		Windows Closed
17b.	10	55.9	On	6.8	1.67	77		
17c.	10	53.9	Off	12.4		77		Windows Open
18a.	10	50.1	Off	17.6				Windows Open, Tail Wind
18b.	10	50.0	Off	17.2				Windows Closed, Tail Wind
18c.	10	50.0	On	14.9	1.15	83	44	
19a.	10	60.6	Off	12.1				Hilly
19b.	10	60.5	On	7.3	1.66			Hilly
20a.	10	60.1	Off	10.1				
20b.	10	58.9	Off	9.8				Windows Open
20c.	10	60.6	On	6.5	1.51			
21a.	10	55.1	Off	15.0				
21b.	10	54.9	On	11.3	1.33	74	45	
22a.	10	59.4	Off	11.4				Windows Open
22b.	10	59.9	Off	13.2				Windows Closed
22c.	10	59.8	On	9.9	1.33			
23a.	5	63.6	On	7.0	1.63			
23b.	5	63.4	Off	11.4				

*March 15, 1981, running at 5100#

would be operating. This represents over ten percent of the gasoline consumed for automobiles and light trucks.

Tables 4 through 9 present the variations in gas mileage versus time. The data show the fluctuations due primarily to hills, even minor slopes, and wind changes. Even though the most level roads in this part of the country were chosen, appreciable fluctuation exists. Without the speed control, one slows for upgrades and speeds up for downgrades, eliminating some of the fluctuations. The speed control does not necessarily save fuel, especially if the engine is loaded. It should be noted the averages of these numbers do not necessarily equal the average miles per gallon.

Conclusions

This study shows that appreciable energy savings are available if the conventional compressor power is eliminated. Savings of 20 to 60 percent are available depending on vehicle loading. The devices presently on the market, which shut off air-conditioning under full load to save fuel, agree with this conclusion. This means as gasoline engines get smaller and with diesel engines just matching minimum car propulsion requirements, the savings could be 40 to 60 percent during air-conditioning use. The system developed in this project has real potential for major energy savings.

TABLE 4. Performance Distribution During Run
Run 9a, 55 MPH, 6.0 MPG Avg., A/C On
15 Second Increments

MPG	Times	MPG	Times
2	2	10	7
3	1	11	3
4	2	12	4
5	4	13	1
6	3	14	0
7	1	15	1
8	5	16	0
9	5	17	0

TABLE 5. Performance Distribution During Run
Run 13b, 51.5 MPH, 11.7 MPG Avg., A/C Off
15 Second Increments

MPG	Times	MPG	Times
7	1	13	16
8	1	14	4
9	1	15	1
10	1	16	1
11	7	17	0
12	10	18	1

TABLE 6. Performance Distribution During Run
Run 16c, 45.0 MPH, 14.5 MPG Avg., A/C On
15 Second Increments

MPG	Times	MPG	Times
10	1	19	3
11	0	20	1
12	5	21	1
13	4	22	1
14	9	23	0
15	8	24	0
16	13	25	1
17	2	26	0
18	1	29	1

TABLE 7. Performance Distribution During Run
Run 19b, 60.5 MPH, 7.3 MPG Avg., A/C On
15 Second Increments

MPG	Times	MPG	Times
3	1	11	4
4	2	12	3
5	4	13	2
6	2	14	3
7	2	15	3
8	3	16	2
9	1	17	2
10	2	23	1

TABLE 8. Performance Distribution During Run
Run 20a, 60.1 MPH, 10.1 MPG Avg., A/C Off
15 Second Increments
Hilly

MPG	Times	MPG	Times
2	1	13	4
5	2	14	2
6	2	15	4
7	2	16	0
8	1	17	2
9	3	18	1
10	3	19	1
11	3	20	1
12	5	23, 24, 27	1 each

TABLE 9. Performance Distribution During Run
Run 22a, 59.8 MPH, 9.9 MPG Avg., A/C On
15 Second Increments

MPG	Times	MPG	Times
3	1	13	1
4	1	14	6
5	1	15	1
6	0	16	1
7	0	17	2
8	5	18	1
9	3	19	0
10	5	21	1
11	2	23	1
12	3	24	1

V. GENERAL TECHNICAL STUDIES

A number of technical areas were studied theoretically to determine the design parameters for the thermal compression system. Summaries of these studies follow.

