LIQUID NITROGEN PROPULSION SYSTEMS FOR AUTOMOTIVE APPLICATIONS: CALCULATION OF THE MECHANICAL EFFICIENCY OF A DUAL, DOUBLE-ACTING PISTON PROPULSION SYSTEM

Thomas B. North, B.B.A., M.B.A.

Thesis Prepared for the Degree of

MASTER OF SCIENCE

UNIVERSITY OF NORTH TEXAS

May 2008

APPROVED:

Mitty C. Plummer, Major Professor
Philip Foster, Committee Member
Seifollah Nazrazadani, Committee Member
Nourredine Boubekri, Chair of the Department of Engineering Technology
Oscar García, Dean of the College of Engineering
Sandra L. Terrell, Dean of the Robert B. Toulouse School of Graduate Studies

North, Thomas B., Liquid Nitrogen Propulsion Systems for Automotive Applications: Calculation of the Mechanical Efficiency of a Dual, Double-Acting Piston Propulsion System. Master of Science (Engineering Technology), May 2008, 39 pp., 4 tables, 15 illustrations, references, 19 titles.

A dual, double-acting propulsion system is analyzed to determine how efficiently it can convert the potential energy available from liquid nitrogen into useful work. The two double-acting pistons (high- and low-pressure) were analyzed by using a Matlab-Simulink computer simulation to determine their respective mechanical efficiencies. The flow circuit for the entire system was analyzed by using flow circuit analysis software to determine pressure losses throughout the system at the required mass flow rates. The results of the piston simulation indicate that the two pistons analyzed are very efficient at transferring energy into useful work. The flow circuit analysis shows that the system can adequately maintain the mass flow rate requirements of the pistons but also identifies components that have a significant impact on the performance of the system. The results of the analysis indicate that the nitrogen propulsion system meets the intended goals of its designers.

ACKNOWLEDGEMENTS

First and foremost, I would like to thank my major professor, Dr. Mitty C. Plummer, for his patience, guidance, wisdom, and wealth of knowledge, which have contributed immensely toward my research work. My sincere gratitude goes to Dr. Philip Foster and Dr. Seifollah Nazrazadani for being my thesis committee members and for giving their valuable time to guide me and to review my thesis.

I would also like to thank Dr. Igor N. Kudryavtsev for his assistance with the MATLAB-Simulink computer simulation and Mr. Jerry Davis for his assistance with the flow circuit analysis software.

I am also indebted to my parents, family, and friends for their constant encouragement, patience, and moral support throughout my educational endeavors.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

CHAPTER 1

INTRODUCTION

As the world progresses through the 21st century, it is faced with increasing fuel prices, tougher emissions regulations, and a push for renewable energy sources. These consequences are evident in the increased availability of ultra-low emissions gas vehicles and gas-electric hybrids. The main flaw of both of these advancements is that they are still dependent on gasoline and that they produce hazardous emissions, even though in smaller amounts. Alternative fuel research has also increased greatly, focused largely on hydrogen and fuel-cell technology [18].

Liquid nitrogen is one possible alternative energy carrier, because it can be cheaply produced, is non-flammable, produces only the emission of nitrogen back into the environment, and is renewable. Heat exchangers convert liquid nitrogen into gas up to ambient temperature, and also produce the needed pressure to power a propulsion system. This heating is done by the atmosphere without any additional heat sources, resulting in a simple, reliable, and potentially effective propulsion system [14, 15, 16, 17, and 18].

Background of the Problem and Statement of Need

There are questions regarding limitations in the mechanical efficiency of liquid nitrogen propulsions systems as viable alternatives to hydrocarbon-based engines. Some of these questions are addressed through the analysis of restrictions in the flow-circuit, including valves used to control the system while in operation.

As oil prices continue to rise and the world supply of oil continues to diminish, alternative methods of propulsion will become more important, to reduce dependency on oil, and to help reduce harmful exhaust emissions. A propulsion system that doesn't dependent on oil and doesn't pollute the environment, such as the liquid nitrogen system, is needed. Production of liquid nitrogen could be facilitated by using renewable power sources as well, such as wind or solar power [15 and 18].

Purpose of the Study

Liquid nitrogen propulsion systems have been proven to function while producing only the emission of nitrogen gas [18]. The purpose of this research is to answer the question: "Can a liquid nitrogen propulsion system be produced that achieves a mechanical efficiency (η_{II}) of 90%, for use in an experimental automotive application?"

Statement of the Problem

The problem addressed by this study is to calculate η_{II} for a practical system while optimizing the flow of nitrogen gas with minimal restrictions through the flow circuit, including all plumbing, valves, pistons, tanks, and other components. This study addresses the issue by calculating all losses in the propulsion and warming systems and comparing these values to the total energy available for propulsion to arrive at an overall efficiency.

Scope of the Study

This study is limited to a single nitrogen propulsion system design. The system design consists of two double-acting pistons (Fig. 1). The pistons are sized to produce similar amounts of force at their given operating pressures. The liquid nitrogen is heated by the atmosphere to generate pressure to approximately 500 psi to power the high-pressure piston. Each stroke of the piston powers one drive wheel. The exhausted gas from the high-pressure piston is accumulated and reheated at approximately 90 psi to power the low-pressure piston. Again, each stroke powers one drive wheel. The exhausted gas from the low-pressure system is released into the atmosphere. All piping and valves from the tank to the high-pressure piston is referred to as flow circuit one and has an initial pressure of 500 psi. All piping and valves between the high-pressure piston and the low-pressure piston will be referred to as flow circuit two and will have an initial pressure of 100 psi. The vehicle speed to be tested at will be 30 mph at a temperature of 20 °C.

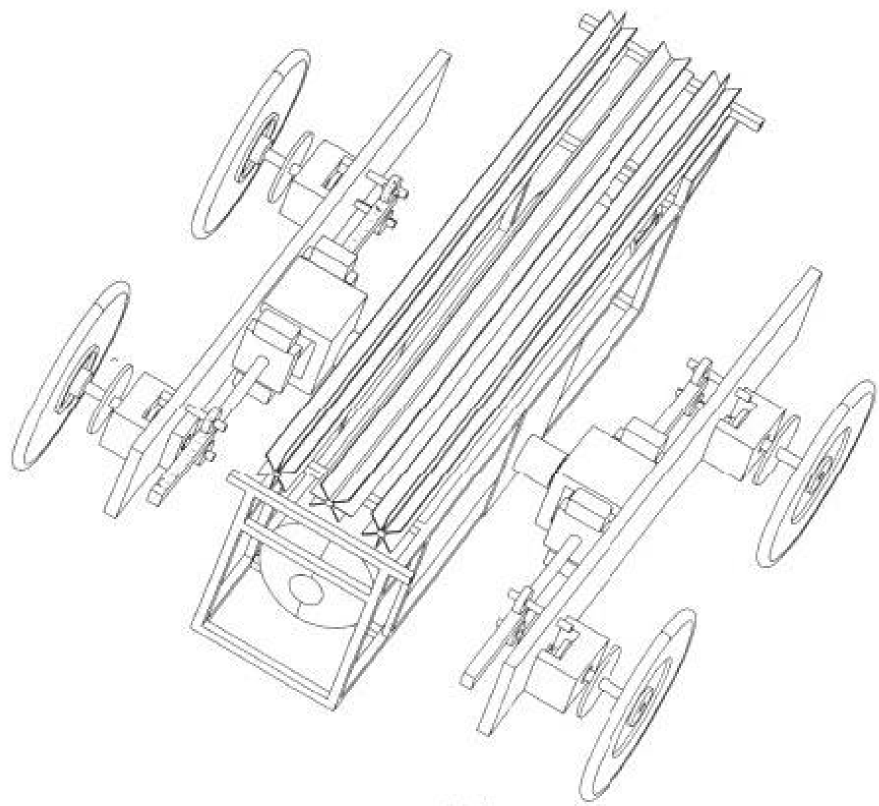

Fig. 1 – Drive system schematic of a dual, double-acting nitrogen propulsion system.

Significance of the Study

This study will add to the body of knowledge pertaining to alternatively fueled propulsion systems for automotive applications and aid in determining whether liquid nitrogen propulsion systems should be studied further as a viable alternative to gas-engines or other methods of propulsion. This further develops the possibility of an abundant, renewable, energy-carrier that produces only the emission of nitrogen gas. The results of this study may help foster further research and possible development of nitrogen propulsion systems in the future [14, 15, 16, 17, and 18].

Research Questions

Two research questions are addressed in this study:

1. Is the η_{II} (mechanical efficiency) of the double-acting pistons at least 90%?

The two pistons were analyzed using a MATLAB Simulink computer simulation of pneumatic engine operation using a double acting piston [2]. Inputs include the mass of the piston and all moving parts (piston, rod, rack, gear, etc.), useful areas of each cavity, input and output pressures, and required piston velocity.

