Background Information

Problem

The human respiratory system consists of muscles and organs that are responsible for gas exchange between the human body and its environment. Within the lungs, diffusion is responsible for the delivery of oxygen to the blood and the removal of carbon dioxide as waste. This cycle enables the body to metabolize energy via cellular respiration and is therefore necessary for human life. The respiratory system is also responsible for the elimination of toxic waste, temperature regulation, and the stabilization of blood acid-alkaline balance [1].

The respiratory system can be compromised by a variety of factors including bacterial infection, the inhalation of pollutants, genetics, and more. Despite the availability of cost-effective treatments and prevention methods, respiratory disorders are among the most common health problems worldwide. In 2010, the United States alone accounted for 6.8 million pediatric visits for respiratory disorders (Fig. 1) [2].

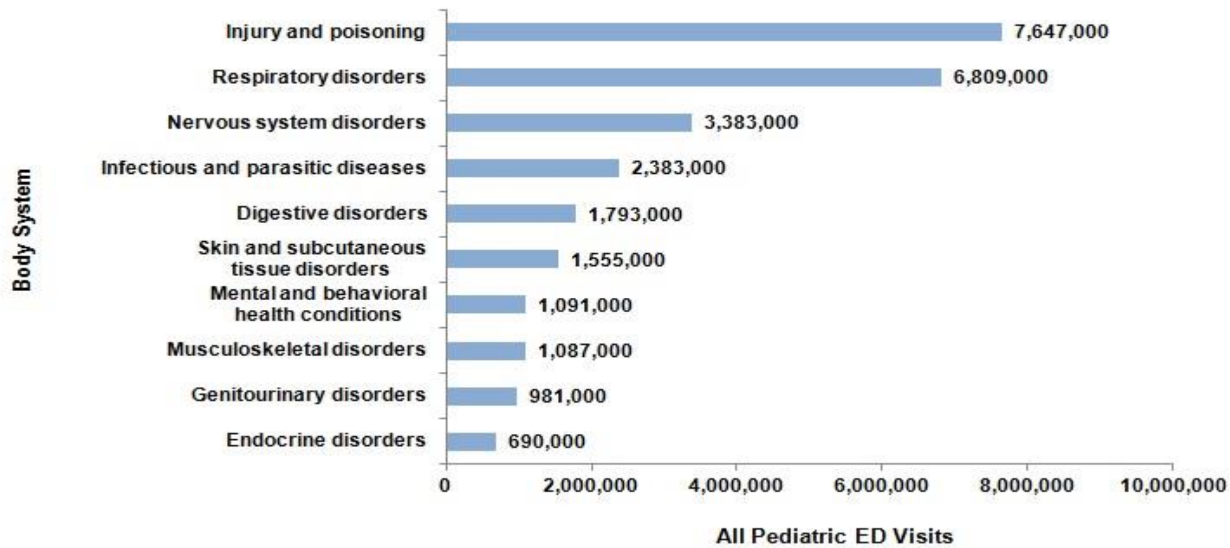

Figure 1: Respiratory disorders are extremely common [3].

In much of the developing world the problem is worse and commonly results in fatalities. This may be attributed to lower air quality, higher rates of toxins, and scarcity of medical resources particularly during times of natural disasters. For this reason, creating respiratory therapies that can be used in the most remote of locations is of great importance.

Treatment

Respiratory diseases make up four of the planet's top ten illnesses affecting humans worldwide, cumulatively claiming nearly 17% of global deaths between Tuberculosis, respiratory cancers, Chronic Obstructive Pulmonary Disease (COPD), and Lower Respiratory Infections (LRI) [4]. Most of these diseases disproportionately affect populations in low-income countries—over 90% of COPD deaths, for example, take place in the poorest parts of the world, according to the World Health Organization [5]. In some cases, these diseases are treated with medications administered via oral steroids or subcutaneous injections; unfortunately, with either of these approaches, medication is typically carried to all parts of the body, inducing harmful side effects including insomnia, weight gain, osteoporosis, high blood pressure, and high blood sugar [6].

Medicines inhaled directly into the respiratory tract, however, offer a more targeted approach, ensuring that high concentrations of aerosol antibiotics are delivered directly to the infected sites. There are two primary methods for administering this form of medication; inhalers, and nebulizers. Inhalers— typically manifesting as small, plastic handheld devices— are usually considered the more convenient and portable option, delivering a single puff of medicine into a patient's airways with each use. Unfortunately, while cost-effective, they are often used improperly, with one study showing that only 5 percent of patients sampled used their Meter-Dosed Inhalers correctly [7]. This problem is especially exacerbated among infected children, elderly people, and disabled individuals, who may need assistance is utilizing a handheld inhaler. These drawbacks in functionality and design can result in wasted medication, limiting the effectiveness of inhalers in treating the diverse scope of respiratory illnesses worldwide [8].

Nebulizers, on the other hand, can serve as a more intuitive and comprehensive tool for combatting a wide variety of harmful respiratory illnesses. Nebulizers are optimized for larger doses of "breathing treatment," with most sessions lasting an average of ten minutes or more, depending on the type of illness being treated [9]. Because many patients have a combination of conditions affecting different areas of their respiratory system (such as COPD and chronic sinusitis) they may also choose to use a nebulizer for their ability to modify the desired particle size, allowing them to deliver larger droplets to their lungs' upper airways, and smaller particles into the lower airways, adjusted at their convenience. Although nebulizers and inhalers are typically viewed as substitutes, they can also complement one another as treatment choices, with many studies showing that supplementary nebulization sessions can lead to better treatment results than using an inhaler alone [10].

Pressure Vessel Overview

Design Introduction

This device is completely powered by a manual pump and regulated with solely mechanical devices. The defining characteristic of the design is a cylindrical pressure vessel. To build pressure in the tank, the patient or individual distributing the treatment must repeatedly apply force to the top of the foot pump. Air will then flow through a hose connected to the tank and begin building pressure. A flap is located at the entrance to the tank to allow air to flow in, but not to exit. Attached to the pressure vessel is a pressure relief valve and a regulator. The pressure relief valve ensures that the pressure inside the tank does not exceed its limitations. The regulator allows air to flow out of the vessel once it reaches a given pressure. After the air leaves the regulator, it will flow into a second enclosure that acts as a diffuser. The combination of the diffuser and pressure vessel works to eliminate the treatment being delivered to the patient in waves caused by the inconsistent frequency of pumping. A nozzle is connected to the diffuser to restrict the outgoing air flow. The desired hose, medicine cup, and breather will attach directly to the nozzle, but will not be included in the overall design seen in Figure 19.

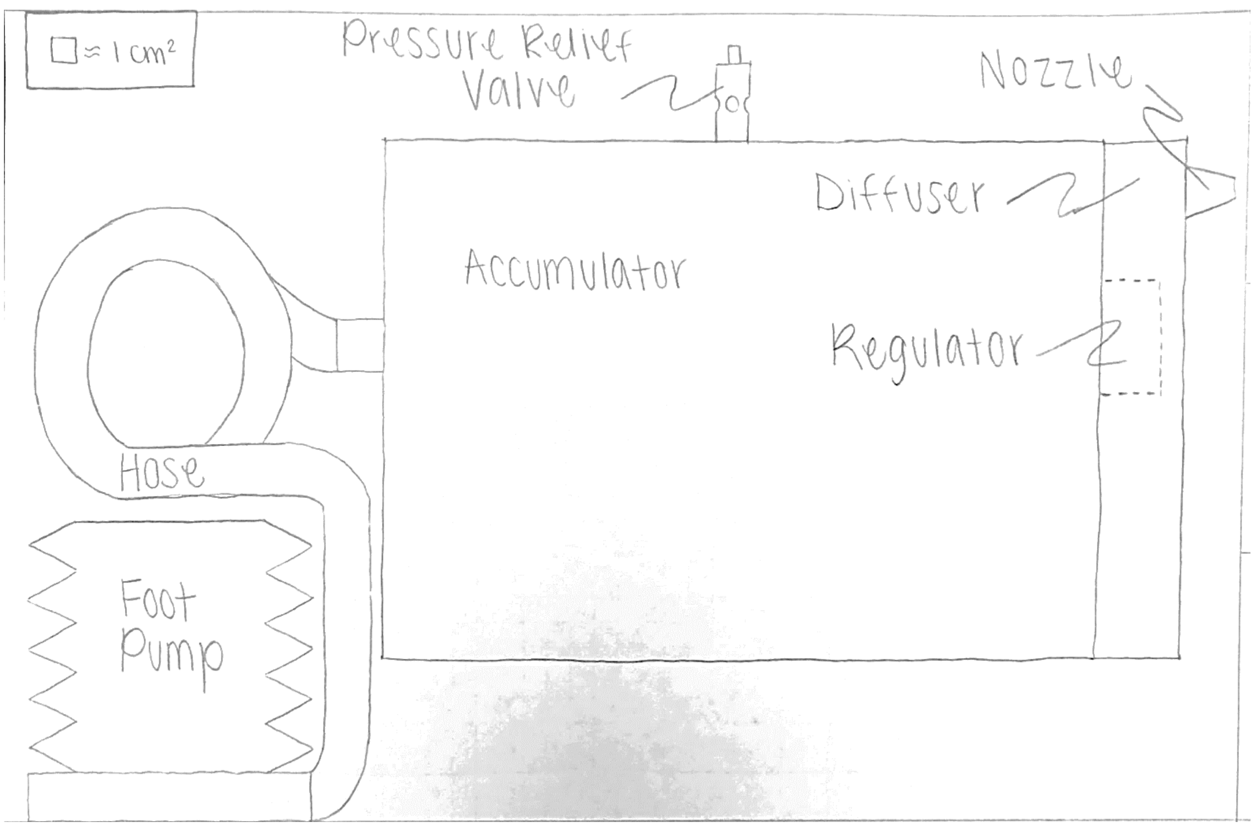

Figure 19: Pressure vessel conceptual design.

Assumptions

The nozzle portion of the tank was designed assuming steady state, isentropic, compressible flow. The air is considered an ideal gas with constant specific heats and a constant specific heat ratio. While this design is expected to operate at different pressures and temperatures the pressure vessel was optimized for operation at normal pressure and temperature, 20^{o} C and 101.325 kPa. All gas and material properties are taken for this temperature and pressure, they are assumed constant for the purposes of calculations. The following list summarizes the assumptions used during calculations.

- Steady State
- Isentropic
- Compressible Flow
- Air can be Modeled as an Ideal Gas
- Normal Temperature and Pressure
- Constant specific heat ratio
- Constant - constant pressure specific heat

All calculations for the pressure vessel design were calculated using U.S. customary units and were converted to metric for the purposes of this report. The values in this section are subject to additional error based on unit conversion.

Design Requirements

The pressure vessel was designed to operate with the Phillips Respironics Side Stream nebulizer. The side stream is paired with the Innospire Essence compressor system; the pressure vessel is designed to mimic the volumetric flowrate and outlet pressure of the Innospire Essence. According to the brochure provided by Phillips the volumetric flowrate is 7 liters per minute and the outlet pressure is 6.8947 kPa [44]. For these conditions the treatment time is listed between five and seven minutes, the pressure vessel was designed for a ten-minute treatment.

Table (VIII) lists the design vales as well as the assumed constants used in the design of the pressure vessel.

Pressure Vessel Design Values

Property	Symbol	Value	Units	Rationale
Volumetric Flowrate	$\dot{\forall}$	7	lpm	Phillips Respironics Brochure
Outlet Pressure	P_b	6.8947	kPa	Phillips Respironics Brochure
Treatment Time	$t_{treatment}$	10	Minutes	Overestimating Phillips Brochure
Ambient Temperature	T	293.15	K	Normal Temperature
Ambient Pressure	P_a	101.325	kPa	Normal Pressure

Table (VIII). Design values for the pressure vessel design. All values were converted from U.S. customary units to metric for the purposes of the report.

Pressure Vessel Components

A. Air Receiver

The air receiver is designed to hold the mass of air required for a full treatment under pressure. The air receiver is designed as a cylinder to better withstand the stresses caused by pressurization. An initial concept design of the air receiver, or accumulator, can be seen in Figure 20.