Sizing

It was determined that a minimum air-conditioning capacity of 6000 B/hr ($\frac{1}{2}$ ton) would be required. Conventional automobile units have a capacity directly related to motor RPM. At lower RPM, the output is less than $\frac{1}{2}$ ton but at the higher RPM, many systems put out over 3 tons. The project director has an eleven foot camper with a 5000 B/hr air-conditioner which works on 120 volts. This system can freeze out four people (when there is no cooking) under almost any weather conditions. To establish the 6000 B/hr with Refrigerant 12, a flow rate of 120 lbm/hr with a cooling energy availability of 50 B/lbm of refrigerant is required. If the compression units are cycled ten times an hour, 12 lbm are required per cycle. Using three units, this is 4 lbm per unit per cycle. If 30 percent of molecular sieve weight is available in refrigerant per cycle, the individual compression units must contain about 13.3 lbm of molecular sieve. Pages of sizing calculations have been made with the general result that the individual unit size should be in the ballpark of six inches in diameter and three to four feet long. All units tested on the truck have been six inches in diameter and three feet long. Optimum size will have to be determined from accumulated data.

Refrigerant and Molecular Sieve Combinations

A general study was made to determine the best refrigerant and molecular sieve combination. The refrigerant Table 10 has been prepared to cover truck

TABLE 10. Possible Refrigerants and Their Properties

Refrigerant	Mol wt.	At -20°F		At 50°F		At 120°F		Safety ** Level
		Psia	h_{fg}^*	Psia	h_{fg}^*	Psia	h_{fg}	
11 CCl_3F	137	1.42	85	8.78	79.7	32	73.3	6
<u>12</u> CCl_2F_2	120	15.2	70.9	61.4	62.9	172	52.5	6
13 $CClF_3$	104	126	50	390	30.9			6
<u>22</u> $CHClF_2$	86.5	24.8	97	107	83.6	274	67	5a
113 $CCl_2F-CClF_2$	187.4	.43	72	3.9	67.5	15.4	62.9	4-5
114 $C_2Cl_2F_4$	170.9	3.4	62.8	20.5	57.2	62.7	50.7	5
717 NH_3		18.3	583	89.19	527	286	455	2
13B1 $CBrF_3$	148	48	46.6	161	38.5	400	24.7	
23 CHF_3	70	152	82.6	471	500			6
115 CF_3CF_2Cl	154	22.7	51.6	87.0	43.5	237	39.5	6
142b CH_3CClF_2	100.5	6.5	94.6	30.2	89	96.8	78.8	5a
<u>152a</u> CH_3CHF_2	66.05	12.4	137.6	54.4	127.9	166	108.9	6
216 $C_3Cl_2F_6$	220.9	.713	60.5	5.4	54.3	22.8	38.2	6
C318 C_3F_8	200	5.08	53.0	27.2	47.5	92	39.4	6

*Units in B/lbm

**Underwriters' Laboratories Classification of Comparative Hazard to Life of Gases and Vapors

refrigeration through automobile air conditioning. A consultant was hired to determine the best molecular sieve.

Generally, the lower molecular weight refrigerants are expected to give the best performance based on percent absorption by the molecular sieve, latent heat of vaporization, and molecule size to pass through the pores in the molecular sieve.

Based on availability, cost, present use and molecule size, actual experimental studies have been limited to Refrigerant 12 (presently used in all automobiles) and Refrigerant 22 (used in most home air-conditioning). The vast majority of testing used Refrigerant 12 in order to match the conventional system. Refrigerant 22 is expected to give better performance.

Ten to fifteen types of molecular sieves were considered. Linde molecular sieve 13x was chosen due to it's ability to absorb the Refrigerant 12 molecule, high temperature properties, and availability. The sieve gave better performance according to the acquired test data or specifications by Linde representatives. The major problem was a cost of about three times that shown on one year old price sheets. The increased cost will play an important roll in the final system cost.

Along this line, major emphasis has been given to the weight of refrigerant absorbed per unit mass of molecular sieve. All tests (from laboratory unit tests to full scale tests on the truck) measured this ratio.