Null: The η_{II} (mechanical efficiency) for each piston is equal to or greater than 90%.

H_{O1}: $\eta_{II\ Piston} \geq 90\%$

Alternative: The mechanical efficiency for each piston is less than 90%.

H_{A1}: $\eta_{II\ Piston} < 90\%$

2. Are the pressure drops in the flow circuits (high pressure and low pressure) more than 10%, based on the mass flow rate (MFR) requirements of each piston?

Each flow circuit is analyzed by utilizing Design Flow Solutions DesignNet 4, NIST

(National Institute of Standards and Technology) Reference Fluid Thermodynamic and Transport Properties data for nitrogen, and MFRs calculated from the MATLAB Simulink simulation. All simulation calculations were hand-checked.

Null: The pressure losses for each circuit are less than or equal to 10%.

H_{O2}: $P_{L\ Circuit} \leq 10\%$

Alternative: The pressure losses for each circuit are greater than 10%.

H_{A2}: $P_{L\ Circuit} > 10\%$

Methodology

1. Gas properties for nitrogen at the required pressures (500 psi and 100 psi) and temperatures (-196 °C and 20 °C) were determined according to NIST reference fluid thermodynamic and transport properties [7].

2. Data for each piston was entered into a MATLAB Simulink computer simulation of pneumatic engine operation to generate MFRs and velocities [10]. These inputs include the mass of the piston and all moving parts (piston, rod, rack, gear, etc.), useful areas of each piston, input and output pressures, gas temperature at inlet, and required piston velocity.

The general equation of the piston's motion (Fig. 2) can be written as

$$M\,(d^2x/dt^2) = p_1S_1 - p_2S_2 - F$$

where

- p_1 – pressure in cavity 1
- p_2 – pressure in cavity 2
- S_1 – useful area of the piston for side 1
- S_2 – useful area of the piston for side 2
- F – resistance force (force of friction and the loading force)

- M – mass of the piston with all moving parts

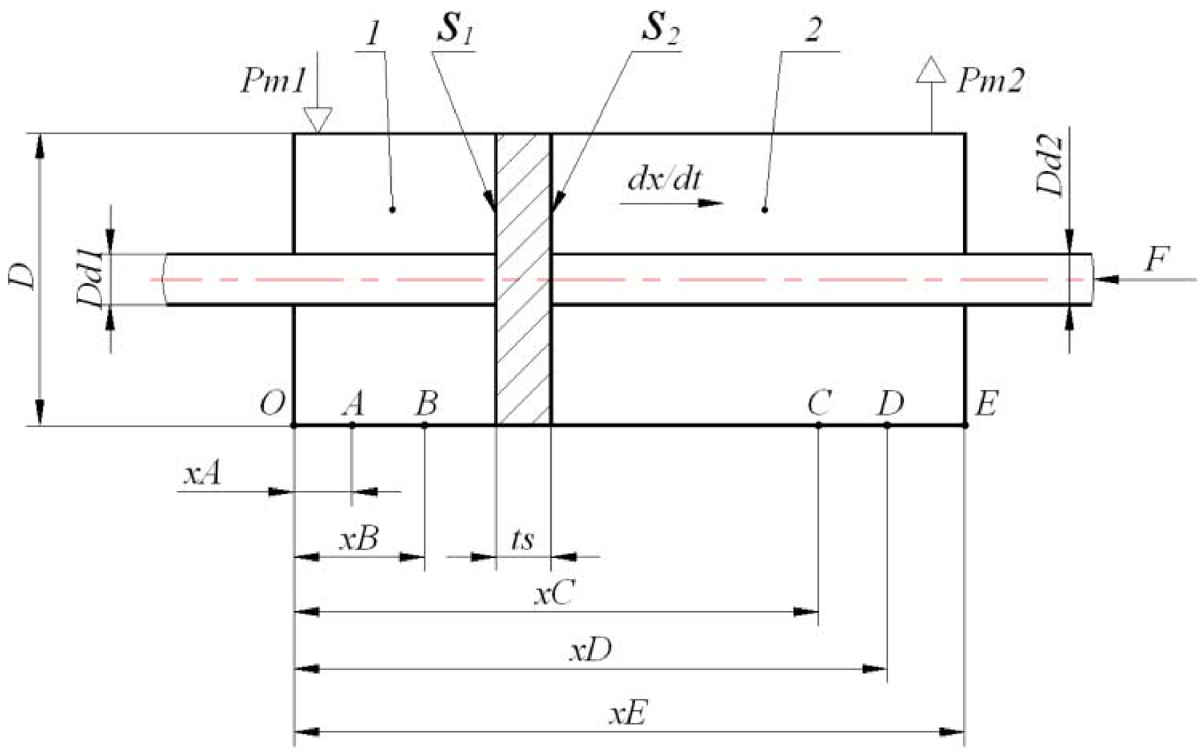

Fig. 2 – Schematic of a double-acting piston with references to the piston motion equation [10].

The MFR required for each piston can be calculated with the following equation [11]:

$$M = \rho \cdot V \cdot A$$

where

- ρ – density of the gas
- V – velocity of the piston
- A – useful area of the piston

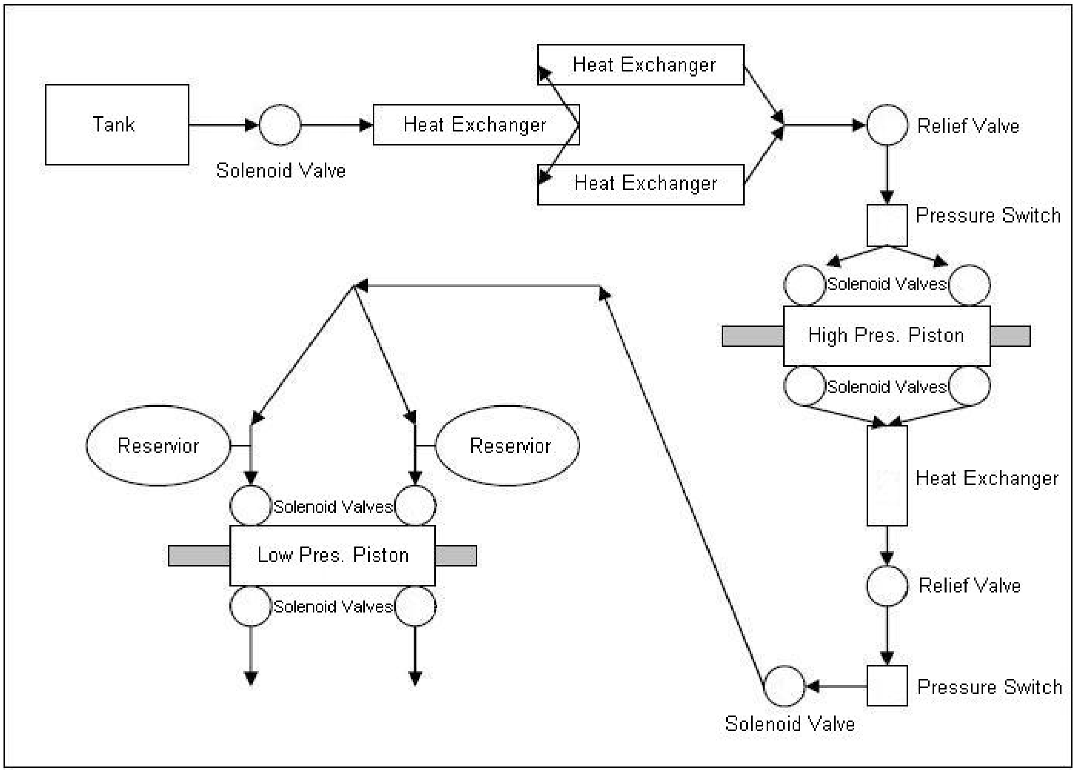

Fig. 3 – Flow diagram of the dual, double-acting piston nitrogen propulsion system.

3. Design Flow Solutions DesignNet 4 will be used to determine the pressure losses of each component in both flow circuits, based on the MFR for each piston. Each section of pipe, each valve, and every fitting was analyzed (Fig. 3). Minor losses were determined by using the following equation [7]:

$$h_L = K (v^2/2g)$$

where

- h_L – minor loss
- K – resistance coefficient
- v – average velocity of flow in the pipe in the vicinity where the loss occurs
- g – gravity

where K would typically have one of the following values [7]:

Reducer – ¾ to ½ in.	K = 0.55
Enlarger – ½ to ¾ in.	K = 0.16
90° Elbow – ½ in.	K = 0.38
Solenoid Valve – ½ in.	K = 9.23
Tee (run-through) – ½ in.	K = 0.54
Reducer – ½ to ⅜ in.	K = 2.96
Solenoid Valve – ⅜ in.	K = 10.86

4. All pressure losses will be summed to determine the total pressure loss for each circuit. If the results show pressure losses of more than 10% for the circuit, the largest pressure losses will be examined to determine whether the pressure loss can be reduced by using a larger or different component in place of a restrictive one.