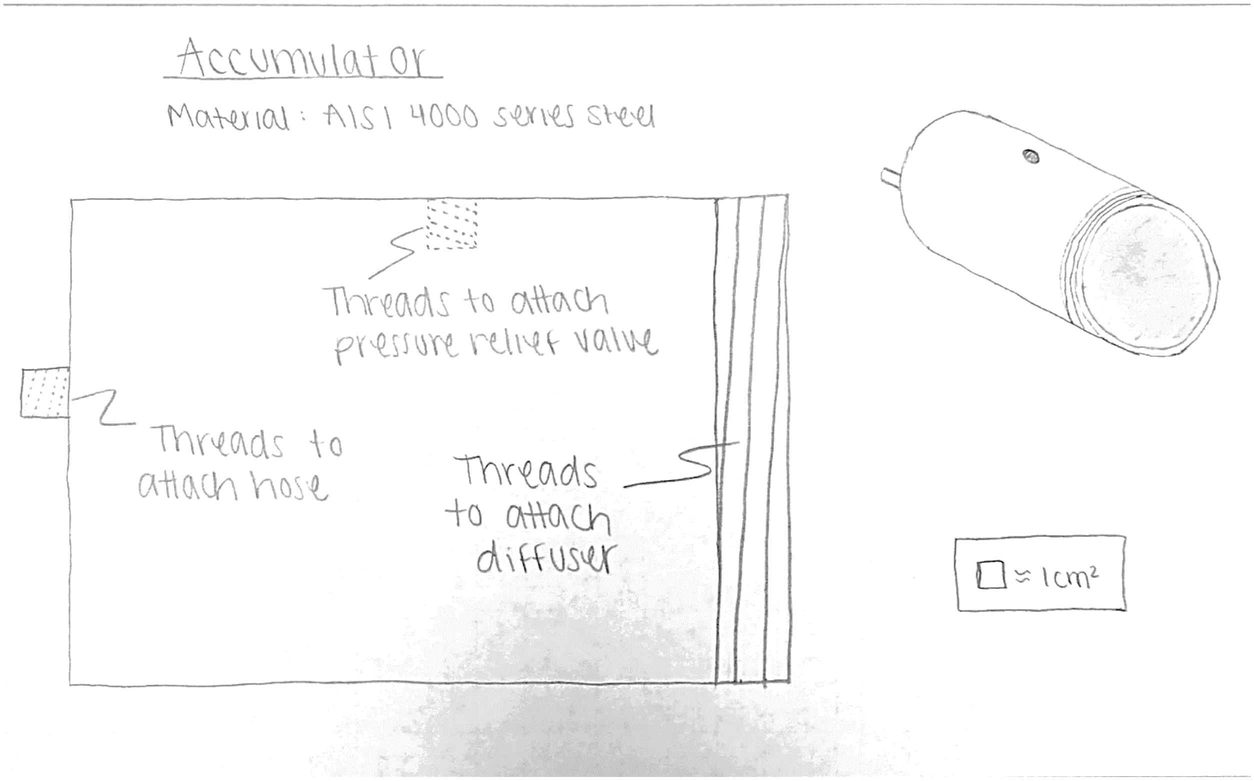

Figure 20: Accumulator design concept.

i. Air Mass Requirement

The pressure vessel was designed so that it could store enough air for a single treatment. The design also allows for less air to be pumped to start treatments, but this will require pumping during the treatment. To determine the total mass of air required for the treatment (22) was used where the volumetric flowrate is 7 liters per minute (0.007 m^3/min), and the density of air at 293.15 ^{O}C is 1.204 kg/m^3.

$$\dot{m} = \rho_{air} \cdot \dot{\forall} = \left(1.204\ \frac{kg}{\cancel{m^3}}\right) \cdot \left(0.007\ \frac{\cancel{m^3}}{min}\right) = 0.0084\ \frac{kg}{min} \quad (22)$$

Approximately 0.0084 kg/min are required to flow out of this device into the Side Stream nebulizer. To calculate the mass of air required for the entire treatment equation (23) was used. The treatment time was estimated to be 10 minutes which is three minutes longer than the specified treatment time, this is to account for user error during the treatment and to allow a generous amount of time for the device to reach steady state operation where the design parameters were calculated.

$$m_{required} = \dot{m} \cdot t_{treatment} = \left(0.0084 \frac{kg}{\cancel{min}}\right) \cdot (10\ \cancel{min}) = 0.084\ kg \tag{23}$$

The mass required to flow out of the device is equal to the amount of mass that must be stored in the tank for a single treatment. This is because the treatment must be able to take place with a single charge. The ideal gas law, (24), was used to create a relation between pressure and volume that satisfies the mass of air that must be stored within the tank. Where the ratio R/M is the specific gas constant and is 287.058 J/kg · K.

$$\begin{aligned} P_T \cdot \forall &= \frac{m_{required}}{M} \cdot R \cdot T \\ &= (0.084\ kg) \cdot \left(287.058 \frac{J}{kg \cdot K}\right) \cdot (293.15\ K) = 1594.44\ m \end{aligned} \tag{24}$$

Figure 21 below uses the result to plot the tank pressure against the volume of the tank.

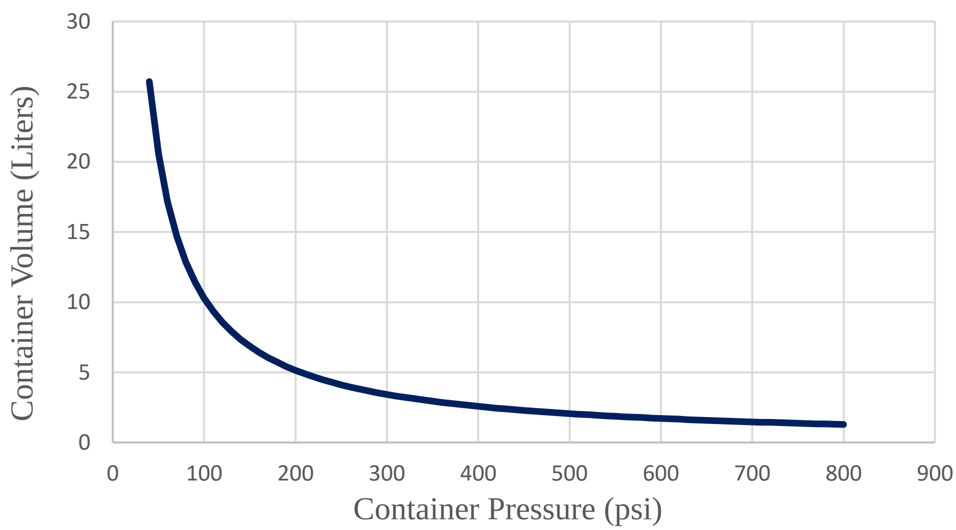

Figure 21: The pressure-volume relation. The relation is done with container pressure in psi and volume in inches cubed. The graph volume is plotted in liters for clarity.

The figure was used to decide the volume and pressure requirements of the tank. A volume of 10 liters was chosen which must be pressurized to 689.476 kPa (100 psi). The above graph only holds true for air modeled as an ideal gas at a temperature of 20 °C.

B. Tank Material and Properties

The initial design for the tank will use an AISI 4000 series steel with a thin lead-free brass alloy used as an inside lining for the tank. The structure of the tank will be provided by the steel alloy, the brass alloy serves as antimicrobial agent to prevent bacteria buildup as the tank is used. Lead free is required as the air will be in direct contact with the lungs.

The tank was approximated as a thin walled pressure vessel and the modified Goodman equation, (25), was used to analyze the dynamic loading on the system. The loading profile on the pressure vessel ranges from no stress at 101.3529 kPa (no gauge pressure) and maximum loading at 689.476 kPa.

$$\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ult}} = \frac{1}{n} \tag{25}$$

With the loading profile the alternating and mean stress are equal and are equal to half the stress on the system created by pressure, (26) solves the modified Goodman equation for the alternating stress, the alternating and mean stress are the same number.

$$\sigma_a = \frac{1}{n} \cdot \left(\frac{1}{\left(\frac{1}{S_e} + \frac{1}{S_{ult}} \right)} \right) \tag{26}$$

The longitudinal and hoop stress were calculated with the thin walled pressure vessel approximation using (27)-(28).

$$\sigma_l = \frac{P_T \cdot r}{2 \cdot t} \tag{27}$$

$$\sigma_h = \frac{P_T \cdot r}{t} \tag{28}$$

The hoop and longitudinal stresses were solved for the thickness required to prevent failure under 10^6 cycles. The larger of the two required thicknesses was used. The lower limit of AISI 4000 steels was used as it is easy to weld. The endurance limit is 137895.15 kPa and the ultimate strength is 4502276.51 kPa [45]. For a radius of four inches the required thickness is 0.001047 m.

The weight of the tank was calculated by multiplying the volume of the material by the density of the material. Equation (29) was used to calculate the material volume of the tank. This equation assumes a length of 0.3048 m (12 in) which is needed to set the volume of the tank equal to the designed volume of ten liters.

$$\forall_{material} = (\pi(r+t)^2 - \pi r^2) \cdot l + (\pi(r+t)^2)2t \tag{29}$$

The volume of the material was calculated using the largest density for the AISI 4000 series steel which is 07750.37 kg/m^3 [45]. The weight of the tank was calculated by multiplying the material volume by the material density (30).

$$W_{Tank} = \forall_{material} \cdot \rho_{material} \tag{30}$$

The mass of the tank was estimated to be a maximum of 20.826 N (4.68 lb) at the largest possible density for AISI 4000 series steel.

To estimate the cost of the steel used in the tank McMaster car AISI 4130 steel is used. The steel costs $42.33 per kg ($19.20 per lb). Multiplying this by the mass of the tank the estimated material cost of the tank is $89.86 [47].

C. Diffuser

The diffuser is designed to stagnate the flow or air out of the regulator. An initial design concept of the diffuser can be seen in Figure 22. To estimate the size and material cost of diffuser, it is designed to be a shell around the volume of the regulator. The diffuser length is 0.028575 m (1.125 in). The radius is the same as the tank, but the thickness is decreased as the tank only has to hold 103.421 kPa (15 psi). The same thickness formula used for the receiver was used for the diffuser, the calculated thickness is 0.0015 m (0.006 in). The mass of the diffuser is 0.965 N (0.217 lbs). The diffuser material is the same material as the tank, so they can be welded together. The estimated material cost of the diffuser is $4.17 using the same material as the receiver.

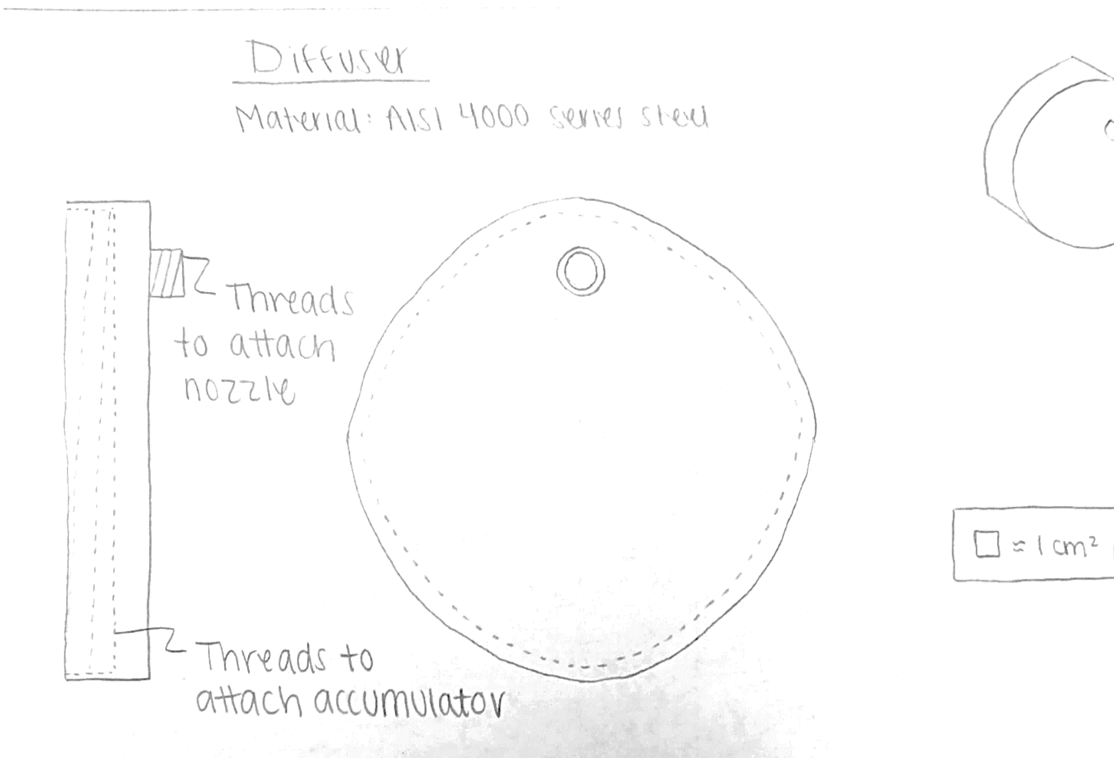

Figure 22: Diffuser design concept.