Heat Transfer

With the rapid cycling of the compression units, the heat transfer becomes a critical factor. Numerous general calculations were made. It became obvious that the conduction paths through the molecular sieve material were insufficient. To correct this, studies of additional paths in the form of heat conducting ribs were made (Figure 4). The first units were tested with the ribs and questionable results were obtained. The heat flow path lengths were reduced by distribution

of the hot and cold gases throughout the molecular sieve material through a series of parallel tubes (Figures 10 and 11). Results indicate this configuration is in the ballpark. The next generation of compression systems could use tubes with internal and external fins of high conducting material. These are commercially available. They could increase heat transfer rates by as much as a factor of five. Due to the involved geometrics and numerous modes of heat transfer, the only accurate evaluation is in full scale tests.

Controls

The controls for the conventional air-conditioning system are not changed. However, the controls for the new compression unit are involved and new concepts are required. A number of mechanical switching systems for switching the hot and cold gases were analyzed. These included rotating, sliding, and flip-flop systems. The last two were used in full scale testing. It was determined that the actual control of the mechanical systems should be with pneumatic actuators operating off the truck vacuum system.

In addition to looking for innovative concepts ourselves, a contract was made with the Mechanical Engineering Design Clinic at The University of Alabama. A design team of three senior mechanical engineering students spent over 300 hours looking at the system. No new innovative concepts were discovered. The project ended as a study of the proposed controls systems.

Other Applications

A small amount of work was done to apply the system to truck refrigeration and solar systems. Feasibility looks good at this time.

VI. LABORATORY TESTS

A number of basic tests were made using laboratory equipment. A system was designed and built to determine the basic relationships between the refrigerants and molecular sieve. Due to budget cuts, this system was scaled down but still gave some reasonable results.

The system is shown in Figure 13. It is designed to hold molecular sieve material while refrigerant is added or removed in known quantities to test absorption and regeneration. During these processes, pressure, temperature and energy change were monitored. Some problems existed due to the scale of the unit. The pressure gage, for example, became a major heat transfer surface and storage chamber for condensed refrigerant. Still, many hours of data were taken with the unit and some useful results were recorded.

The results acquired include:

1. Absorption mass of refrigerant as a function of molecular sieve mass,
2. Temperature increase and energy rejection which results from the absorption process,
3. Demonstration of the possibility of molecular sieve contamination,
4. Percentage of refrigerant which can be regenerated,
5. Problems of heat transfer within the molecular sieve, and
6. Verification that desired temperatures were available using molecular sieve Linde 13x absorption.

In general, the technical concept of the system was verified. The problem of proving the practical and economic concepts remained.

Approximately 150 hours of actual run time were logged using the laboratory system. Approximately 40 hours consisted of calibrating the energy losses of

W.J. Schaetzle & Associates, Inc.

Plug Detail
Hard Soldered to Tube

Note: All Joints Hard Soldered

Figure 13. Basic Experimental Laboratory System

the system with temperature. This was done by filling the system with water and determining the equilibrium temperature for each power input. It was assumed that the heat loss would equal the input during regular runs. A few summaries of significant runs follow.

Summary 1 - December 18 to December 21, 1980

The system was filled using Linde Molecular Sieve 13x. The weight of the molecular sieve was 216 gm. The system was connected to a Refrigerant 12 bottle and gas inserted quickly to 63 Psig. The valve was closed and within 3 minutes the pressure fell to 30 Psig. This vacuum corresponds to a temperature of 33°F in a refrigeration cycle. The molecular sieve temperature rose from 70°F to 116°F from the energy liberated in the absorption process. Temperatures shown are thermometer temperatures. These are well below actual temperatures. Actual temperatures are probably in the 140°F to 180°F range. The pressure was again raised to 63 Psig and the valve closed. The system went to 54 Psig in five minutes. After a few hours the pressure decreased to 48 Psig which represents 52°F in a refrigeration system. Overall, only 41 gm of refrigerant was absorbed representing 41/216 or 19 percent of the sieve mass. This could be sufficient for an actual system but was disappointing. The system did absorb a reasonable amount of refrigerant to the pressure (corresponding saturation temperature range) required and the absorption energy was noticeable.

On December 21, 1980, energy was added to the system with temperature rising to 192°F. Over 33 gm of refrigerant was released. This verified the regeneration phase and was encouraging.