The end result is a summary that indicates modifications necessary to attain losses of less than 10% (the null hypothesis) for each circuit or indicates that it is unlikely that losses can be reduced to that level using the proposed technology (existing heat exchangers, piping, valves).

Limitations

This research is limited to the use of NIST-generated properties, calculated losses, and energy extraction. The dual-piston version of the liquid nitrogen powered car is the base case for the computer models. The wheel speed used to determine the MFR of the piston was limited to 30 mph (44 ft/s). The nitrogen was limited to the temperature range of -196 °C to 20 °C, and pressures of 500 psi, 100 psi, and 14.7 psi. The analysis of circuit components will be limited to pressure losses based on interior roughness or the resistance coefficient. The research material for this study was limited to library resources (including electronic resources) available at UNT and NIST calculations. Facilities were limited to those available at UNT.

Assumptions

The following assumptions were made:

- Testing conditions are constant.
- Pressure loss in the piston cavity is negligible at the valve closing.
- Ambient temperature is 20 °C.
- The Design Flow Solutions DesignNet 4 software and the MATLAB Simulink computer simulation of pneumatic engine operation are suitable for the purposes of this research.
- The computational models reasonably approximate reality.
- The nitrogen temperature range specified in the procedure is appropriate for the purposes of this research.

CHAPTER 2

REVIEW OF THE LITERATURE

The development of liquid nitrogen propulsion systems is not widespread. In 1996, UNT and the University of Washington both developed vehicles powered by liquid nitrogen without the knowledge of the other's development (Fig. 4 and 5). In the years following, one additional liquid nitrogen powered vehicle was developed by the Kharkov National Automobile and Highway University (KNAHU) along with help from UNT (Fig. 6). A second liquid nitrogen powered vehicle is currently being developed at UNT, powered by a dual, double-acting piston system. The development of this new vehicle is focused on improving the performance and efficiency of liquid nitrogen propulsion systems. The intent is to aid in determining whether liquid nitrogen propulsion systems should be studied further as a viable alternative to gas-engines or other alternative methods of propulsion [18].

Fig. 4 – UNT "LN2 Cool Car."

Fig. 5 - University of Washington’s LN2000 vehicle.

Fig. 6 - Kharkov National Automobile and Highway University's liquid nitrogen vehicle.

Cryogenic Heat Engines

A cryogenic heat engine uses a "cryogenic substance to produce useful energy" [17]. The cryogenic substance is placed into an enclosure or tank where it is allowed heat up, generating pressure. The resulting pressure is used to do work, such as turning a motor or extending a cylinder. This process is very similar to a steam engine, in which water is boiled by an external heat source to produce steam. The steam is under pressure and is used to do work. The primary difference between the two types of engines is that the cryogenic engine uses ambient heat from the atmosphere to generate pressure. The ambient atmosphere provides the heat, bringing the nitrogen from -196 °C up to near ambient temperature [17]. The challenge for researchers is getting the most work out of the available energy, $[|h| \times (T_{amb} - T_0)]/T_{amb}$, that liquid nitrogen offers. Starting with $|h|$ =796 kJ/kg, T_{amb} =296 K, and T_0 =77 K, the available energy from the nitrogen is 588.93 kJ/kg [5]. A dual stage engine with a reheat in between stages was selected as a more efficient way to extract energy from the nitrogen before it is expelled into the atmosphere (Fig. 7 and 8). High-efficiency, multistage turbines might also be looked at in the future as research continues. [18]

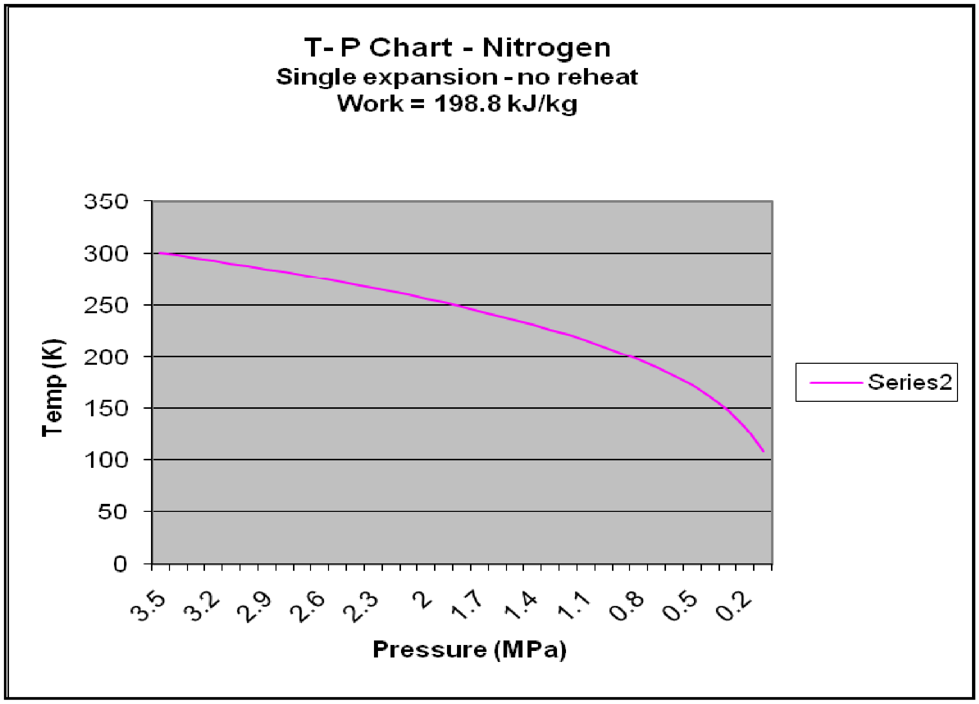

Fig. 7 – T-P chart of a single nitrogen expansion (no-reheat).
Total work achieved is 198.8 kJ/kg.

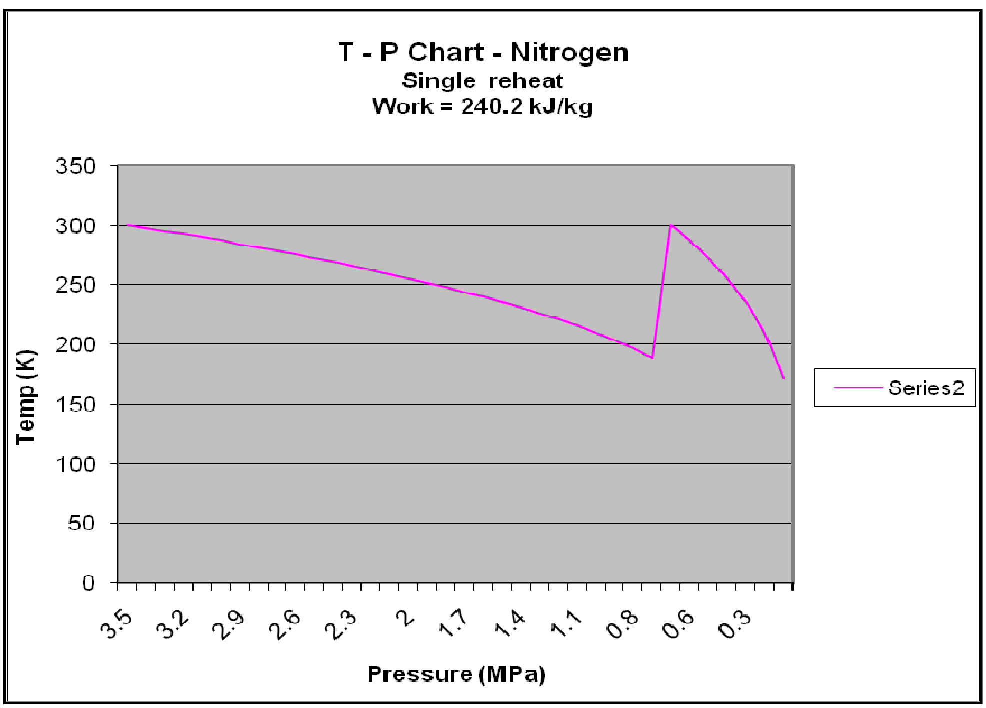

Fig. 8 – T-P chart of a nitrogen expansion with a single reheat.
Total work achieved is 240.2 kJ/kg.

Double-Acting Piston Simulation

There is considerable interest in the use of a double-acting pneumatic piston for nitrogen engine operations [10 and 18]. This interest includes the development of a computer simulation designed to determine the basic operating parameters of the piston being considered [10]. These parameters include piston speed, gas consumption, power output, and mechanical efficiency, which are the primary areas of interest for this study. The model considers a single stroke of the piston from one end to the other. During this stroke, gas fills the intake side of the piston while gas is exhausted from the opposing side of the piston. The position of the piston during the closing and the opening of the intake and exhaust valves are also considered, along with the mass of the piston and the load it is acting against. This simulation uses fixed-valve timing based on piston position. This mathematical model has been compiled into a MATLAB-SIMULINK for use in determining the operating characteristics of various double-acting pistons (Fig. 9).