D. Nozzle

The outlet area is determined volumetric flowrate and the air velocity. The volumetric flowrate equation was manipulated to calculate the outlet velocity of the air based on the area (31).

$$\vec{V}_{air} = \frac{\dot{\forall}}{A} \tag{31}$$

This equation was plugged into the Mach number equation to solve for the required outlet area in terms of Mach number (32). The only criteria placed on the Mach number at this point is that it must be less than one to be considered subsonic flow.

$$A_o = \frac{\dot{\forall}}{Ma_o \cdot C_{Air}} \tag{32}$$

Both the speed of sound and the volumetric flowrate are set quantities regardless of the design, so they were plugged into the equation yielding (33). The speed of sound through air at 20 °C is 343.21 m/s and the volumetric flowrate is 7 liters per minute (0.0001167 m^3/s).

$$A_o = \frac{1}{Ma_o} \cdot (3.39928 \cdot 10^{-7}\ in^2) \tag{33}$$

Table A-13 in [46] can be used to determine the critical area ratio A_o/A^*. The outlet critical area ratio A_o/A^* is determined by the outlet Mach number and the inlet critical area ratio A_i/A^* is determined by the inlet Mach number. The inlet Mach number is designed to be low to achieve properties close to the stagnation properties. Equation (34) was derived using equations in [46] where A_R is called the area ratio and is equal to A_i/A_O, and the left-hand term is a function of the internal Mach number.

$$\frac{A_i}{A^*} = \frac{A_o}{A^*} \cdot A_R \tag{34}$$

i. Pressure Considerations

The following equation must hold true for nozzle to be under subsonic flow and avoid choked flow through the nozzle where P^*/P_o is the critical pressure ratio. For the following equations P_i is the inlet pressure, P_o is the inlet stagnation pressure and P_b is the outlet pressure. The stagnation pressure is approximated to be equal to the inlet pressure as the velocity is designed to be low.

$$\frac{P_b}{P_o} > \frac{P^*}{P_o}$$

$$P_b > P^*$$

The critical pressure ratio is determined by the stagnation pressure and the specific heat ratio of air. Equation (35) was used to determine a relationship between the back-pressure and the inlet pressure.

$$\frac{P^*}{P_o} = \left(\frac{2}{K+1}\right)^{\frac{K}{K-1}}$$

$$\frac{P_b}{P_i} > \left(\frac{2}{1.4+1}\right)^{\frac{1.4}{1.4-1}}$$

$$\frac{P_b}{P_i} > 0.5283 \qquad (35)$$

P_b is pressure at the outlet and is 10 psi. The ratio between P_b and P_i determines the Mach number at the outlet. Figure 23 shows how the determination of the inlet pressure and Mach number define each of the other elements in the nozzle calculations above.

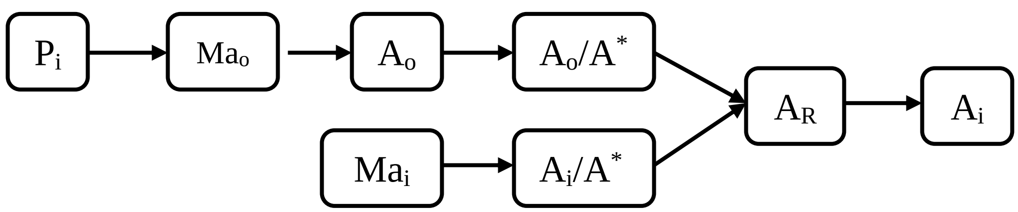

Figure 23: The process by which the inlet pressure and Mach number determine nozzle properties.

E. Regulator

The regulator will be designed to operate as a valve set to a specific pressure when open the regulator will create a set pressure in the diffuser that allows the subsonic nozzle to work. When closed no fluid flow will take place. In the final stages of this design the regulator will be designed however to estimate the cost and size of the regulator an off the shelf regulator is specified. A regulator from McMaster-Carr was specified with an outlet pressure range between 0 and 30 psi. The regulator costs $35.07 [48]. The volume occupied by the regulator is $6.22 \cdot 10^{-5}$ m^3. The regulator is estimated to weigh 2 N based on Cad files provided by McMaster-Carr. The regulator used must be a dual stage regulator to maintain the same output pressure as the air receiver is depleted [49].

F. Pressure Relief Valve

The air tank falls under ASME Boiler code and is required to have a pressure relief valve. An off the shelf valve that has a burst pressure of 125 psi was specified to meet this requirement. The valve costs $9.19 and weighs approximately 0.36935 N [50].

G. Check Valve

The check valve is attached to the inlet of the air receiver. A flap valve will be used, a valve from McMaster-Carr is used to estimate the cost of the device. The check valve will cost $11.48 and the weight is considered negligible [51].

H. Foot Pump

The foot pump needs to provide enough mechanical advantage to allow a user to continue adding air to the tank at 100 psi. For the final design the pump will be designed but a pump was cited from McMaster-Carr to estimate costs. The pump costs $32.40 and weighs approximately 0.25 pounds or 1.11 N [52]. The size estimate of the pump also uses the pump from McMaster-Carr and is approximated as a 5.08 cm radius tube that is 60.69 feet tall. The size of the foot pump is estimated to be 4917.84 cm^3.

Energy Requirements

The only energy input this system requires is the that needed to pressurize the vessel. The amount of energy a user has to input is assumed to be equal to that required to compress air to the required pressure of 689.476 kPa. Equation (36) was used in order to calculate the required input energy.

$$E = n \cdot R \cdot T \cdot \ln\left(\frac{P_o}{P_i}\right) \tag{36}$$

The total amount of input energy required is 1.3 kJ. The average person can put between 50 W and 140 W. Equation (37) can be used in order to calculate the required charge time based on the input energy.

$$T_{Charge} = \frac{E}{Watts} \tag{37}$$

At the lowest input energy of 50 W the tank will take 26 minutes to fill, at the highest input of 140 W the tank will take 9.32 minutes to fill. For the purposes of the decision matrix the estimated input energy was taken from the lowest value of 50 W for which the estimated charge time is 26 minutes.

Design Summary

The following two tables summarize the decision matrix parameters for both the power and the compressor system. For the purposes of the decision matrix the foot pump is considered the power source and the rest of the design is considered the compressor system. The required power for the compressor system is assumed to be 1 W which is the power required to turn the T-Ball valve into the open position. The compressor system is composed of eight components and the power system is composed of ten components.

Power System		
Objective	Value	Units
Cost	32.40	Dollars
Required Power	50	Joules/s
Weight	0.113	Kg
Size	4917.83	cm^3
Charge Time	26.0	Minutes
Complexity	10	Components

Table (IX). Summary of decision matrix parameters for the power system.

Compressor System		
Objective	Value	Units
Cost	149.77	Dollars
Required Power	1	Joules
Weight	2.26	Kg
Size	10800	cm^3
Complexity	8	Components

Table (X). Summary of decision matrix parameters for the compressor system. Weight was calculated by dividing the weight by 9.81 m/s^2.

Standards, Codes, and Main Sources Used

The nebulizer was designed following relevant sections of both international and domestic codes and standards. The air tank was designed following the ASTM Boiler Code to ensure the safety of the patient or personnel operating around it. Material selection for the air tank was done referring to both section II and VIII of the boiler code as well as ISO 10524-1. Materials for all other components where medical or food grade and compliant to ISO 10524-1. Specific section of codes or standards are cited when relevant for the design of a particular part. The Machinery's Handbook was vastly used to determined tolerances or standard specifications for threads. The parker O-ring handbook was utilized for determining seals in the piston of the pump. Shigley's Mechanical Design textbook was extensively used for theoretical fatigue and stress analysis of components. General material properties used for material selection where pulled, primarily, from Matweb as this source has a vast library of materials and tempers. However, values listed there were compared to properties listed in manufacturer's websites to ensure accuracy.

Product Key Features and Functionality

The deciding factor that highlighted this concept's over the other two, when performance matrices were done, was the simplicity of the design. Pressurized air allows the use of fewer parts – especially moving ones - which in turn means less possibility of failure. Reliability once again was deemed highly important due to the environment the product is aimed at operating in. For this reason, the device has relatively few components when compared to the other two concept designs.

The design can be split into two major parts. The charging system and the delivery system. The charging system consists of a foot pump. It should be able to pump upwards of 700 kPa. Though multiple components of the nebulizer were chosen to be purchased instead of manufactured, the pump was designed to meet the desired specifications while ensuring reliability. A foot pump was chosen as this allows the user to use his or her own full weight to compress a single piston and charge the tank to 700 kPa. The pump was designed for mechanical advantage. A lever arm is used to facilitate the operation of the pump when reaching the higher-end pressures the tank will be charged to. These will act back on the piston; the use of a lever will allow a smaller individual to be able to pump the device if need be. The pump will then be connected to the air tank with a flexible hose, allowing to locate the foot pump in comfortable position without having to drag the entire device. Furthermore, upon reaching higher pressures, if the foot pump were to move due to a loss of balance of the user the whole nebulizer will not be at risk of being turned over and damaged because of the pump's physical displacement. At the other end of the hose, a one-way check valve will control flow direction, allowing air from the pump to go into the tank only.

Past the one-way check valve, a cross fitting is utilized to allow all valves and connection to be made in one location. Moreover, the use of a cross prevents the need for multiple hole to be drilled into the pressure vessel, reducing the number of areas where stress concentrations would occur, and thus the risk of failure. The cross will be connected to the air tank via a solid line and a tee fitting. A gauge to read the tank pressure will be attached to the third corner of the tee fitting. An aluminum inlet/outlet piece with male threads will be manufactured on a lathe to avoid using a male-male fitting between the tank and the cross.

Back on the cross fitting, a third side of the fitting will have a pressure relief valve installed to ensure the system will not be over pressurized past the intended design pressure. On the forth corner of the cross a butterfly valve, pressure relied valve and diffuser are attached in series. The butterfly valve acts as an on/off switch. It is manually operated and fully mechanical, negating the need for electronics of any kind. This was done both for cost and, once again, to reduce the chance of failure. Furthermore, the repair of any of these simple mechanical devices would be much easier to perform given limited resources compared to electrically controlled components.

A pressure regulator reduces the tank pressure down to the same operating pressure of the Philips nebulizer design; since this design's air delivery specifications is to match that of the Philips device. Following a regulator, a diffuser reduces the velocity of the air flow to prevent harm or discomfort to the patient and match the desired flow rate. From here, the air flows through a tube, into the medicine cap, and finally into the mouth piece.

Both the medicine cap and mouthpiece of the Philips nebulizer (or similar design) are to be used. Connecting all the valves are male to male fittings. ¼" NPT fittings were used throughout. Apart from the diffuser, all valves or fittings cited above shall be purchased at an adequate supplier. McMaster is quoted (unless specified otherwise) in the rest of the report for parts and pricing. Finally, the air tank stores the air to be used later to atomize the medicine. Volume was set to provide one full medicine delivery session on a full charge. The tank shall be manufactured, material selection and manufacturing details will be discussed later. Figure 40 shows a schematic of the pressure tank subsystem.

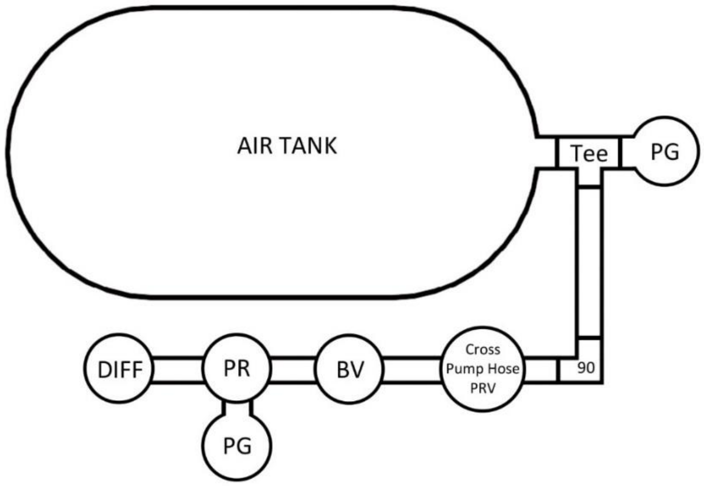

Figure 40: Nebulizer schematic. Top view of the system. PG = pressure gage, Tee = T\tee fitting, 90 = 90° fitting, BV = buttefly valve, PR = pressure regulator, DIFF = diffuser, Cross = cross fitting, PRV = pressure releif valve. The PRV is located out of page, the pump hose attaches on the underside of the cross fitting.