Summary 2 - December 23, 1980

The system had been heated for degassing and was under vacuum for 24 hours. Refrigerant 12 was added to the system. The following sequence of events occurred:

On injection pressure rose to 55 Psig

After 1 minute pressure decreased to 30 Psig or 33°F saturation temperature

The sieve material showed a rise in temperature to 94°F on thermometer, however, temperature must be over 125°F

Recharged to 55 Psig

Pressure down to 38 Psig in one minute, sieve temperature rising

Recharged to 55 Psig (saturation pressure of charging tank)

Pressure down to 34 Psig (37°F) in 2 minutes

Recharged to 55 Psig

15 seconds to 48 Psig

1 minute to 43 Psig

2 minutes to 40 Psig (44°F)

3 minutes to 39 Psig

Recharged to 55 Psig

15 seconds to 49 Psig

1 minute to 45 Psig

2 minutes to 43 Psig

3 minutes to 42 Psig

5 minutes to 39 Psig (42°F)

2 hours to 39 Psig (18°F)

The results indicate the possibility of rapid absorption; however, only 28 gm of refrigerant were added. Sixty percent of refrigerant was absorbed in 12 minutes and 75 percent in two hours. The percentage of refrigerant mass absorbed is 28/216 or a disappointing 13 percent. Based on this, about eight percent could be oscillated in the cooling and heating process. Absorption rate was too low.

Summary 3 - December 28, 1980

Due to water and flux which had been in the system, it was decided to change the molecular sieve material. The change was made and Refrigerant 12 was added as rapidly as possible. In thirty minutes, Refrigerant 12 equal to 43 percent of molecular sieve weight had been added. This was the best result to date and very encouraging. It indicates regular cycling of 30 percent of molecular sieve weight could possibly be accomplished in 20 minutes. This is more than sufficient to make a practical system for automobile air-conditioning. This turned out to be the last run on the laboratory system. More runs were planned; however, a full scale compression unit to be installed on the truck was being designed so all subsequent work was done on such units. Results on full size units verify the results of this run.

Conclusions and Recommendations

The laboratory tests verified the technical feasibility of this project. They also verified that in order to acquire good qualitative and quantitative data, a larger more elaborate system would be required.

VII. COMPRESSION UNIT TESTS

A developing series of tests with compression units using exhaust gases for thermal energy were designed, fabricated, installed and evaluated. Five different units were fabricated and four of these were evaluated on the truck. Problems on each unit were analyzed and corrected in the next stage. After the fifth unit, a second identical unit was fabricated and the pair of matched units were installed on the truck. The conventional compressor was disconnected and the new compression units incorporated into the truck air-conditioning system. As with the laboratory tests, the technical feasibility has been verified with these units. All indications are that the final units are practical. Air-conditioning has been available but not to acceptable standards. Existing problems center on controls, mainly the effective cycling of hot and cool gases through the units.

The sketch of the first unit is shown in Figure 14. The unit consisted of concentric 3 inch and 6 inch pipes. The volume between the pipes was filled with molecular sieve material. Steel ribs were designed into the length of the system to give the required heat transfer. Around the outside of the outer ribs, sheet metal was wound and fastened with clamps. A flexible steel hose from the exhaust to a sliding plate provided hot gases. A lever to the truck cab slid the plate back and forth. The heating could be either on or off. A blower (used Volkswagen defroster blower) was connected to the space between the sheet metal and 6 inch pipe. Cooling was provided by turning the blower on. Two refrigeration hose connections were welded to the top of the six inch pipe. One was connected to a Refrigerant 12 bottle and one was connected to a pressure gage. Three thermo-

Figure 14. First Compression System Sketch

couple wells entered the molecular sieve chamber. One measured exhaust gas temperatures into the unit.

Problems started with construction. Originally the pipe was to be cut and welded to the ribs; however, pipe buckling eliminated this option. The inner ribs were welded together and inserted by pressure into the inner pipe. The ribs in the chamber were tacked to the inner pipe and the outer pipe pressure fitted over the ribs. The outer ribs were also tacked to the outer pipe and the sheet metal stretched over these ribs. This was a major compromise on the heat transfer. Ends were welded to the pipes for sealing. A $\frac{1}{2}$ inch plug on one end allowed the filling of the unit with molecular sieve.

Due to the construction problems, the heat transfer capability was compromised. The ribs created problems in the filling process, restricting the flow of molecular sieve material (1/8 inch pellets). The cutting oil, welding flux, and other welding residue, etc. caused internal contamination.