Flow of Fluids through Valves, Fitting, and Pipe

The transportation of fluids from one point to another is most commonly performed through the use of pipe, along with fittings for redirection, and valves to control flow. This flow is affected by the properties of the fluid, the properties of the pipe, and the properties of the valves and fittings used throughout the flow circuit [1, 3, 8, 9, 7, 11, 12, and 19]

The physical properties of a fluid determine the pressure drop due to flow through a flow circuit. The fluid can be liquid or gas, hot or cold, and can subjected to small or large amounts of force. The flow characteristics of a fluid are dependent on the following properties [7]:

- Viscosity, a substance's readiness to flow when external forces act upon it
- Specific density, a substance's weight per specific volume

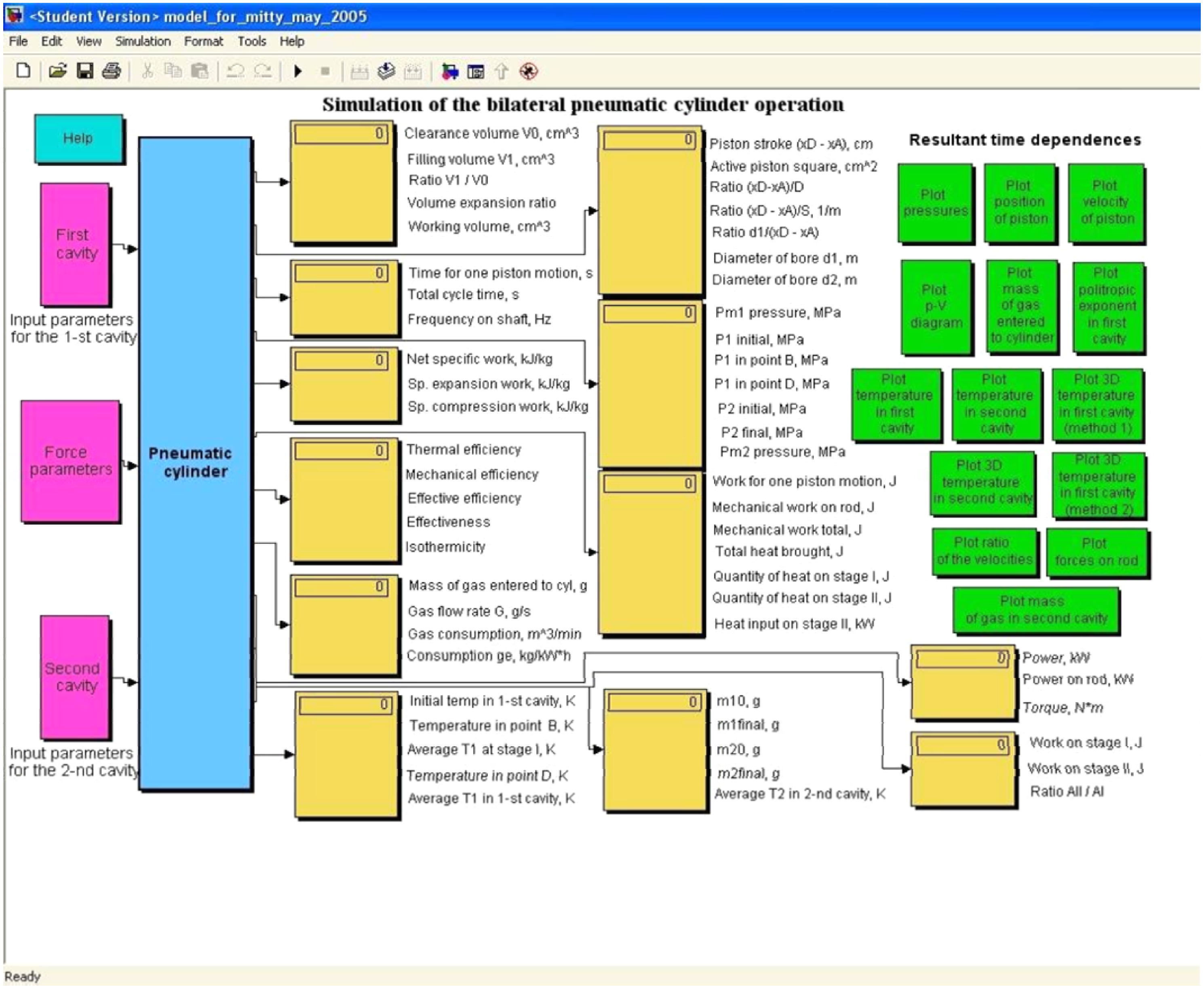

Fig. 9 - MATLAB-Simulink Simulation for double-acting pneumatic piston operation.

- Specific volume, a substance's volume per specific weight (reciprocal of specific weight)
- Specific gravity, ratio of a substance's specific weight at a specified temperature to the specific weight of water at 60 °F

A substance with a low viscosity will flow through pipe with less resistance than a high-viscosity substance would with the same amount of force acting upon it. The viscosity may also change depending on the temperature of a fluid [1, 3, 8, 9, 7, 11, 12, and 19].

The nature of pipe flow is laminar or turbulent, depending on velocity. Laminar flow occurs when fluid flows through the pipe uniformly, without disruptions or turbulence. It can be "characterized by the gliding of concentric layers past one another in orderly fashion" [7], with the innermost cylinder travelling at the highest velocity and the outermost cylinder at zero velocity. As velocity of the fluid is increased, it reaches a critical velocity, at which flow begins to become disturbed. As velocity increases past that critical value, flow becomes turbulent, where "there is an irregular random motion of fluid particles in directions transverse to the direction to the main flow" [7]. Under turbulent flow, the velocity of the fluid is more uniform from the center to the outer wall of the pipe, unlike laminar flow. The critical velocity of a fluid depends on the specific weight and viscosity of the fluid, the pipe diameter, and the velocity of flow (Fig. 10). These four properties are used to determine the Reynolds number, a dimensionless ratio of the dynamic forces of mass flow to the shear stress due to viscosity. Typically, a Reynolds number under 2000 is considered laminar, whereas a Reynolds number above 4000 is considered turbulent. The velocity of nitrogen through the high-pressure side ranged from 5.2 m/s to 35.4 m/s and from 4.1m/s to 15.3 m/s through the low-pressure side [1, 3, 7, 8, 9, 11, 12, 18, and 19].

As a fluid flows though a section of pipe, friction is generated from particles bumping

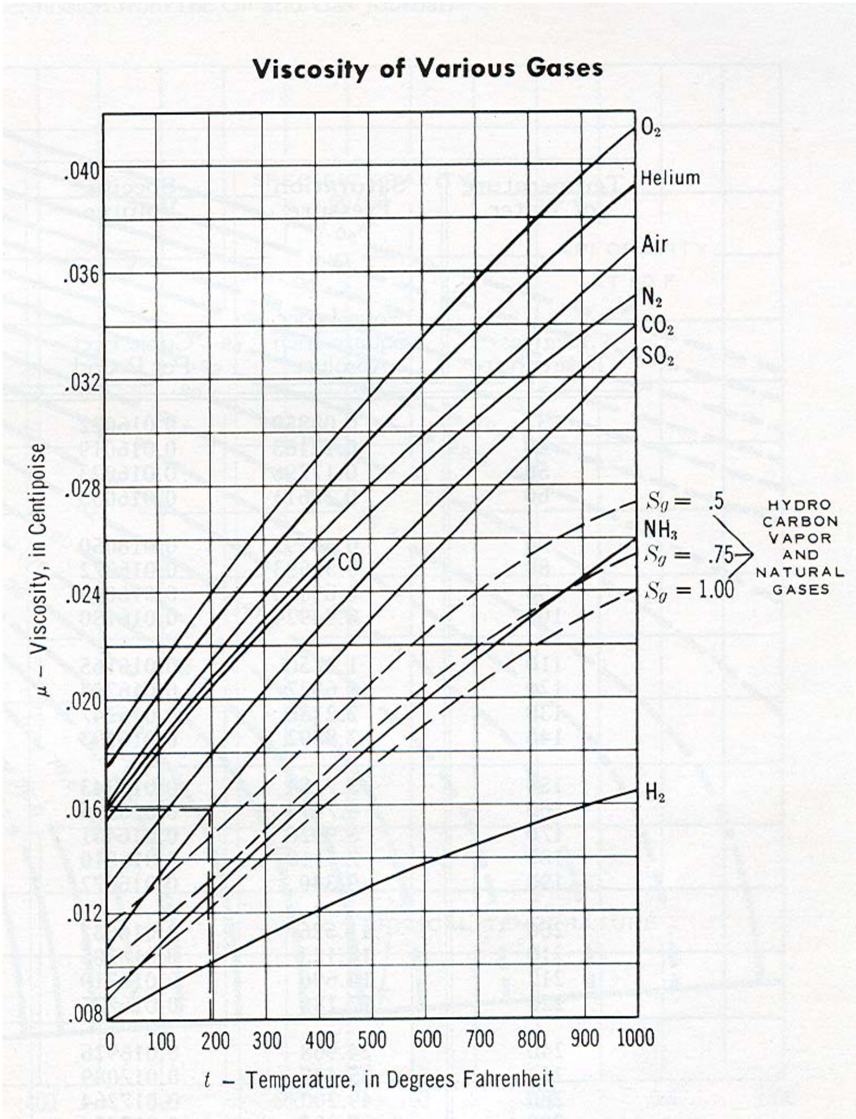

Fig. 10 – Gas viscosity versus temperature [7].

into each other, causing a drop in pressure. The general equation for pressure drops is Darcy's formula, expressed in feet of fluid or drop in pressure [7]. Darcy's formula takes into account the density of the fluid, changes in elevation, the length and diameter of pipe, and the friction factor of the pipe.