OTS Parts

Cross Fitting

In order to reduce the number of fittings necessary and the overall size of the design, a brass cross pipe fitting was selected to connect the foot pump and the pressure relief valve to the system along with joining the tank to the butterfly valve. The fitting selected can be purchased from McMaster-Carr for $17.56. It is rated for a maximum pressure of 1.4 MPa, well above the maximum pressure this device will house. The fitting is manufactured completely out of brass, it is compliant with the ANSI Standard 61 for drinking water and the Restriction of Hazardous Substances directive (RoHS) [1]. Table XVII lists all OTS parts, part number, quantities, and pricing.

Miniature Check Valve

A backflow prevention valve is necessary to ensure that the air being supplied from the foot pump into the pressure vessel does not leave the tank upon entry. This component will be located between the hose coming from the foot pump and the brass cross fitting. The valve selected can be purchased from McMaster-Carr for $19.73. A maximum pressure limitation of 1.4 MPa over-compensates for the pressure vessel's maximum intended pressure of 700 kPa. The valve is manufactured out of brass and stainless steel. It is RoHS compliant and is to be used for water or air [2].

Pressure Relief Valve

Since the ISO 10524-1 standard 6.10 requires that pressure vessels be regulated by a pressure relief valve (PRV), the design includes a PRV set to 724 kPa attached to the brass cross pipe fitting. This relief valve allows for the pressure tank to reach the pressure of 700 kPa necessary to produce one complete ten-minute treatment. Once the pressure reaches the set pressure of 724 kPa, the valve opens and begins to release pressure well before the tank's maximum possible pressure of 2.8 MPa is would be attained. The selected valve is manufactured out of brass and includes a silicone seal. McMaster-Carr sells this specific valve for $5.26. The PRV complies with ASME code VIII for air and inert gases [3].

Brass Ball Valve

An on/off valve is included in the system for the operator to be able to cut off the flow out of the tank while trying to build pressure. When the necessary pressure is reached, the operator can open the valve and allow for air to flow to the patient. The selected valve can be purchased from McMaster-Carr for $8.27. It is manufactured out of brass and PTFE [4]. In the design, the valve is located in between the brass fitting and the pressure regulator.

Regulator

A regulator is necessary to regulate the outlet pressure of the air flowing from the tank. The regulator selected can be purchased from McMaster-Carr for $154.98. The body of this regulator is manufactured out of brass and has a maximum inlet pressure of 1.7 kPa [5]. This particular regulator does not include a pressure gauge. According to ISO 10524-1 standard 6.2.1, all pressure regulators must include gauges [6]. A compatible pressure gauge can be purchased from McMaster-Carr for $10.25 [7]. The selected gauge has a pressure range from 0-103 kPa which complies with the ISO 10524-1 standard 6.2.2.6 which states that all pressure gauges must read up to 133% of the intended pressure (70 kPa) [8]. A second pressure gauge is located before the on/off valve so that the operator can see the tank pressure. This gauge has the exact same specifications as the first but can withstand a pressure of up to 1.1 MPa [6].

Fittings

A tee fitting is connected to the pressure tank inlet and attaches the high pressure gauge and a 5.5” pipe. The selected tee fitting is manufactured out of brass and can be purchased from McMaster-Carr for $8.36 [9]. An elbow fitting is utilized to wrap the train of valves required along the side of the pressure tank. The selected elbow is manufactured out of brass and can be purchased from McMaster-Carr for $7.13 [10]. Both the tee and the elbow fitting can withstand a maximum pressure of 1.4 MPa and are compliant with the ANSI standard 61 for drinking water. Two male hex nipples function as thread male-to-male thread adapters for connections from the elbow fitting to the brass cross and from the brass cross to the brass ball valve. The selected hex nipples are manufactured out of brass and can be purchased from McMaster-Carr for $2.30 each [11].

Piping

The pipes used in the devices are seamless 230-series bronze. They are compliant with ANSI Standard 61 for drinking water and RoHS. The cost of each pipe is $2.77, $3.53, and $4.52 for the 76.2 mm, 102 mm, and 140 mm long pipe respectively [12]-[14]. Stress for the pipe undergoing 700 kPa was calculated to ensure the wall thickness is sufficient.

CALCS

Valve and Fitting Costs

Description	Manufacturer Part Number	Quantity	Cost ($)
Cross Fitting	44299K351	1	17.56
Check Valve	8567T21	1	19.73
PRV	48435K72	1	5.26
On/off Valve	47865K21	1	8.27
Regulator	4677K61	1	154.98
90° Elbow	4429K161	1	7.13
Tee Fitting	4429K251	1	8.36
Hex Nipple	5485K22	2	2.30
76.2 mm Pipe	4568K135	1	2.77
102 mm Pipe	4568K137	1	3.53
140 mm Pipe	4568K351	1	3.38
Gauge 0-15 psi	4089K61	1	10.25
Gauge 0-160 psi	4089K61	1	10.25
	TOTAL		253.77

Table (XVII). Costs for valves and fittings

Key Features and Analysis

Air Tank

The pressure vessel that will store compressed air will be designed following the ASTM Boiler code. Material selection, wall thickness, weld sizes, inlet design and prefabricated parts shall be determined utilizing their respective subsections and parts in the code. Following the loadings that need to be considered listed in section UG-22, the following points shall be touched: stresses due to internal pressure, the use of legs following the non-mandatory Appendix 6, cyclic loading, and differential thermal expansion. The vessel being designed will not have any external loading; the weight of the vessels and its contents during operation shall not be considered given its dimensions, weight, and application; cyclic, dynamic, static, or mechanical loadings due to equipment mounted on the vessel will also be disregarded as not external device shall be mounted on the tank. Due to the sanitary nature of the device materials that are food safe and in accordance to FDA class II code shall be utilized.

1. Tank Material Selection

These two sections are combined as the method for manufacturing the tank plays a role in the material selection process. The tank will not be highly pressurized, yet the boiler code states that for "compressed air service the minimum thickness of shells and heads shall be 2.5 mm" [37]. Therefore, high-strength steels will not be used as this would result in an over-designed and overly expensive air tank. A more cost-effective steel with lower strength may be desired to reduce both cost and manufacturing time. Sheet metal of the required thickness (determined bellow) shall be purchased from a retailer, cut to size, rolled, and welded. The ends of the tank are hemispheres as they provide the optimum stress distribution. These will be purchased pre-made from a manufacturer and welded to the cylindrical body. A threaded inlet/outlet piece will be welded to one of the hemispheres. The design of this piece is discussed later.

The preliminary design concept called for a AISI 4000 series steel shell with a brass coating on the inside to take advantage of copper's antimicrobial properties. The original design has a wall thickness of 1.0 mm that resulted in a factor of safety of 4. The wall would however not be up to code. Increasing the thickness to 2.5 mm would drastically increase the cost and difficulty of manufacturing as stated above. Additionally, the use of a coating is not desired and is to be avoided if possible as it would entail approximatively a week-time to have it applied by a specialized shop. The 4000 series steel with no coating would be exposed to the environment. 4000 series steels are not corrosion resistance. Humidity in the air and condensation caused by pressure changes in the tank would inevitably lead to corrosion. Moreover, due to the lack of accessibility to the inside of the tank for cleaning and the need for the device to run without maintenance anyhow, corrosion build-up on any surface in contact with the air to be inhaled is unacceptable. The tank holds the air that would later be breathed in by the patient, therefore the device would not pass FDA class II codes. Furthermore, 4000-series steel is hard to weld, requiring preheating of the metal prior to welding.

A second option would be 304 stainless steel, which is found in the table of allowable materials in the ASTM Boiler code manual. The corrosion resistance properties of this alloy are ideal for an uncoated tank; A304 is food safe, widely used in industry such as food processing plants and medical equipment. Additionally, A304 is can be welded. However, with a minimum wall thickness of 2.5 mm - two and a half time larger than the minimum thickness calculated with (57) - the weight of the tank would have a mass of 4.9 kg.

The third option would be to select a metal alloy that is weaker and lighter than steel so that the minimum thickness calculated with equation (57) is equal to or slightly above 2.5 mm as to not have excess material. Aluminum alloys with good weldability were investigated as they fit this requirement and 5052 alloy was selected as it fits the design requirements and is widely sold in sheets or plates. Furthermore, 5052 aluminum has good corrosion resistance and would not see a decrease of its material properties in the temperature range the air tank is expected to operate in. The material is available commonly in three treatments: O-temper (annealed), H112 (strain hardened), or H32 (strain hardened and stabilized) [38]. The annealed treatment yields the softest material thus improved machinability however, it has the lowest yield strength and tensile strength. Comparing H112 to H32 5052 aluminum, their thermal properties are identical as are their material properties except for fatigue strength and yield strength were the H32 5052 alloy is about twice as strong [39].

2. Wall Thickness and Fatigue Analysis

The minimum thickness for a pressure containing vessel as given by the ASTM Boiler code for a cylindrical shell shall be greater than the largest of the two thicknesses calculated with (57) or (58), where t is the thickness, P is the inside pressure, R is the inside radius, S is the maximum allowable stress, and E is the joint efficiency [40]. Equation (57) is for "circumferential stress" while (58) is for "longitudinal stress". The maximum allowable stress for 5052 aluminum was pulled from Table 1B found in Section II of the ASTM Boiler Code [41]. The operating temperature of the aluminum was taken to be 100°C as this yielded the lowest allowable stress while remaining in a relatively accurate temperature range for this device. The joint efficiency was pulled from Table UW-12 found in section VIII of the ASTM Boiler Code [42]. This value is unitless. It depends on the weld used and examination performed post-weld. A single butt joint weld was selected with no radiographic examination after the fact. These variables yield the smallest joint efficiency value, if double butt joint welds or any examination was done, the value would increase thus decreasing the minimum thickness. Calculations bellow are for 5052 H32 aluminum.

$$t = \frac{PR}{SE - 0.6P} \tag{57}$$

$$t = \frac{(0.700\ MPa)(0.102\ m)}{(61.1\ MPa)(0.60) - 0.6(0.700\ MPa)} = 1.96E(-3)m$$

$$t = \frac{PR}{2SE + 0.4P} \tag{58}$$

$$t = \frac{(0.700\ MPa)(0.102\ m)}{2(61.1\ MPa)(0.60) + 0.4(0.700\ MPa)} = 9.66E(-4)m$$

For a spherical shell the minimum thickness equation as given by the ASME boiler code is (59). The variables t, P, R, S, and E remains the same for this equation as for the previous two.

$$t = \frac{PR}{2SE - 0.2P} \tag{59}$$

$$t = \frac{(0.700\ MPa)(0.102\ m)}{2(61.1\ MPa)(0.60) - 0.2(0.700\ MPa)} = 9.72E(-4)m$$

The annealed aluminum has a maximum allowable stress of 43.6 MPa, or 70% that of the strain hardened alloy. This resulted in a minimum wall thickness of 2.77 mm. For both cases, due to US suppliers providing sheet thicknesses in inches, the nearest size would need to be 2.54 mm (0.1") for the H32 and 3.175 mm (0.125") for the O-temper. For these two thicknesses the weight of the tank would be 1.6 kg and 2.1 kg respectively. However, the code states that for material with undersized tolerances, it must be taken into account the possibility for the material to be thinner than expected. The manufacturer tolerance where the material would likely be purchased next semester is ±0.127 mm (0.005"). Taking this into account, the 2.54 mm sheet would no longer be up to code. The thickness of the sheet would therefore need to be stepped up one size increment to 3.175 mm regardless of the treatment chosen. Lastly, the text specifies that the thickness is after forming [37]. The final minimum tank thickness was calculated to ensure it was up to code. The area of the cross-section of sheet metal needed is A_s and is determined by the thickness of the sheet (t), Δt is the manufacturer tolerance, and the length (l) needed to have an inside tank radius of 101.6 mm (60). A_t is the cross-sectional area of the tank and is equal to A_s as no material is lost in the shaping process. The area of the tank is that of ring (61). Equations (60) and (61) are then solved to determine the resulting outside radius after forming the tube. Subtracting the inside radius to the outside radius yields the tank wall thickness (t_t).

$$A_s = l(t - \Delta t) = (2\pi r)(t - \Delta t) \tag{60}$$

$$A_t = \pi(R^2 - r^2) \tag{61}$$

$$A_s = A_t$$

$$R = \sqrt{\frac{A_s}{\pi} + r^2} = \sqrt{\frac{2\pi(101.6\,mm)(3.175\,mm - 0.127\,mm)}{\pi} + (101.6\,mm)^2} = 104.6\,mm$$

$$t_t = R - r = 104.6\,mm - 101.6\,mm = 3.0\,mm$$

The minimum tank wall thickness will therefore be within code if the selected sheet metal thickness is 3.175 mm. 5052 aluminum with a H32 strain hardened alloy is selected as the material of choice as price difference is negligible and the stronger material will yield a higher factor of safety, has 10% improved fatigue resistance over the annealed one, and weight is not altered.