Upon completion, the unit was filled with refrigeration cleaning fluid and allowed to stand for a few days. It was rinsed a number of times with the cleaning fluid. Then the chamber was pumped and held at 30 inches of Hg vacuum for a couple of days. The system held the vacuum (30 inches of Hg) without pressure increase for 24 hours.

The system was filled with molecular sieve on March 21, 1981. The system absorbed 6287 grams (13.68 pounds) of molecular sieve. It was pumped for 24 hours, and then held a 30 inch Hg vacuum for 24 hours.

The system was charged on April 8, 1981. A Freon 12 cylinder was connected to the compression system and the resulting data is shown in Summary 4.

Summary 4 Charging Data Summary

Time	Cylinder Pressure	Unit Pressure/ Vacuum	Sat Temp °F	Cylinder Wt.
0	40 Psia	30 inches/		16 lb 12 oz
2	30	28/-152		15 lb 12 oz
4	20	27/-100		15 lb 4 oz
6	18	26/-92		14 lb 15 oz
8	15	24/-80		14 lb 9 oz
10	12	20/-63		14 lb 2 oz
12	10	17/-54		13 lb 15 oz
14	9	12/-42		13 lb 15 oz
16	9	5/-29		13 lb 11 oz
18	8	4/-11		13 lb 7 oz
20	10	8.5/-2		13 lb 5 oz
22	11	9/0		13 lb 3 oz
24	11	10/2		13 lb 1 oz
26	11	10/2		13 lb 0 oz
28	11	10/2		12 lb 15 oz
30	11	11/4		12 lb 15 oz
32	14	12/5		12 lb 15 oz
35	14	13/7		12 lb 14 oz
37	15	14/9		12 lb 13 oz
47	22	22/21		12 lb 11 oz
52	28	26/26		12 lb 11 oz
Heat Lamp on Refrigerant Cylinder				
55	30	29/28		12 lb 9 oz
59	35	34/37		12 lb 6 oz

Summary 4 Continued Charging Data Summary

Time	Cylinder Pressure	Unit Pressure/ Vacuum	Sat Temp °F	Cylinder Wt.
1 hr 2 min	39	38/41		12 lb 4 oz
1 hr 15 min	44	43/46		12 lb 3 oz
1 hr 25 min	51	50/53		12 lb 1 oz
1 hr 35 min	54	53/56		12 lb 0 oz
4 hr 10 min	69	68/68		11 lb 8 oz

The rapid absorption of refrigerant at the start of charging was very encouraging. It was obvious the flow lines were limiting the flow rates. The related saturation temperatures for the pressures were noted. This assures the system can be used for refrigeration trucks where a -20°F must be maintained. The charging rate was also limited by the heat transfer into the refrigerant cylinder. It was covered with frost within four minutes and remained covered for fifty minutes. Eventually a heat lamp was allowed to heat the cylinder and the pressures rose. Altogether 5.25 lbm of refrigerant was absorbed by 13.68 lbm of molecular sieve for a 38 percent mass ratio. This would be satisfying for a practical unit.

If one assumes 5 lbm of refrigerant is compressed every 15 minutes in each of three parallel compression units, then 45 lbm are compressed per hour. At 60 B/lbm this gives 3600 B/hr. This value exceeds many automobile air-conditioners in city traffic. By increasing the outer tube diameter from 6 inches to 8 inches, the capacity could be doubled. These numbers show a reasonable size system can be constructed.

The system was installed on the truck and testing was initiated on April 9, 1981. The temperature readings were useless as the response time and conduction through the wells gave erroneous readings. Upon putting exhaust gases through the compression unit, the pressure rose to 182 Psig in ten minutes. The refrigerant

erant cylinder was connected to the unit and was expected to absorb the heated refrigerant. However, it heated very rapidly and it's saturation temperature reached saturation pressure quickly. For safety reasons the system was shut down and the excess pressure bled to the atmosphere. After cooling started, it took 30 minutes for the pressure to drop to 150 Psig. It became obvious that the cooling system was totally inadequate. Some changes were made; however, the cooling problem was never solved. Insulation helped heating but had very little effect on cooling. A few days later, in the heating cycle, the pressure exceeded 450 Psig and burst a hose. A desperate stop and retreat were required to escape a cab filled with a poisonous gas. The system was recycled in short cycles for more than a month and it always regenerated and refrigerant was bled to the atmosphere. The regeneration on a continuous cycle completed the technical feasibility verification. In spite of the problems, the system continued to absorb refrigerant.