Pressure losses in a flow circuit are caused by friction, changes in direction, obstructions, and changes in the cross-sectional area of the flow. Valves and fittings represent a large percentage of flow losses through a flow circuit. A large amount of research and testing has been done over the years to gather pressure loss data on a large range of valves and fittings, but it is impossible to test every size and type of valve or fitting. Through the extrapolation of available pressure loss data, we now have commonly used concepts to determine these losses: resistance coefficient K, equivalent length L/D, and flow coefficient C_v (Fig. 11). The resistance coefficient K is the friction of an equivalent length of a section of straight pipe that would cause the same pressure drop as the fitting or valve under the same conditions [4 and 7].

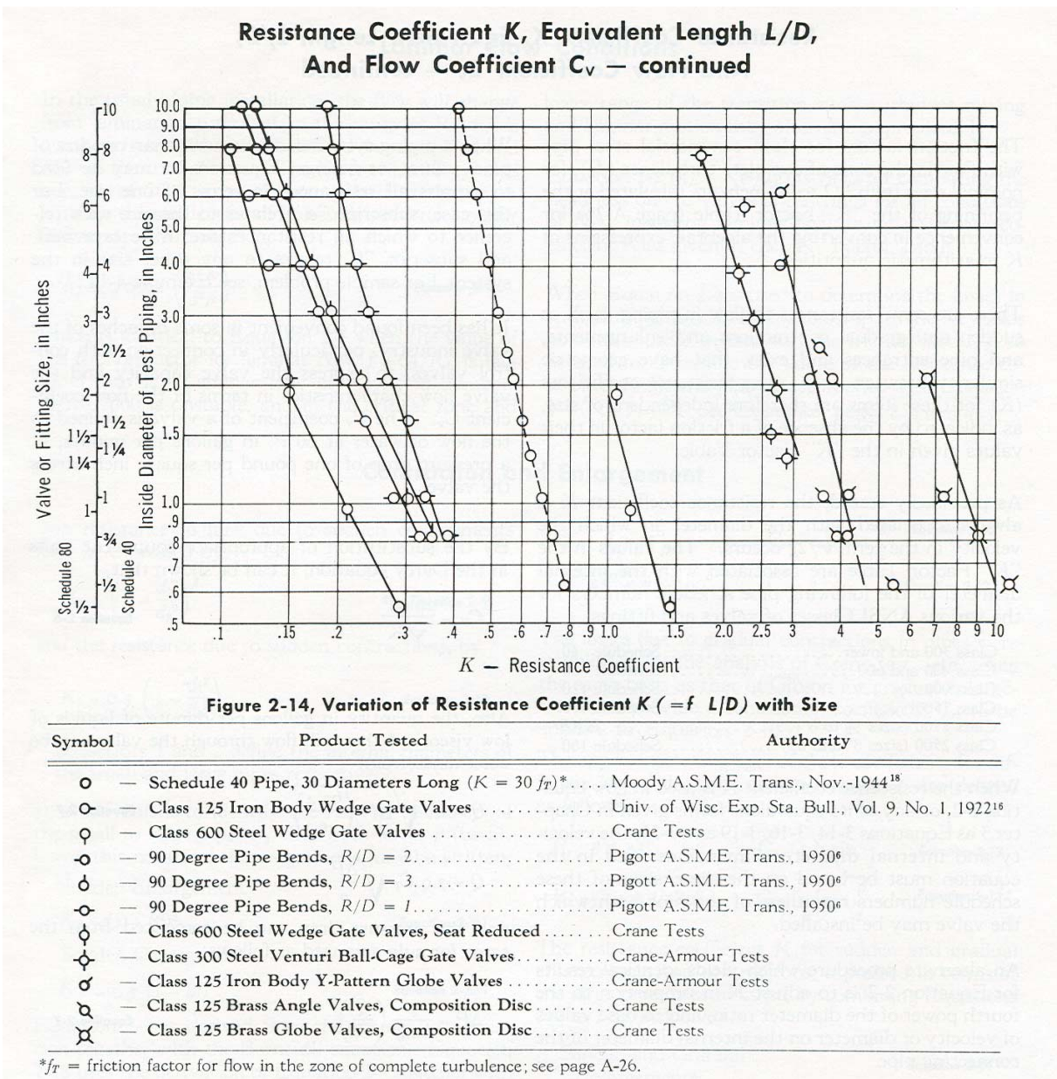

Fig. 11 – Crane tesistance coefficient chart [7].

Contribution of This Study

This study contributes to the understanding of cryogenic heat engines and the potential for liquid nitrogen as a viable energy-carrier for automotive applications. It also contributes to the understanding of the operation of double-acting pneumatic pistons and the flow characteristics of fluids though pipes, valves, and fittings.

Summary of the Chapter

The amounts of literature available on cryogenic heat engines are limited but very relevant to this study. The literature available for the double-acting piston is somewhat limited but is directly linked to current research. Fluid flow analysis is a very established because the principles have not changed in many years, leading to the availability of a very thorough and comprehensive literature.

CHAPTER 3

METHODOLOGY

Gas properties for nitrogen at the required pressures (500 psi and 100 psi) and temperatures (-196 °C and 20 °C) were determined according to NIST reference fluid thermodynamic and transport properties [13]. The tables for these properties were used for inputs into the piston simulation and the flow circuit analysis software.

Table 1 – Nitrogen-specified state points [13].

	Temp (°C)	Pressure (MPa)	Density (kg/m3)	Enthalpy (kj/kg)	Entropy (kg/kg-K)
1	-196.150	3.500	816.050	-120.310	2.802
2	20.000	3.500	40.450	296.440	5.742
3	20.000	0.600	6.905	302.910	6.286
4	20.000	0.100	1.150	304.060	6.822

Data for each piston was entered into a MATLAB Simulink computer simulation of double-acting pneumatic engine operation to generate flow velocities [10]. These inputs include: the mass of the piston and all moving parts (piston, rod, rack, gear, etc.), useful areas of each piston, input and output pressures, gas temperature at inlet, and required piston velocity. The general equation of the piston motion (Fig. 2) can be written as follows:

$$M\,(d^2x/dt^2) = p_1S_1 - p_2S_2 - F$$

where

- p_1 – pressure in cavity one
- p_2 – pressure in cavity two
- S_1 – useful area of the piston for side one

- S_2 – useful area of the piston for side two
- F – resistance force (force of friction and the loading force)
- M – mass of the piston with all moving parts

The mass of the high-pressure piston is approximately 25 kg, whereas the low-pressure piston is approximately 30 kg.

The MFR required for each piston can be calculated with the following equation [11]:

$$M\ dot = \rho \cdot V \cdot A$$

where

- ρ – density of the gas
- V – velocity of the piston
- A – useful area of the piston

The required velocity of the piston is determined by the diameter of the wheels used, the piston rod travel, the diameter of the gear acted on by the rack, and the desired vehicle speed. To achieve the desired velocity of the piston under the load of the vehicle, valve timing must be determined.

The load of the vehicle at 30 mph is calculated using the following equations [2]:

$$P_{over} = V\ (F_d + F_g)$$

where

- V – velocity of the vehicle
- F_d – force due to drag
- F – force due to gravity

where F_d is calculated using the following equation [2]:

$$F_d = (C_d) * (\rho V^2/2) * A$$

where

- C_d – coefficient of drag
- ρ – density of air at sea level
- V – velocity
- A – surface area interacting with the air (front cross-section of the vehicle)

and where F_g is calculated with the following equation [2]:

$$F_g = (C_r) * (M)$$

where

- C_r – coefficient of rolling resistance
- M – mass of the vehicle

Valve timing must be optimized so that the piston will achieve full travel while not using any more nitrogen than necessary. To achieve this optimization, the intake valve closing position was tested in 0.25-inch intervals starting at 6 inches of piston travel. This testing was performed for each piston.