Because the device will be able to deliver one full treatment for every full charge, one cycle equals one treatment. It takes at least 40 minutes to fully charge and 10 minutes to discharge. This system does therefore have a very slow cycle. Fatigue might not even be a concern in most of the components of this nebulizer design. High cycle is said to start when a device endures 10^3 stress cycles (or one thousand uses) in its life time; infinite fatigue life is said to start at 10^6 cycles. Let one suppose the device is to be used once an hour every day. This would entail that it would fail after a month and eleven days or 114 years of use for 10^3 and 10^6 cycles respectively. The pressure vessel was approximated as a thin walled pressure vessel. The modified Goodman criterion is given in (25), where S_f is the fatigue strength, S_{ut} is the ultimate tensile strength, σ_a is the amplitude stress, σ_m is the midrange stress, and n is the design factor of safety [43]. The stress for the air tank is modeled as sinusoidal fluctuating stress, where the minimum stress due to a 70 kPa pressure exist when discharged, and the maximum stress due to 700 kPa occurs when fully charged. When discharged a 70 kPa pressure remains in the tank as the regulator will be set to this pressure. Unless the PRV is manually opened, said pressure will remain.

$$\frac{\sigma_a}{S_f} + \frac{\sigma_m}{S_{ut}} = \frac{1}{n} \tag{25}$$

Aluminum alloys do not have an endurance limit, instead they have a fatigue strength, S_f. For 5052 with an H32 temper the fatigue strength for 10^8 cycles of reversed stress is 117 MPa [44]. The maximum and minimum stresses are calculated utilizing the equations for hoop and longitudinal stresses in a thin walled pressure vessel, (28) and (27) respectively, where P is the pressure. Results are tabulated in Table XVIII.

$$\sigma_h = \frac{Pr}{t} \tag{28}$$

$$\sigma_l = \frac{Pr}{2t} \tag{27}$$

The amplitude stress and the midrange stress are given by (62) and (63). Results are also tabulated in Table XVIII.

$$\sigma_a = \frac{|\sigma_{max} - \sigma_{min}|}{2} \tag{62}$$

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} \tag{63}$$

Stress Values of Tank

Pressure	σ_h	σ_l	σ_a	σ_m
700 kPa	23.7 MPa	11.9 MPa	-	-
70 kPa	23.7 MPa	1.19 MPa	-	-
N/A	-	-	11.3 MPa	12.4 MPa

Table (XVIII). Values of stress for the tank.

The factor of safety in (25) was thus calculated with the value from Table XVIII. The ultimate tensile strength of 5052 H32 aluminum was taken from the ASTM Boiler Code Section II D Table 1B, which also matched the value in Shigley's Table A24.

$$\frac{11.3\ MPa}{117\ MPa} + \frac{12.4\ MPa}{235\ MPa} = \frac{1}{n}; n = 6.7$$

3. Inlet Piece

The inlet piece was designed in accordance to the ASME Boiler Code Section VIII UG-16b that states that for a pressure retaining component the minimum wall thickness must be 1.5 mm. To standardize the design and reduce the need of fittings, the part will have a ¼ NPT male thread. The minimum wall thickness along the side of the inlet shall thus be dictated by the thread specifications. The minimum thickness will then be the distance from the outside surface to the root of the thread. Stress calculations to ensure this thickness is within the design factor of safety follow. When all valves are closed the pressure inside the tank would act on all surfaces. All resultant forces would cancel on all surfaces but the one directly opposed to the inlet as shown in Fig. 41. The maximum tensile stress on the inlet would occur when the tank is fully charged. The maximum resultant force is calculated with (64). Since the fittings used are ¼ NPT on all corners, forces acting on the valves (and thus the cross) are taken to be equal. A_f is the area of the ¼" inlets/outlets of the valves.

$$F = PA_f = P(2\pi r) \tag{64}$$

$$F = 2\pi(700\ kPa)(3.175\ mm)^2 = 44.3\ N$$

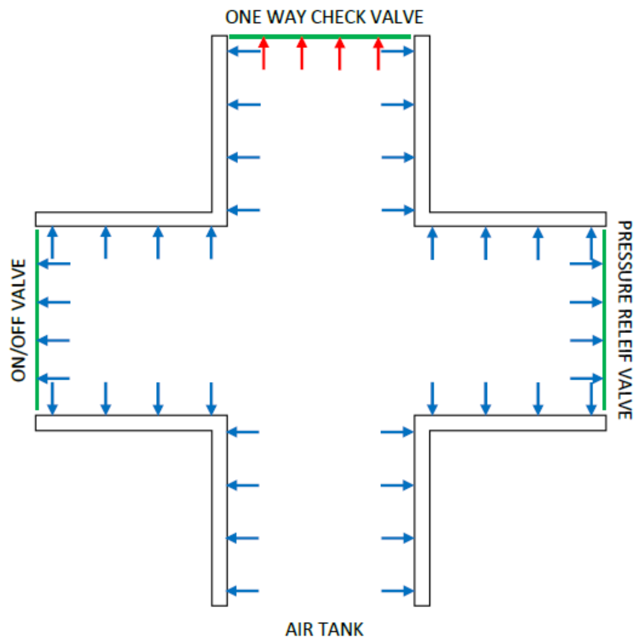

Figure 41: 2D cross-section of cross fitting. Equal pressure acting on inside walls illustrated by arrows. Valves illustrated by green line.

The factor of safety is given by (65). Given a design factor of safety of 4, the minimum cross-sectional area was calculated. This cross-sectional area exists where the NPT threads are cut. At the root of the threads a stress concentration factor K exists which is a factor of the radius of the root. The root radius was calculated to be 0.12 mm based on the geometry of a ¼ NPT threads shown in the Table 2 of the Machinery's Handbook [45]. The stress concentration factor was then determined to be 3 based on research done by the US army on stress concentrations in screw threads [46]. The yield strength of the material was pulled from the Mc Master. 2024 T351 aluminum is utilized here as 5052 is not available in bar stock form. 2024 aluminum has good machinability and like 5052 has good weldability.

$$n = \frac{\sigma_y}{K\sigma} = \frac{\sigma_y}{K\frac{F}{A}} \tag{65}$$

The minimum cross-sectional area of the NPT thread at the root of the thread is given by (66). Where the inside radius is that of the inlet passage, and the outer radius is approximated to be that of the minor diameter (as specified for 18tpi NPT threads in the Machinery's Handbook). The outer diameter was rounded down while the inner diameter rounded up to obtain the most conservative result.

$$A_{tension} = \pi(r_{outer}^2 - r_{inner}^2) \tag{66}$$

$$A_{tension} = \pi((5.61\;mm)^2 - (3.19\;mm)^2) = 66.9\;mm^2$$

$$n = \frac{255\ MPa}{3\frac{44.3\ N}{(66.9E-6)\ mm^2}} = 96$$

Next, the stress due to shear of the thread will be calculated. The force utilized will be the same as for the above calculations. It will be assumed that all the force acts on one thread. The shearing area of a thread is approximated as the thickness of the thread at the base times the circumference of the circle at that location. The thickness of the thread will be approximated as the pitch of the thread minus the “width of flat”. Values are once again collected from the Machinery’s Handbook. The factor of safety will be calculated with (65), where K is taken to be 1 and the area is calculated with (67).

$$A_{shear} = 2\pi rt \tag{67}$$

$$A_{shear} = 2\pi(5.16\ mm)(1.41\ mm\ -\ 0145\ mm) = 41.0\ mm^2$$

$$n = \frac{255\ MPa}{\frac{44.3\ N}{41.0E(-6)\ m^2}} = 236$$

From the results above, the part will not fail in tension when loaded under the predetermined parameters. The threads will not shear under the force generated by the 700 kPa tank pressure. The fatigue life of the part will now be determined. The loading cycle is the same as for the air tank. FEA analysis was performed on the part. A 44.3N was applied on one end to simulate the force generated by the compressed air.

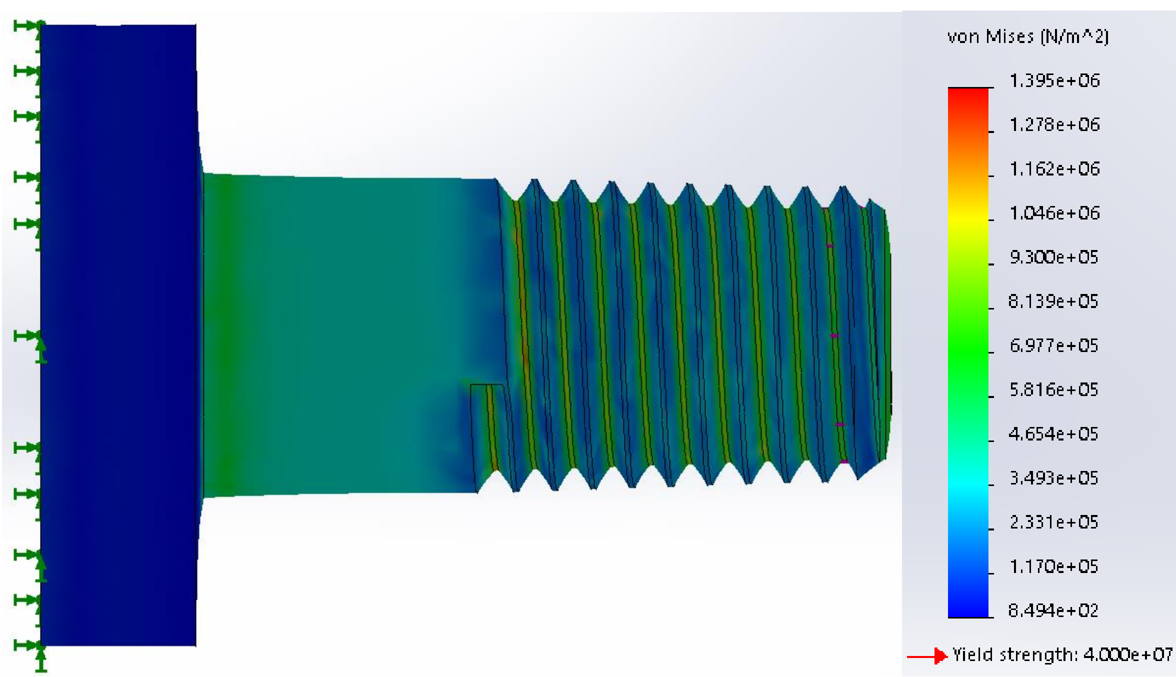

Figure 22: FEA analysis of inlet piece, fixed on the left, 44.3N force on the other end