Conclusions from the first full scale compression system showed the need for increased heat transfer. A major change in the method of cooling was also required.

A new design was initiated near the middle of April using the cross-section shown in Figures 10 and 11. This distributes the heating and cooling gases throughout the molecular sieve material. To maximize the heat transfer in this unit, it was decided to use hard copper throughout the system even though the cost was high. The small tubes for airflow were quickly located. The large outside tube would have taken months for delivery. As a result, a steel outer tube was utilized. The differences in expansion rate of the dissimilar metals never allowed the unit to be completed. After welding, joints cracked continuously as they cooled. The unit was junked before testing.

The same design using six 3/4 inch inner tubes and a 6 inch outer tube was constructed totally from steel. The unit checked out during leak tests and

was filled with molecular sieve. Once mounted on the truck, the pumping tests failed but not by a large amount. All joints were checked and rechecked; however, pumping tests still failed by a small amount. It was assumed debris and oil were slowly evaporating so hot gases were passed through the unit for over 2500 miles. The pumping tests failed to improve and replaced connections failed to help. In desperation, refrigerant was injected anyway. Leaks were visible all over the unit. Removal showed thermal fatigue cracks had appeared at joints over the entire unit. Pipes had cracked at connections to the end plates. End plates had cracked from the outer tube. The cooling and heating had had a terrible effect.

A few design changes were incorporated into the unit. The tubes were given a U-bend into their centers. Thermal expansion or contraction would now create a small force expanding or contracting the U-bend as a spring for a very short distance. Stress would be relatively small. The end plates were made an oval shape. Temperature changes caused a shape change but only very minor stresses.

A unit was fabricated, leak tested and installed on the truck. Approximately 1000 miles of testing including heating and cooling produced zero cracks.

A second unit identical to this unit was fabricated, leak tested and installed on the truck. This unit was filled with Linde 13x molecular sieve weighing 9.33 lbm from a new sealed container on September 24, 1981. It was leak tested and mounted on the truck. The unit absorbed 5.6 lbm of Refrigerant 12 on September 29, 1981 for a mass ratio of 0.60. Clean molecular sieve material helped. The explanation of the high absorption as given by the Linde vendor and verified by our consultant is that under ideal conditions, pressure and temperature, a layer of liquid forms on the sieve surface. This also explains large variations in absorption data in the Literature.

Based on this result and others, a mass ratio of 0.3 (0.3 lbm of refrigerant and per lbm of molecular sieve) can be considered a reasonable value for the refrigerant to be compressed each cycle.

The present and final configuration for this study is identical to Figure 9 except that only two sub-systems are utilized. The switching system is also identical to the ones in Figure 9. A more detailed view of the switching system is shown in Figure 10.

Upon installation, a mechanical switching system connected to the levers alternated exhaust gases and cool air to each compression sub-system. The system has been changed to pneumatic actuators connected to each level as shown in Figure 9. Leakage of exhaust occurs around the valves, especially in the cooling configuration.

A ram type inlet has been installed on the cold manifold to provide cold air for testing on the open highway (Figure 15). This has not worked effectively.

During the first week of October 1981, the compression units were incorporated into the truck air-conditioning system. The system has provided a 20 Psi pressure difference between the cold side and the hot side of the air-conditioning cycle. In order to have effective air-conditioning, a minimum pressure difference of 60 Psi is required. Automobile air-conditioning units require a capability of 130 to 150 psi pressure difference. This system falls short of the requirements for effective air-conditioning at this time.

The problem results from the method of switching gases. The compression units minimum temperature has been measured at about 275°F rather than the 120°F to 150°F anticipated. This problem is being corrected.

A second minor problem has occurred in the connections to the main air-conditioning system. Since the engine is mounted on rubber mounts, the exhaust system has a continuous movement relative to the car body. The units were connected with copper tubing to the conventional air-conditioner. Fatigue breaks have occurred regularly in connecting joints. This can be solved by using a length of rubber hose between the compression units and the car body. This is done in a conventional system between the compressor, which is mounted

Figure 15. Ram Air Inlet to Manifold

on the engine, and other components mounted on the car body. Since these are special hoses and require high temperature material in this case, broken joints have been repaired as they occur.