The simulation will determine the mechanical efficiency of the piston and the MFR required to maintain the desired velocity of 30 mph to aid in the flow circuit analysis.

Design Flow Solutions DesignNet 4 was used to determine pressure losses of each component in both flow circuits, based on the MFR for each piston. Each section of pipe, each valve, and every fitting was analyzed (see Appendix A). Minor losses were determined by using the following equation [7]:

$$h_L = K (v^2/2g)$$

where

- h_L – minor loss
- K – resistance coefficient
- v – average velocity of flow in the pipe (or component) in the vicinity where the loss occurs
- g – gravity

All pressure losses were summed to determine the total pressure loss for each circuit and the required initial pressure to sustain the flow rate and pressure requirements of each piston. Type K Copper tubing ½ inch in diameter was specified as the piping for use in this analysis. The heat exchanger piping is ¾ inch in diameter, schedule 20, aluminum tubing.

If the results show pressure losses of more than 10% for the circuit, the largest pressure losses are examined to determine whether the pressure loss can be minimized by using a larger component in place of the restrictive one.

The end result is a summary that indicates modifications necessary to attain losses of less than 10% (the null hypothesis) for each circuit or states that it is unlikely that losses can be reduced to that level with the proposed technology (existing heat exchangers, piping, valves).

CHAPTER 4

DATA COLLECTION

Piston Simulation Data

The propulsion system utilizes 28-inch-diameter wheels, driven by a 2 inch diameter pinion gear, driven by a gear rack that travels 12 inches. The wheel will rotate six times per piston stroke. The test velocity is 30 mph, or 13.4 m/s, requiring the wheel to rotate six times per second, thus requiring the piston to travel a single stroke in 1.0 seconds. This is calculated with the following equation:

$$T_{stroke} = R_s / (V/D*\pi)$$

where

- T_{stroke} – amount of time that the piston must travel one full stroke to maintain a given velocity
- R_s – revolutions per one piston stroke
- V – desired velocity
- D – diameter of wheel

The given force that the pistons must apply is 675 lb or 3000 N to overcome the force of friction and drag. The pistons have the following dimensions:

- High-pressure cylinder
  - piston diameter – 2.5 in. (0.0635 m)
  - shaft diameter – 1 in. (0.0254 m)
  - piston throw – 12 in. (0.3048 m)
  - rod length – 30 in. (0.762 m)
  - mass – 55lb (25 kg)

- valve diameter – 0.25 in. (0.00635 m)

- Low-pressure cylinder
  - piston diameter – 6 in. (0.1524 m)
  - shaft diameter – 1.375 in. (0.03493 m)
  - piston throw – 12 in. (0.3048 m)
  - rod length – 31.375 in. (0.79216 m)
  - mass – 66lb (30 kg)
  - valve diameter – 0.5 in. (0.0127 m)

These dimensions were used to run the simulation and the piston position for intake valve closure was changed from 6 inches of piston travel in 0.25-inch increments. The goal was to find the shortest position that allowed full piston travel while not using anymore nitrogen than was necessary. The following data was recorded from those simulations (Tables 2 and 3):

Table 2 – High-pressure piston simulation results.

High-Pressure Piston				MFR (g/s)	Mass (g)	mph	kph
Inlet Valve Closes		Exhaust Valve Closes				30	48.28
Bx (in.)	Bx (m)	In.	m			Power (kW)	Mech Eff. (%)
6.00	0.1524	11.5	0.2921	12.92	12.10	0.67	0.9966
6.25	0.1588	11.5	0.2921	13.51	12.65	0.71	0.9966
6.50	0.1651	11.5	0.2921	14.10	13.20	0.75	0.9966
6.75	0.1715	11.5	0.2921	14.68	13.75	0.78	0.9966
7.00	0.1778	11.5	0.2921	15.25	14.29	0.81	0.9966
7.25	0.1842	11.5	0.2921	66.28	14.82	3.77	0.9966
7.50	0.1905	11.5	0.2921	73.26	15.35	4.04	0.9966
8.00	0.2032	11.5	0.2921	84.21	16.43	4.38	0.9966

Table 3 – Low-pressure piston simulation results.

Low-Pressure Piston				MFR (g/s)	Mass (g)	mph	kph
Inlet Valve Closes		Exhaust Valve Closes				30	48.28032
Bx (in.)	Bx (m)	In.	m			Power (kW)	Mech Eff. (%)
6	0.1524	11.5	0.2921	11.15	10.45	0.7316	0.9954
6.25	0.15875	11.5	0.2921	11.885	11.135	0.77255	0.9954
6.5	0.1651	11.5	0.2921	12.62	11.82	0.8135	0.9954
6.75	0.17145	11.5	0.2921	13.355	12.505	0.85445	0.9954
7	0.1778	11.5	0.2921	47.35	13.54	2.954	0.9954

This flow rate data was used at the required pressures for analyzing the flow circuit for each piston. Summations of all minor losses throughout each flow circuit were performed (Table 4). The following results for those calculations were recorded:

Table 4 - Summation of all minor losses in each flow circuit.

	MFR (g/s)	Inlet Pressure (psi)	Outlet Pressure (psi)	Pressure Drop (psi)	Change (%)
High	66.28	548.4	500	48.4	8.83
Low	47.35	106.3	100	6.3	5.93

CHAPTER 5

RESULTS AND ANALYSIS

The simulation results for the high-pressure piston show that under the required load, the intake valve must stay open through the first 7.25 inches of piston travel to achieve a full piston stroke and the necessary velocity (Table 2). Achieving a full piston stroke requires an MFR of 66.28 g/s or 0.06628 kg/sec. The high-pressure piston achieves a mechanical efficiency of 0.9966 or 99.66% for the piston alone.

The results for the low-pressure piston show that under the required load, the intake valve must stay open through the first 7.0 inches of piston travel to achieve a full piston stroke and the necessary velocity (Table 3). This requires an MFR of 47.35 g/s or 0.04735 kg/sec. The low-pressure piston achieves a mechanical efficiency of 0.9954 or 99.54%.

The piston simulation results are based on a constant load, so that the same amount of force is necessary for each piston motion. Under this constant load, if the intake valve closes before the piston travels 7 or 7.25 inches (high pressure and low pressure, respectively), the piston will not achieve a full stroke.

The flow circuit analysis of the high-pressure flow circuit shows that to achieve an outlet pressure of 500 psi at 66.28 g/s requires an initial pressure of 548.4 psi, which is an 8.83% loss through the circuit, an efficiency of 91.17%. The largest single point of pressure loss was due to the 36.1-psi drop at the control valve (Fig. 12) because a valve ¼-inch diameter was used in comparison to the rest of the flow circuit (½-inch diameter). A control valve with a larger diameter would reduce the amount of pressure loss, but the high-pressure piston only allows for valves of up to ⅜-inch diameter (Fig. 13).

Fig. 12 – High-pressure solenoid valve.

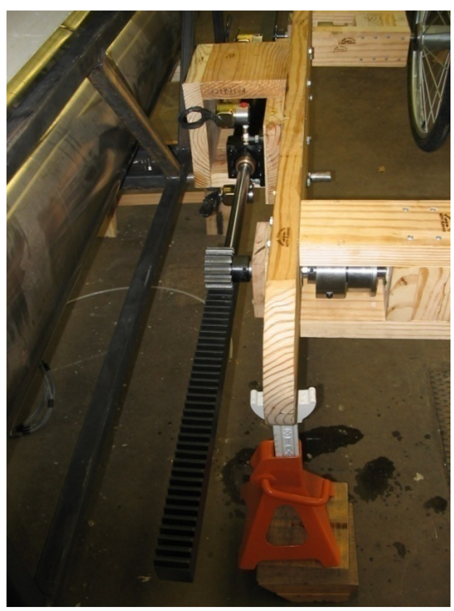

Fig. 13 – High-pressure drive assembly.

The flow circuit analysis of the low-pressure flow circuit shows that to achieve an outlet pressure of 100 psi at 47.35 g/s requires an initial pressure of 106.3 psi which is a 5.93% loss through the circuit, an efficiency of 94.07%. There were no significant losses of pressure at any particular point of this flow circuit (Fig. 14 and 15).

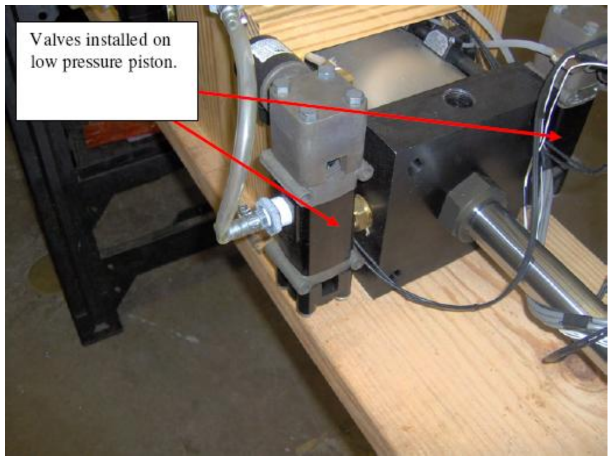

Fig. 14 – Low-pressure solenoid valve.