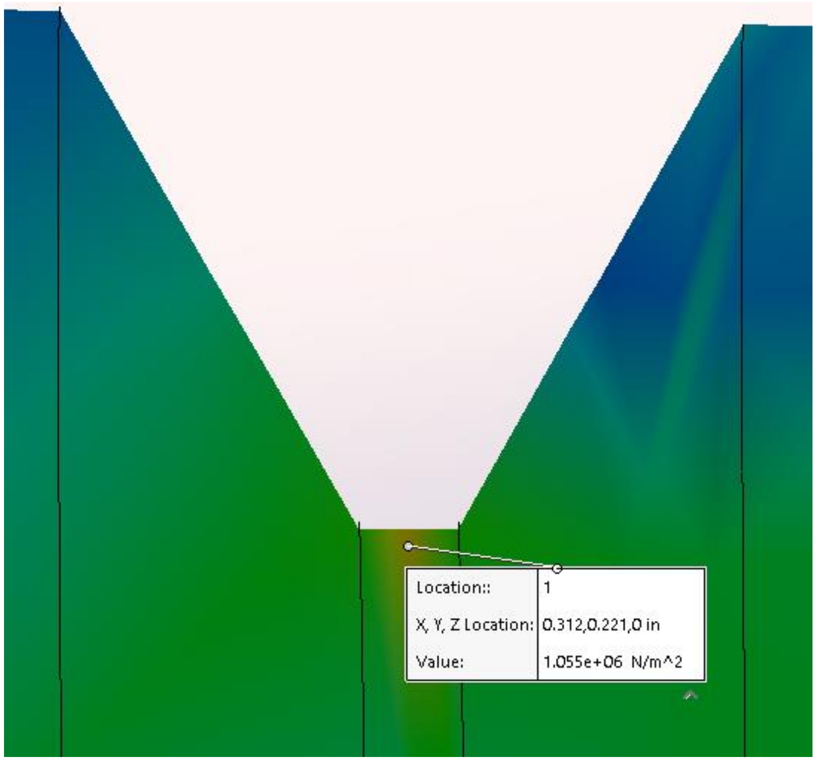

Figure 43: FEA analysis of inlet piece. Stress at the root is shown

4. Manufacturing

Since there are no moving parts in this part of the device all manufacturing tolerances are standard machining tolerances tabulated in the drawings. For the inlet piece ample material has been incorporated in the design to accommodate these machining tolerances and ensure enough material is available for welding. The ends of the air tank will be purchased, as previously mentioned. The cylinder and legs will be made from sheetmetal. The sheetmetal will be cut utilizing a shear ensuring tolerances are met. The edges will be cut to later weld the ends with a butt weld. The sheetmetal plate purchased from the manufacturer will be ordered so that all the pieces for one air tank may be cut from it. Figure 44 shows the sheet layout. The legs will be bent utilizing a brake. The cylinder will be meticulously rolled into the desired dimensions.

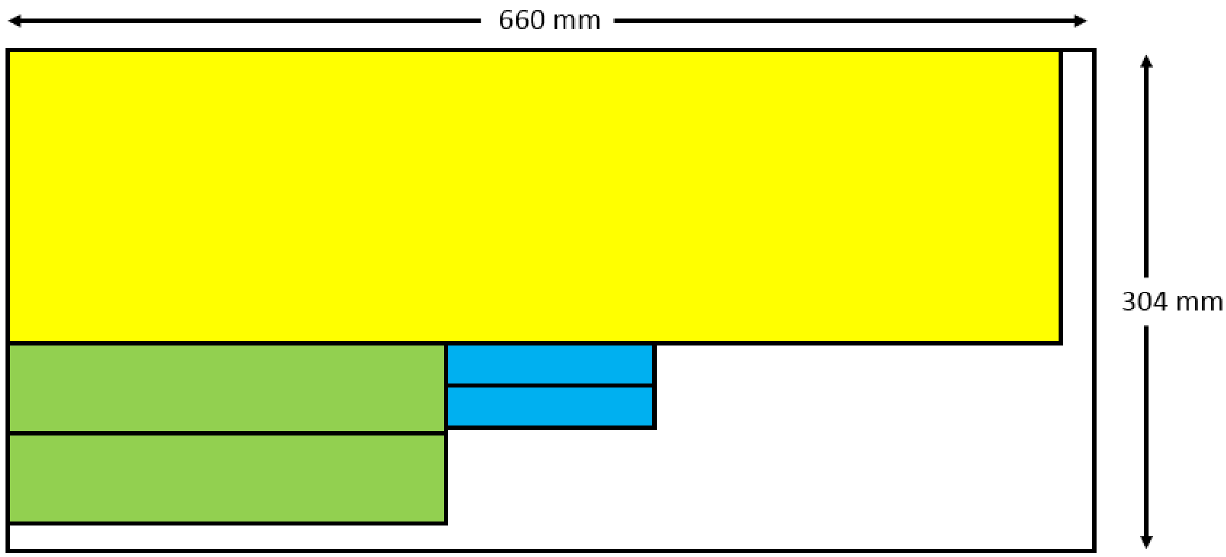

Figure 44a: Sheet metal plate layout with parts to be cut. Yellow = cylinder piece, green = tank stand, blue = pipe support. White is unused aluminum, if multiple units are made, two stands and support pieces can be cut in the white space shown.

The piece will then be welded along the seam with a V shaped butt joint weld. The inlet piece will be welded in place followed by the hemispheres, also with a V shaped butt joint. The volume of 5052 welding wire required to weld the cylinder and hemispheres is the area of the V (see Fig. 44) multiplied by the length and circumference of the cylinder. Equation (68) shows the volume calculation.

$$V_{weld} = A_{butt} L_{longitude} + 2A_{butt} L_{circumference} \tag{68}$$

$$V_{weld} = \left(\frac{1}{2}(3.66\ mm)(3.18\ mm)\right)(172\ mm) + 2\left(\frac{1}{2}(3.66\ mm)(3.18\ mm)\right)(2\pi(102\ mm))$$

$$V_{weld,cylinder} = 8460\ mm^3$$

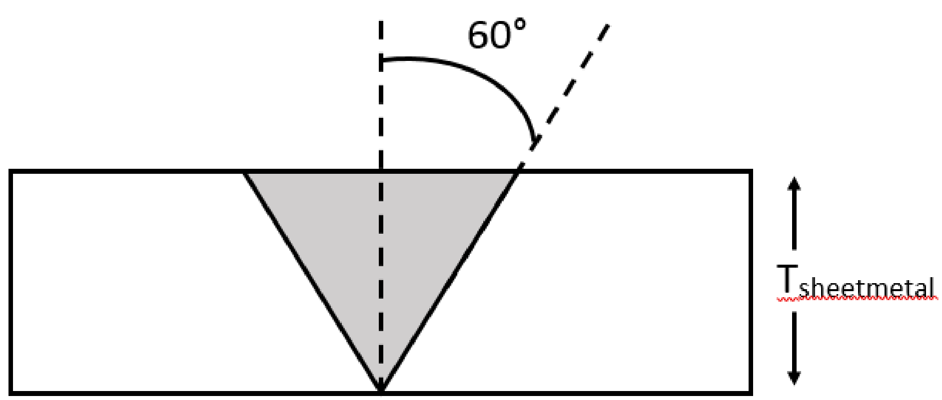

Figure 44b: Butt joint weld cross-section

The inlet piece will be entirely machined on a lathe - turned to desired specifications and NPT threads cut. Machining is the best option as it will result in a stronger piece compared to a casted one. Additionally, because of to the small amount of parts made and the relative simplicity of it, the part can be manufactured by students in the workshop. Threads will be cut instead of rolled as the cutting threads is more cost effective for low volume production.

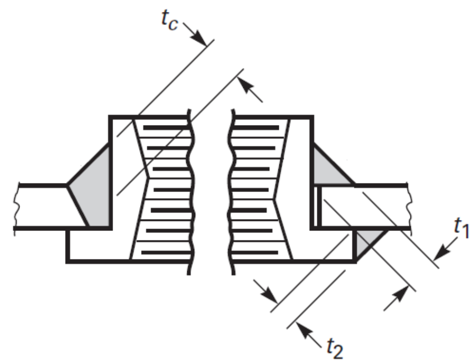

Figure 45: Weld thicknesses for inlet/outlet piece.

The bottom of the part will have a thickness double that of the shell thickness for added reinforcement and adequate surface area for welding. The 6.25 mm passage will be tapered on the tank side to ensure smooth air flow and reduce losses. A taper for a ¼" flat head machine screw was used so that the air passage may be manufactured utilizing the tooling required for a ¼" flat head through hole. The part will be welded according to UW-16 with fillet welds on both the inside and outside surfaces. The sum of thicknesses of the fillet welds must be greater than or equal to 125% the thickness of the shell (69) as seen in Fig. 45. The volume of 5052 weld wire required to secure the inlet piece to the air tank was calculated with (70). The areas of the two fillets shown in Fig. 44 were estimated to be equal. Therefore, the thickness t_1 and t_2 are equal to each other. As per the Boiler code the sum of the two is equal to 125% the tank wall thickness. The area of the triangle was then multiplied by the length of the weld which equals the circumference of the circle at each location. The radius of the lower weld as seen in Fig. 45 is 9.5 mm, the radius of the top weld is 6.9 mm (see part drawing for clarification). The cross-section of the fillet weld is illustrated in Fig. 46.

$$t_1 + t_2 = 1.25 t_{min} \tag{69}$$

$$V_{weld} = A_{fillet,} L_{circumference\ inner} + A_{fillet} L_{circumference\ outer} \tag{70}$$

$$V_{weld} = \left(\frac{1}{2}(2mm)(4mm)\right)(2\pi(12.7\ mm)) + \left(\frac{1}{2}(2mm)(4mm)\right)(2\pi(6.86\ mm))$$

$$V_{weld,inlet} = 492\ mm^3$$

$$V_{weld,total} = V_{weld,cylinder} + V_{weld,inlet} = 8950\ mm^3 \tag{71}$$

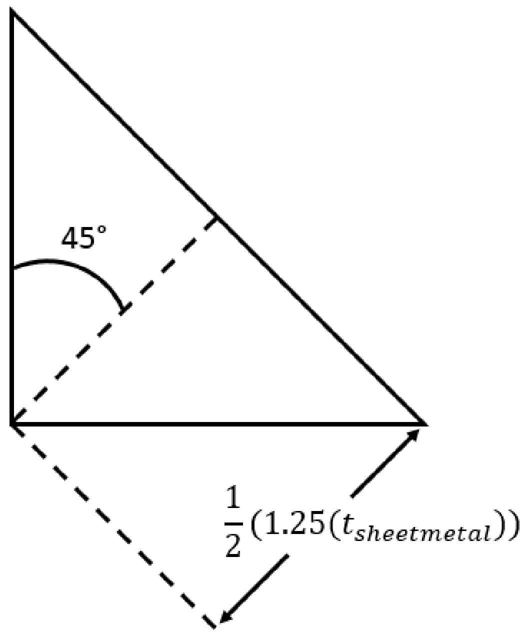

Figure 46: Fillet weld cross-section

5. Cost

Required materials for the tank parts are listed in Table XIX. Price was calculated per volume. For the turned piece and extra 50 mm of round bar stock was included in the price calculation. The purpose of the extra material is to ensure there exists enough material to attach the work piece to the lathe chuck. For the sheet metal purchased, an extra 4 mm of length is added to the needed length and width of each part to account for material lost due to cutting and manufacturer tolerances. Both the round bar stock and the sheet metal was priced from Online Metals. The hemispheres were quoted from a vendor (Wagner Metal Works). The cost for each excluding shipping is shown in the table. For the sheet metal, it must be noted that the price quoted is to manufacture one unit. However, for a plate twice as wide to manufacture two units the cost of sheet metal per unit drops to $40.56. The total material cost to manufacture one tank is $180.80. The amount of weld wire required was calculated with an assumed deposition efficiency of 50% (for a conservative approximation) to be 13.4 cm^3 or 36.3 g of aluminum wire. This cost related to this amount of wire was not included in the tables as it is negligible relative to the rest of the components. Fig. PTSD and the equations bellow it show how that result was achieved. Total cost for the tank system including prices tabulated in Table XVII is therefore $434.57.

Tank Material Cost

Description	Material	Dimensions	Cost ($)
1/8” sheet metal	5052 H32 Al	305 mm x 660 mm	53.04
¾” round bar stock	2024 O Al	94.0 mm	2.26
8” ID hemisphere	3003 Al	n/a	62.75
	TOTAL		180.80

Table (XIX). Cost for tank materials.