At the time of this report, the two final units have been in place for over 20,000 miles. From all indications, the thermal fatigue problems have been solved.

For the last year, work has continued slowly using the limited resources of W. J. Schaetzle & Associates, Inc. It is hoped an effective system will be operating in the very near future. It is anticipated the system will be demonstrated for a number of parties at the annual SAE meeting in February.

A set of large colored drawings has been completed to explain the concepts and the applications of the thermal compression system. These will be used to actively promote the system.

VIII. ADDITIONAL WORK

Additional work is being continued at the expense of W. J. Schaetzle & Associates, Inc. The work includes:

A. Solving the control problems for changing the hot and cool air flow. Pneumatic actuators have been purchased and mounted. Slightly stronger units are probably required. Matching springs with the present system is expected to help. A new air blower has been purchased and will be mounted in the near future.

B. Technical papers will be written on the vehicle performance study and on the overall air-conditioning system. These are expected to be submitted in the next few months for the 1984 annual SAE meeting.

C. Additional support is being sought to continue the project on a larger scale. Some interest has been shown and additional support will be sought at the 1983 annual SAE meeting in February.

Each system phase has been shown to work effectively. Effective operation of the total system is a matter of some additional effort.

IX. CONCLUSIONS

Based on the work completed on this project, the following conclusions are made.

A. Eliminating the work into the conventional compressor in automobile air-conditioning will save 20 to 70 percent of the energy used to operate the vehicle during periods when air -conditioning is used.

B. The molecular sieve material will absorb refrigerants to sufficiently low pressures to provide air-conditioning and refrigeration to -20°F.

C. At below exhaust gas temperatures, the system will regenerate the refrigerant at a sufficiently high pressure to provide the required compression. Even though a catalytic converter is utilized, the temperature increase and energy increase provided are not required for operation of the compression unit.

D. A mass ratio of refrigerant absorbed and rejected relative to molecular sieve weight of 0.3 is reasonable with the Linde 13x, Refrigerant 12 combination.

E. The thermal fatigue problems can be overcome with proper design. More than one year of operation on two units, with over 20,000 miles, has shown no fatigue problems in the final units.

F. Based on the above numbers, a practical and economically feasible system can be developed.

G. If the systems are built anywhere in the United States, Alabama will have economic gain since the major plant producing molecular sieve is located in Mobile, Alabama. Local research, development and production of units is a possibility.

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APPENDIX I**TRUCK USED FOR EXPERIMENTS**

A 1979 truck was purchased to evaluate the Automobile Air-Conditioning Unit. A truck rather than an automobile is used because the experimental apparatus is mounted under the body in the vicinity of the exhaust system. A truck has the required space for the equipment under the truck bed. This truck is used for all experimental work, the new experimental air-conditioning system and mileage performance with conventional air-conditioning. The 1979 model was purchased new in June 1980 due to a large discount price on the old model year and an unusually well equipped truck. Since the truck is used in performance testing the specifications are as follows:

- 1979 149 inch wheelbase club cab S/L pickup
- Rear axle ratio 4.10 with anti spin differential
- Front axle capacity 3300 lb.
- 8100 lb. gross vehicle weight package plus auxillary rear springs rated at 585 lbm
- 360 CID 8 cylinder 4 BBl engine
- Automatic transmission with auxillary transmission oil cooler
- Emission reduction control system (used for experiments but not required as truck is over 8500 lbm GVW)
- 117 amp alternator
- Power steering and brakes
- 16.5 x 6.75 heavy duty wheels with 9.50 x 16.5 10 ply rating steel belted radial tires
- Heavy duty shock absorbers front and rear
- Stabilizer bar on front only
- Conventional air-conditioning system
- Automatic speed control (used with all performance tests)
- Additional instrumentation included:
 - Electric tachometer
 - Vacuum guage
 - Volt meter
- Radio AM/CB transceiver
- Extended outside mirrors
- Two gasoline tanks (18 gal. and 21 gal.)
- Other equipment not effecting performance

Unloaded vehicle weight around 5100 lbm

Loaded vehicle weight around 8600 lbm

Weights represent weighings with each combination

Actual weight varies with number of passengers, gasoline, etc.

Variation (-300 lbm +500 lbm)