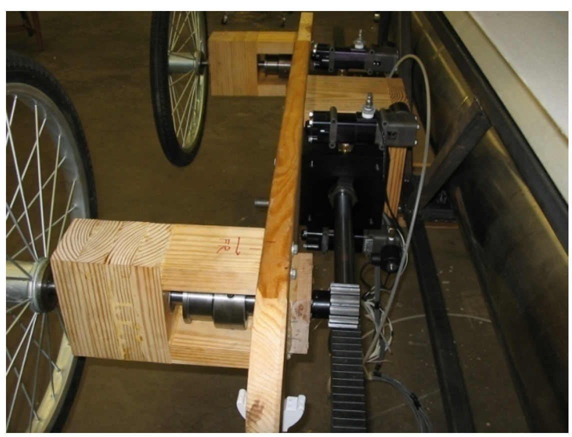

Fig. 15 – Low-pressure drive assembly.

The flow circuit analysis was performed with one control valve open and the other control valve closed, just like in normal operation. The analysis was performed with no consideration for elevation change because any elevation change would be negligible.

CHAPTER 6

SUMMARY, CONCLUSIONS, AND RECOMMENDATIONS

Summary of the Study

The purpose of this study was to perform a mechanical efficiency analysis of a dual, double-acting piston nitrogen propulsion system. This study focused on the two main components of this system, the driving pistons and the flow circuits to power those pistons.

The double-acting pistons were simulated to achieve their full piston stroke while using the least amount of nitrogen necessary. The pistons would prove themselves to be highly efficient with proper valve timing that is adaptive to the load. The study also shows the difference in power of each piston at its respective pressure.

The flow circuit analysis shows that both flow circuits can adequately handle the pressure and flow requirements that the pistons require. The analysis also showed that the control valves on the high-pressure circuit are not ideal for this application (Fig. 12). The large decrease in diameter at the valve, ½ inch down to ¼ inch, increased the pressure loss. The piston valve ports allow for a valve up to a ⅜ inch in diameter and could possibly be drilled larger, but no other valve was available at the time that operated on a 12-volt system, and that could handle the pressure requirements. Development of larger-diameter high-pressure valves appears to be a necessary enabling technology worthy of research based on these results.

Answer to the Research Questions

1. Is the η_{II} (mechanical efficiency) of each piston at least 90%?

The two pistons were analyzed with a MATLAB Simulink computer simulation of a pneumatic engine operation using a double-acting piston [2]. Inputs included the mass of the piston and all moving parts (piston, rod, rack, gear, etc.), useful areas of each cavity, input and output pressures, and required piston velocity.

Null: The η_{II} (mechanical efficiency) for each piston is equal to or greater than 90%.

H_{O2}: $\eta_{II\ Piston} \geq 90\%$

Alternative: The mechanical efficiency for each piston is less than 90%.

H_{A2}: $\eta_{II\ Piston} < 90\%$

Answer: The results from the MATLAB Simulink computer simulation show that the high-pressure and low-pressure pistons achieved η_{II} values of 99.66% and 99.54%, respectively. These values are within the null hypothesis: therefore the answer is yes - the η_{II} of each piston is at least 90%.

2. Will the pressure drops in the flow circuits (high-pressure and low-pressure) be more than 10%, based on the MFR requirements of each piston?

Each flow circuit was analyzed by utilizing Design Flow Solutions DesignNet 4, NIST (National Institute of Standards and Technology) Reference Fluid Thermodynamic and Transport Properties data for nitrogen, and MFRs calculated from the Simulink simulation.

Null: The pressure losses for each circuit are less than or equal to 10%.

H_{O1}: $P_{L\ Circuit} \leq 10\%$

Alternative: The pressure losses for each circuit are greater than 10%.

H_{A1}: $P_{L\ Circuit} > 10\%$

Answer: The use of Design Flow Solutions DesignNet 4, NIST (National Institute of Standards and Technology) Reference Fluid Thermodynamic and Transport Properties data for nitrogen, and MFRs calculated from the Simulink simulation resulted in pressure losses of 8.83% and 5.93% for the high-pressure and low-pressure flow circuits, respectively. These values are within the null hypothesis: therefore, the answer is yes – pressure losses for each circuit are less than or equal to 10% at 30 mph.

Conclusions

After analyzing this research study, the designers concluded that this dual double-acting piston nitrogen propulsion system meets their intended goals. It would be beneficial to look for alternative control valves for the high-pressure circuit that are less restrictive that the ones currently implemented.

Strengths of the Study

The main strength of this study that it shows that there is a viable potential for cryogenic heat engines. This study should help encourage future research into the potential use of cryogenic heat engines for automotive applications.

Recommendations for Further Research

Further cryogenic heat engine research should continue in its current direction, working to get the more work out of the potential that liquid nitrogen offers. The use of lighter materials for these systems would also be beneficial. It would also be beneficial to develop future simulations that allow variable loads and valve timing, to maximize efficiency at any piston speed. More research to improve high-pressure valves would also be helpful.

APPENDIX

COMPONENTS AND FITTINGS TO BE ANALYZED WITH DESIGN FLOW SOLUTIONS

Liquid Nitrogen Tank – 45 gallon – 3/8 inch NPT female outlet

Solenoid Valve – 500 psi – 12 V – ½ inch NPT female inlet and outlet

Heat Exchanger (3) – 8 fin (4-inch fins) – 96-inch length – aluminum – ¾-inch NPT male

Pressure Relief Valve – 500 psi – ½-inch NPT male inlet

Pressure Switch – 500 psi – ¼-inch NPT male

Piston Control Valves (high pressure) (4) – 12 VDC – ¼-inch NPT female inlet and outlet

High Pressure Piston – 500 psi – 12-inch stroke – 2.5-inch diameter, 3/8-inch NPT female inlets and outlets

Heat Exchanger – circular fins (1½-inch diameter) – ½-inch copper type K tubing

Pressure Relief Valve – 100 psi – ½-inch NPT male inlet

Pressure Switch – 90 psi – 1/8-inch NPT male

Solenoid Valve – 100 psi – 12 V – ½-inch NPT female inlet and outlet

Low-Pressure Reservoir (2) – 4-inch diameter by 12-inch length – 1/8-inch NPT female inlet

Piston Control Valves (4) – 12 VDC – ½-inch NPT female inlet and outlet

Low Pressure Piston – 100 psi – 12-inch stroke – 6-inch diameter – ½” NPT female inlets and outlets