Diffuser

1. Operation and Functionality

The purpose of this diffuser is to ensure the volumetric flow rate required for proper delivery of the medicine is achieved. The medicine cup used, in this instance, is from the Philip's Respironics line and has specification requirements of a volumetric flowrate of 7 liters per minute (0.007 m3/min). Considering the density of air at 26.67^{o}C is 1.204 kg/m3 one can find that the required mass flow rate for proper function of this medicine cup is approximately 0.0084 kg/min. The accumulator that the cup hose is connected to on the Respironics jet nebulizer base has a specified 10 psi (68.9 kPa) as well. Considering these specification values allows one to define the velocity at which the air should be exiting the nozzle into the tube using the relations defined by equation (72) below.

$$\dot{m} = \rho A v \tag{72}$$

Where $\dot{m}$ is the mass flow rate, ρ is the density of air, v is the velocity of the outgoing air, and A is the area that it's flowing out of. The density and mass flow rate are already known, the area that the air is flowing through is the nozzle that the medicine cup hose is fit around which has an inner diameter of 5.64 mm, so the equation can be solved with only one unknown.

$$(0.0084 \frac{kg}{min})(\frac{1}{60}\frac{min}{s}) = (1.204 \frac{kg}{m^3})(\frac{.00564\ m}{2})^2(\pi)v$$

Solving for v gives a velocity of roughly 2.82 m/s which means that if the regulator, which is set to release air at 68 kPa, puts out air at this speed then it can be connected to the hose directly. To figure this out, one must calculate the velocity that the air is coming out of the regulator. The velocity coming out at 68 Kpa is given by (73) below

.

$$v = C_v \sqrt{\frac{2\gamma}{\gamma - 1}\frac{\Delta P}{\rho}} \tag{73}$$

Where C_v is the velocity constant for air which is roughly 0.661, γ is roughly 1.4 for air, ΔP is the pressure change from upstream of the regulator to downstream, and ρ is the density of air at 68.9 kPa. Plugging in these known values results in a velocity of 6.192 m/s.

$$v = 0.661 \sqrt{\frac{2(1.4)}{1.4 - 1}\frac{(689 - 68.9)kPa}{1.204 \frac{kg}{m^3}}}$$

With the velocity known but being twice as large as needed, it must be reduced to the proper speed before reaching the nozzle that the medicine cup hose is attached to. Hence, a diffuser must be designed to reduce the speed. To do this, a modified version of (72) is used considering the conservation of mass (74).

$$\rho_1 A_1 v_1 = \rho_2 A_2 v_2 \tag{74}$$

Where the subscript 1 denotes the inlet of the diffuser and the subscript 2 denotes the outlet. Equation (74) is used to define the wide end of the diffuser as the density should remain consistent as the air passes through the diffuser, the velocities entering and exiting the diffuser are known, and the inlet area is defined by the pipe feeding air into the diffuser as shown in Fig. 47 to the right. This means the inlet area A_1 has an interior radius of 4.45 mm. Inputting numbers for (74) results in an outlet radius of roughly 9.41 mm.

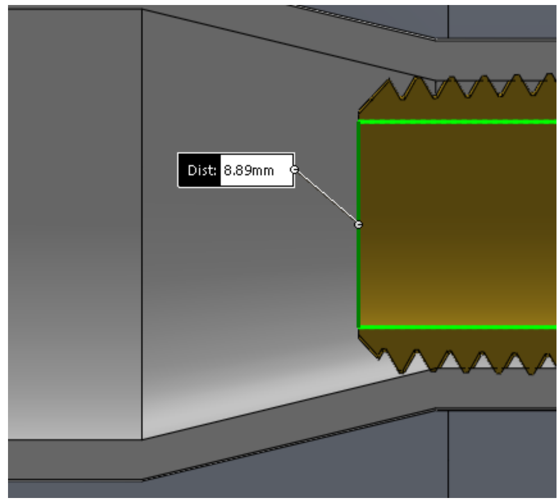

Figure 47 - Inlet of Diffuser

$$(.00445\ m^2\pi)(6.192\frac{\mathrm{m}}{s}) = (\pi r^2)(2.82\ \frac{\mathrm{m}}{s})$$

With the inlet and outlet of the diffuser set, the material must be picked to determine how thick the walls must be to achieve the desired factor of safety.

2. Material and Manufacturing

This diffuser handles pressures up to 69 kPa and considering it to be a pressure vessel would limit the material selection to only metals due to the ASME Boiler and Pressure Vessel Code section VIII UG-4. This part is incredibly simple and does not hold pressure as the outlet is always open, not to mention 69 kPa (10 psi) is a very small maximum pressure that enters the diffuser in the first place. So, it was determined that the part could be made of a polymer, similar to how the Phillip's Respironics Jet Nebulizer uses a plastic accumulator with a nozzle on the end for the hose.

The question then comes into play, which polymers are medical grade that still have the strength for this role? For a polymer to be "medical grade" it must pass two test standards, ISO 9001 and ISO 13485 which both relate to the purity and "tendency to flake off" of the polymer chosen. Obviously, a polymer that deposits plastic flakes into the air flow would be a poor choice. There are further tests required for different medical applications but the two stated above are vital to the nebulizer project specifically. A worldwide company called Lustran makes a medical grade ABS plastic that meets the requirements for this project and satisfies the necessary standards, ABS 3-CL. [58] This plastic resin can be injection molded or 3D printed and is relatively cheap at $0.30/g. [59]

The fact that this resin is easily injection molded and the small outlet nozzle on the diffuser requires a machining process with relatively high precision results in two options for manufacturing. A prototype could be 3D printed but with low tolerances and leak prone layering problems this would not be a solution that the product could be brought to market with. Injection molding is the best option to achieve the necessary tight tolerances in mass production but with molds that cost nearly $3,000 each, cost would be a dilemma in the prototyping phase. This plastic excels because it is both injection mold and 3D print compatible giving the design process options in multiple phases of production.

3. Thickness and Fatigue Analysis

From the datasheet provided by the plastic retailer, the tensile yield strength of the resin is roughly 13,100 kPa [58]. Assuming the uniform thickness would be easier and/or cheaper in the injection molding process gives multiple options for diffuser analysis. First, the required cross-sectional area of the wall in which the force is acting should be calculated with a desired factor of safety of 4. Following this, the required thickness of the threaded area of the diffuser should be calculated considering the stress concentrations in that region. This calculation should also be solving for a factor of safety of 4 to determine the thickness. Whichever calculation results in a higher thickness determines the overall thickness of the diffuser. Considering this, a fatigue analysis checking the part for 10^6 cycles must be run to ensure that the fatigue factor of safety is

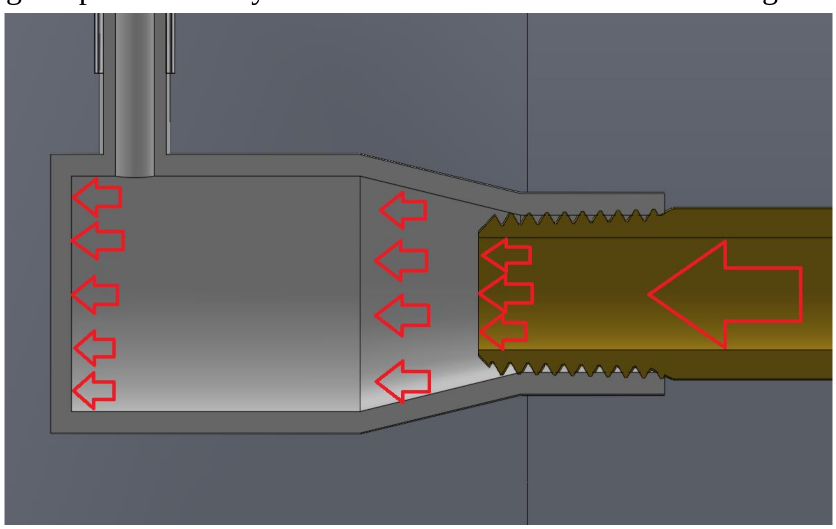

Figure 48: Flow into Diffuser

also acceptable.

First, the flow and pressure must be considered using Fig. 48 above. The majority of the force hits the back wall of the diffuser which also has the smallest surface area in the diffuser interior. At 68.9 kPa this force hits a circular back wall of diameter 20.8 mm. Using (75) below gives the force that this wall is experiencing.

$$F = \frac{\sigma}{A} \tag{75}$$

Where F is the force that the diffuser experiences. Applying the dimensions and pressure of the diffuser given above results in a force of 84.23 N acting on the back wall of the diffuser. It is known from (65) that the factor of safety is determined by the yield stress of the material divided by the max stress that the cross section of that material will experience. If one is designing for a factor of safety of 4, then the following equation results in a σ_{max} of 3,275 kPa.

$$\sigma_{max} = \frac{13{,}100 \text{ kPa}}{4}$$

Designing the cross-sectional area for this σ_{max} is a matter of applying it to (75) with the A being determined by the thickness (t) being multiplied by the back-wall diameter (20.8 mm). Solving for thickness in this case results in a back-wall thickness of at least 1.3 mm to achieve the desired factor of safety.

Next, the thickness necessary for the threading must be considered. For female threads in tension, the cross-sectional area calculation is given in (61) with the input radius {r} being defined by the OD of the pipe going into the diffuser. According to the machinery handbook, these threads have a stress concentration of 3 which is applied to (65) to find the necessary cross-sectional area for (61). This results in a required thickness of 1.65 mm for the diffuser to achieve a factor of safety of 4 on the threading. Because this thickness is larger than what is needed for the back wall cross-sectional area it will be used for the uniform thickness of the diffuser.

With this thickness in mind, a fatigue analysis for the max stress the diffuser will experience can be done. Considering that the nozzle is open to atmospheric pressure the minimum pressure that the diffuser experiences is 0 kPa. This means that the minimum stress the diffuser undergoes (σ_{min}) is 0 kPa. Applying (62) and (63) shows that the amplitude and mid-range stresses will be equivalent. Inputting the max stress calculated in (25) above results in the equation below.

$$\frac{1638 \; kPa}{6{,}550 \; kPa} + \frac{1638 \; kPa}{13{,}100 \; kPa} = \frac{1}{n}$$

Using 6,550 kPa for S_f and 13,100 kPa for S_{ut} at 10^6 results in a factor of safety of 3.56. This is more than enough to prove that the product can last a million cycles and should do so 3 times over.

4. Geometry and Closures

While the geometry of the diffuser is relatively simple, it fits together with other parts, one of which is an interference fit, making closure equations necessary. The thread for the ¼” NPT pipe that’s feeding the diffuser will be cut using a tapping tool and therefore can have variation in size. The thickness of this portion also has a factor of safety of 3 meaning that the variation can be extreme, and this would still probably not result in function failure. On the other hand, the nozzle that the hose connects to from the medicine cup must be precise. Assuming that variations in nozzle size will have a negligible effect on the flow properties then the interference fit between the nozzle and hose will be the analyzed dimension.

For this dimensional analysis (76) is used to determine most material conditions and least material conditions.

$$x = x'(1 \pm P_x) \tag{76}$$

Where P_x = the precision of the machining method used to create the part, x' is the nominal dimension of the part (measured with calipers or designed), x is the part dimension in either it's most-material condition or least material condition. Assuming the tube is Tygon PVC tubing with a tube extrusion precision of 1/10 and the diffuser injection molding has a precision of 1/10, Table XX is produced.

Closure Values for Diffuser Interface

Feature	Nominal Dimension	Machining Method	Precision
Tubing ID	5.64 mm	Tube Extrusion	$.1\frac{mm}{mm}$
Diffuser Nozzle OD	6.9 mm	Injection Molding	$.1\frac{mm}{mm}$

Table XX: Closure Values for Diffuser Interference Fit

Considering that we want an interference fit here, (77) represents the necessary interference in a worst-case scenario of the interference being minimal.

$$-Gap_{min} = \text{Tubing ID}_{MMC} - \text{Diffuser Nozzle OD}_{LMC} \tag{77}$$

The combining (76) and (77) yields a Gap minimum of -0.006 mm which means there will still be interference in the worst-case scenario.

5. Cost

The cost of the diffuser is relative to the manufacturing procedure used to make it. The raw material cost is $0.30/gram and the diffuser would be roughly 10 grams resulting in 3 dollars per diffuser. Unfortunately, molds for injection molding can be $3000-$4000. If the prototype is 3D printed, as discussed earlier, the cost for 1 prototype diffuser is roughly $3 with similar tolerances.

Tank Stand

1. Functionality Analysis

The tank is stabilized using two pieces of bent sheet metal. The sheet metal has the same material as the tank which is 5052 H32 Aluminum so that it can be welded to the tank body. Due to assembly purposes, the stand of the tank must be welded to the body as the last step in the assembly process. The tank must be able to rotate to be assembled to the rest of the system, and the stands would interfere with this process. The weight of the tank has been calculated to be 2.1

kg. The stand analysis is modeled in two different failure methods. The first method models the stand as a column to assure that the weight of the tank will not cause the stand to buckle. The second method addresses the failure that could happen due to the maximum moment at the bend of the sheet metal. The relevant material properties from the sheet metal are the Younge's Modulus, E, which is 713797 kg/cm^2, and the yield strength, σ_y, which is 1988 kg/cm^2. The first failure mode is represented in (78). A free body diagram was created to represent this failure analysis and presented in Fig. 49. The assumption was made that the entire weight of the tank would be on each leg of the stand to ensure a factor of safety was present.