Installation Hardware List

Item	Description	Purpose	Material	Qty	Price	Extended Price
Item 1	**Reservoir**			1		0
	1/2" x 3/8 Male Threaded Nipple	Connection of Check Valve to Tank & Connection of Check Valve to Union	Brass	2	$ 5.00	$ 10.00
	1/2" Female Threaded Union	To allow easy disconnect of copper tube to storage tank.	Brass	1	$ 5.41	$ 5.41
	1/2" Male Threaded Adapter	For connection of Copper Tube to storage tank.	Copper	1	$ 3.49	$ 3.49
	1/2" Tubing	Connect Storage Tank to Solenoid	Copper			
Item 2	**Emergency Shutoff Valve**					$ -
	This Item is Item 3 (Dual Purpose)	Solenoid Shutoff Valve is also emergency shutoff valve.				
Item 3	**Solenoid Shutoff Valve**			1		$ -
	1/2" Male Threaded Adapter	For connection of tubing to shutoff valve on either side.	Copper	2	$ 3.49	$ 6.98
	Pipe Clamps	For securing tubing on either side of valve to frame.		2	$ 0.50	$ 1.00
Item 4	**Heat Exchanger**			3		$ -
	3/4" Female Threaded Union	For easy removal of heat exchangers. Install on each end of heat exchanger.	Brass	6	$ 7.44	$ 44.64
	3/4" to 1/2" Threaded Nipple	For connection of 1/2" tubing to 3/4" union	Brass	6	$ 5.00	$ 30.00
	1/2" Female Threaded Adapter	For connection of tubing to unions on either side of heat exchangers	Copper	6	$ 2.51	$ 15.06
	1/2" x 1/2" x 1/2" Slip X Slip X Slip TEE	Two heat exchangers are connected in parallel	Copper	2	$ 0.45	$ 0.90
	1/2" Tubing	Needed to connect Item 3 to Item 4. Also used to connect heat exchangers. See diagram in report.	Copper			
	1/2" Elbow		Copper	5	$ 0.28	$ 1.40
Item 5	**Pressure Relief Valve (500 PSI)**					$ -
	1/2" Female Threaded Adapter	For connection of tubing to inlet side of valve. For connection of tubing coming from outlet side.	Copper	2	$ 2.51	$ 5.02
	3/4" x 1/2" Threaded nipple	For connection of valve to femail threaded adapter.	Brass	1	$ 5.00	$ 5.00
	Pipe Clamps	For securing tubing on either side of valve to frame.		2	$ 0.20	$ 0.40
Item 6	**Pressure Switch (500 PSI)**					$ -
	1/2" x 1/2" x 1/4" Slip X Slip X Thread TEE	Pressure switch is installed into threaded tee that is brazed to 1/2" tubing	Copper	1	$ 3.36	$ 3.36
	1/2" Tubing	Needed to connect Item 5 to TEE fitting needed for Item 6. Also used for connecting Item 6 Tee Fitting to Item 7	Copper			
Item 7	**High Pressure Piston Control Valves**					$ -
	1/2" x 1/2" x 1/2" Slip X Slip X Slip TEE	Tubing branches from single line to the inlet ports of the two control valves used at the inlet of the piston	Copper	1	$ 0.45	$ 0.45
	1/2" x 1/2" x 1/2" Slip X Slip X Slip TEE	Tubing goes from outlet ports of control valves located on outlet ports of the high pressure piston. Line goes from two branches to one.	Copper	1	$ 0.45	$ 0.45
	1/2" Male Threaded Adapter	Needed for connection of tubing to union	Copper	4	$ 3.49	$ 13.96
	1/2" Female Threaded Union	Needed for easy disconnect of High Pressure Piston from system.	Brass	4	$ 5.41	$ 21.64
	1/2" x 1/4" Male Threaded Nipple	Used to connect Union to Valves	Brass	4	$ 5.11	$ 20.44
	1/2" Tubing	For Connection of Item 8 to Item 10				
Item 8	**High Pressure Piston**					$ -
	3/8" x 1/4" Male Threaded Nipple	Needed for connection of Control Valves to High Pressure Piston.	Brass	4	$ 3.49	$ 13.96
Item 9	**Heat Exchanger**					$ -
	1/2" Slip Coupling	Used to connect heat exchager inline with tubing.	Copper		$ 0.22	$ -

Installation Hardware List

Item	Description	Purpose	Material	Qty	Price	Extended Price
Item 10	**Pressure Relief Valve (100 PSI)**	**McMaster Car Item# 6872K11**		1	**$ 317.06**	**$ 317.06**
	1/2" Female Threaded Adapter	For connection of tubing to inlet side of valve. For connection of tubing coming from outlet side.	Copper	2	$ 2.51	$ 5.02
	3/4" x 1/2" Threaded nipple	For connection of valve to femail threaded adapter.	Brass	1	$ 5.00	$ 5.00
	Pipe Clamps	For securing tubing on either side of valve to frame.		2	$ 0.20	$ 0.40
Item 11	**Pressure Switch (90 PSI)**					**$ -**
	1/2" x 1/2" x 1/2" Slip X Slip X Thread TEE	Pressure switch is installed into threaded tee that is brazed to 1/2" tubing	Copper	1	$ 3.60	$ 3.60
	1/2" Tubing	Needed to connect Item 10 to TEE fitting needed for Item 11. Also used for connecting Item 6 Tee Fitting to Item 12	Copper			
	1/2" x 1/8" Reducer Bushing	Installed into tee so that switch can be installed	Brass	1	$ 1.71	$ 1.71
Item 12	**Solenoid Shutoff Valve**					**$ -**
	1/2" Male Threaded Adapter	For connection of tubing to shutoff valve on either side.	Copper	2	$ 3.49	$ 6.98
	Pipe Clamps	For securing tubing on either side of valve to frame.		2	$ 0.20	$ 0.40
Item 13	**Low Pressure Reservoir**					**$ -**
	1/2" x 1/2" x 1/2" Slip X Slip X Slip TEE	Tubing branches from single line to the inlet ports of the two control valves used at the inlet of the piston	Copper	1	$ 0.45	$ 0.45
	1/2" x 1/2" x 1/2" Slip X Slip X Slip TEE	Tubing leading to each of the inlet control valves on the low pressure pistion branch to low pressure reservoir.	Copper	2	$ 0.45	$ 0.90
	1/2" Female Threaded Union	For easy removal of reservoirs	Brass	2	$ 5.41	$ 10.82
	1/2" Male Threaded Adapter	For connection of tubing to union.			$ 3.49	$ -
	1/2 x 1/8" Reducer Bushing	For steping down in size from union to inlet of Low Pressure Reservoir	Brass	2	$ 1.71	$ 3.42
	1/8" x 1/8" x 2" Threaded Nipple	For connection of Reservoir to unions	Brass	2	$ 0.94	$ 1.88
Item 14	**Low Pressure Piston**					**$ -**
	1/2" x 1/2" Threaded Nipple	For connection of control valves to Piston	Copper	4	$ 1.11	$ 4.44
Item 15	**Low Pressure Piston Control Valves**					**$ -**
	1/2" Female Threaded Union	For easy removal of piston from system	Brass	2	$ 5.41	$ 10.82
	1/2" Male Threaded Adapter	For connection of tubing to union	Copper	2	$ 3.49	$ 6.98
	1/2" x 1/2" Threaded Nipple	For connection of valves to union	Brass		$ 1.11	$ -
Piping	**Type K Copper Tubing Annealed**	**100' Roll Anderson Barrows**		1	**$ 145.00**	**$ 145.00**

Total **$ 728.44**

BIBLIOGRAPHY

Anderson, Blaine W. *The Analysis and Design of Pneumatic Systems*, John Wiley & Sons, Inc., Krieger Publishing Company, 2001. 1967.

Avallone, E. A. and T. Baumeister. *Marks' Standard Handbook for Mechanical Engineers*. 11th ed. New York: McGraw-Hill, Inc., 1997.

Binder, R. C. *Fluid Mechanics*. Englewood Cliffs, NJ: Prentice-Hall, Inc., 1973.

Calgaro, C., J. F. Coulombel, and Thierry Goudon. *Analysis and Simulation of Fluid Dynamics*. Basel, France: Birkhauser Publishing, 2007.

Çengel, Yunus A. and Michael A. Boles. *Thermodynamics: An Engineering Approach*. 4th ed. New York: McGraw Hill, 2002.

Colebrook, C. F. "Turbulent Flow in Pipes," *Journal of the Inst. Civil Eng.* **11**, p. 133, 1938.

Crane Co. Engineering Division. "Flow of Fluids through Valves, Fittings, and Pipe." Crane Co. Technical Paper No. 410, Crane Co.; 21st printing, 1982.

Kambe, T. *Elementary Fluid Mechanics*. Hackensack, N.J.: World Scientific Publishing, 2007.

Koumoutsakos, P. and I. Mezic. *Control of Fluid Flow*. Berlin: Springer Publishing, 2006.

Kudryavtsev, I. N., A. V. Kramskoy, A. I. Pyatak, and M. C. Plummer. "Computer Simulation of Pneumatic Engine Operation," *International Scientific Journal for Alternative Energy and Ecology,* 3(23), pp. 80-89, 2005.

Mott, Robert L. *Applied Fluid Mechanics*. 5th ed. Upper Saddle River, NJ: Prentice Hall, 2000.

Narasimhan, S. *A First Course in Fluid Mechanics*. Boca Raton, FL: CRC Press, 2007.

National Institute of Standards and Technology. *NIST Reference Fluid Thermodynamic and Transport Properties – REFPROP*. Version 7.0. Gaithersburg, MD: U.S. Department of Commerce, 2002.

Ordonez, C. A. "Cryogenic Heat Engine," *American Journal of Phys*ics **64**: pp. 479-481, 1996.

Ordonez, C. A. and M. C. Plummer. "Cold Thermal Storage and Cryogenic Heat Engines for Energy Storage Applications," *Energy Sources* **19**: pp. 389-396, 1997.

Ordonez, C. A. "Liquid Nitrogen Fueled, Closed Brayton Cycle Cryogenic Heat Engine," *Energy Conversion and Management* **41**: pp. 331-341, 2000.

Plummer, M. C., C. P. Koehler, D. R. Flanders, R. F. Reidy, and C. A. Ordonez. "Cryogenic Heat Engine Experiment," *Advances in Cryogenic Engineering* **43**: pp. 1245-1252, 1998.

Plummer, M. C., C. A. Ordonez, and R. F. Reidy. "A Review of Liquid Nitrogen Propelled Vehicle Programs in the United States of America," *Proceedings of the International Scientific Conference on Automobile Transport and Road Enterprises on the Threshold of the Third Millennium*, Paper No. 629.1, pp. 47-52, 2000.

White, Frank M. *Fluid Mechanics*, 5th ed. New York: McGraw-Hill Higher Education, 1998.