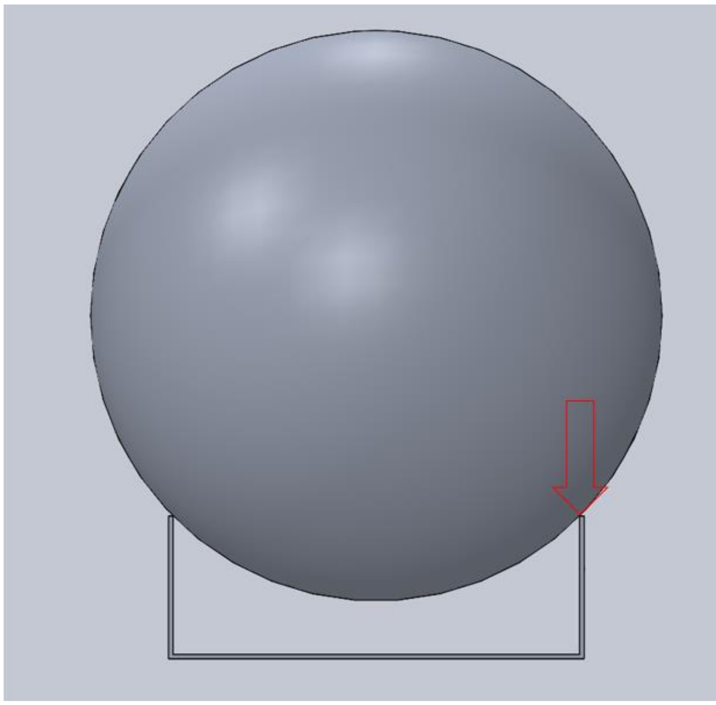

Figure 49: Free body diagram of failure method for air tank stand.

The critical load is represented as P_{cr} in (78) which is the maximum force that would cause the stand to fail from buckling. The Moment of Inertia is represented as I, and the height of the stand is represented as L. Since the weight of the tank is much below the critical load, the stand will not fail from buckling.

$$P_{cr} = \frac{\pi^2 EI}{L^2} = \frac{\pi^2(713797\,\frac{kg}{cm^2})(\frac{1}{12}(0.15875\ cm)(7.62\ cm)^3)}{7.62\ cm} = 5411490\ kg \tag{78}$$

The second failure method was through a maximum moment at the bend of sheet metal. In this model, the assumption was made that the entire weight of the tank was pressing against the side to create the moment. A representation of this is shown in Fig. 50 as a free body diagram.

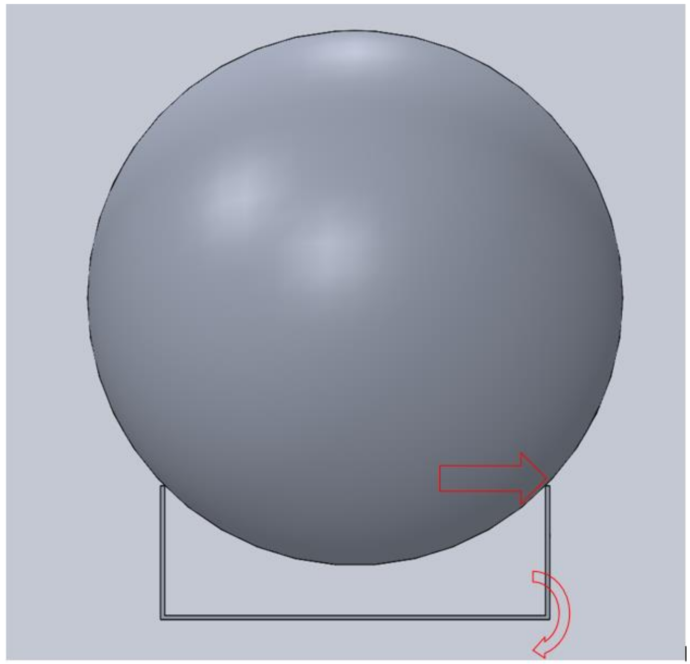

Figure 50: Free body diagram of stand failure analysis at sheet metal bend.

The equation that represents this failure analysis is shown in (79) below. This failure mode assess that the sheet metal will not break from a maximum moment. The σ_{max} in this equation represents the maximum stress that will act on the sheet metal joint. Since 0.434 kg/cm^2 is much lower than the material yield strength, the stand will not break at this point either.

$$\sigma_{max} = \frac{My_m}{I} = \frac{\big((7.62\ cm)(2.1\ kg)\big)(0.079375\ cm)}{(\frac{1}{12}(0.15875\ cm)(7.62\ cm)^3)} = 0.434\ kg/cm^2 \quad (79)$$

Pipe Stand

1. Functionality Analysis

There is one more support mechanism in this system that provides support for the piping pieces in the assembly. The weight of the pipes, regulator, diffuser, pressure gauge, brass ball valve, and check valves are supported by pieces that connect to the tank. To ensure that the weight of these pieces did not cause the system to tilt over, an analysis was performed. A free body diagram is represented in Fig. 51 and Fig. 52 to show how the system could tilt to its side if this portion was heavier than the tank.

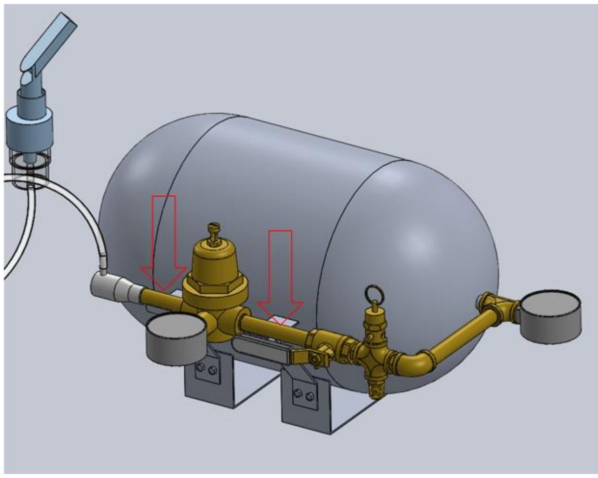

Figure 51: Isometric view of weights that can cause the stand to fail.

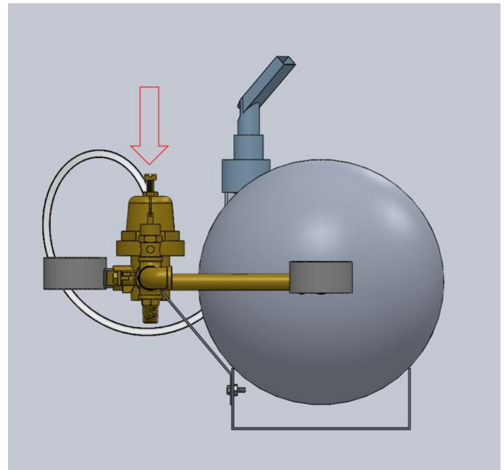

Figure 52: Side view of weights that the stand will support.

Each portion of the assembly that could cause the system to tilt was added up to ensure the systems stability. Each weight that effects this analysis is shown below in Table XXI in kilograms.

Part Name	Weight (kg)
Pressure Relief Valve	0.045
Brass Ball Valve	0.228
Hex Nipple	0.020
Check Valve	0.018
T-Valve	0.058
Regulator	0.680
Cross Valve	0.070
Right-Angle Pipe	0.041
Pressure Gauge	0.059
Pipe Supports	0.027
Hollow Rods	0.009
3” Pipe	0.045
Diffuser	0.010
4” Pipe	0.064

Table XXI: Weight of Piping System

All the weights were added up to a total of 1.4 kilograms which is less than the weight of the tank that is 2.1 kilograms. This ensures that the stand will function properly and support the pipes while the system is operated.

Pump Sub-assembly

Foot Pump Lever

A. Mechanical Advantage

It was decided that a lever action foot pump would be utilized to compress and supply air to the air tank. Knowing that the tank would be filled to 689.5 KPa, the pump had to allow the user to output the necessary force required to compress air to 689.5 KPA. The lever action foot pump was chosen with the intention of utilizing the lever to generate a mechanical advantage.

The lever would consist of a 6061-T6 Aluminum beam, a 6061-T6 Aluminum foot plate, and a dry running pillow block bearing. The pillow block bearing would be press fit onto a 1023 Carbon Steel Rod, allowing the lever to rotate and cycle the pump. The user would stand on the lever to apply a pumping force allowing them to use their body weight to charge the tank instead of physical exertion. The mechanical advantage would increase the force applied by the user. The force from the lever would then be applied to a piston, creating air flow to the tank. In generating the pump design, it was assumed an ideal gas under adiabatic compression was being dealt with. The geometry of the lever action foot pump can be seen in Fig. 53.

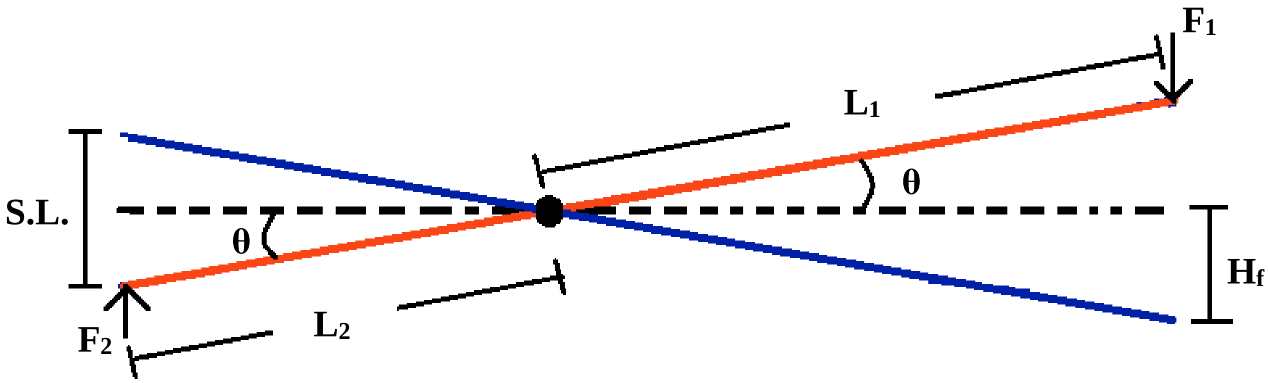

Fig. 53. Free body diagram of lever action foot pump. Orange represents piston at start of compression stroke. Blue represents piston at end of compression stroke.

Geometrical relationships were derived to allow different pump sizing parameters to be determined. First, a relationship between the lengths of the two lever sections was determined based on the force applied at each end of the lever (80). L_1 represents the length of the lever section between the foot pedal and fulcrum. L_2 represents the length of the lever section between the piston and fulcrum. The minimum force applied by a person standing on the lever, F_1, was set at 667.23 N. F_2 represents the force required for the lever to overcome. Knowing that the pump must compress a maximum of 689.5 KPA, F_2 was found in (81) where A_p represents the area of the piston face. F_1 and F_2 are assumed to always act in a parallel orientation. The ratio of lever lengths due to applied force was then derived in (82).

$$F_1 L_1 = F_2 L_2 \tag{80}$$

$$F_2 = 689.5 \cdot A_p \tag{81}$$

$$\frac{L_1}{L_2} = 1033.34 \cdot A_p \tag{82}$$

Next, a relationship between the lengths of the two lever sections was determined based on the maximum vertical travel of the lever. Since the lever was set to increase the force applied by the user, it was known the L_1 section would see the maximum vertical travel. The fulcrum height, H_f, was set to 0.2032, the diameter of the air tank. This was done to keep sizing of the pump similar to the sizing of the tank. The maximum vertical travel was known to be twice the fulcrum height. The relationship between lever lengths and maximum vertical travel of the lever can be seen in (83), where S.L. represents the stroke length of the piston. Equation 83 was then further derived into (84).

$$sin\theta = \frac{S.L.}{2 \cdot L_2} = \frac{0.2032}{L_1} \tag{83}$$

$$\frac{L_1}{L_2} = \frac{0.4064}{S.L.} \tag{84}$$

B. Volumetric Displacement and Fill Time

The volumetric displacement, $\forall_p$, of the piston pump was determined by (85). Equations 82 and 84 were equated to determine the maximum volumetric displacement of the lever action foot pump (86).

$$A_p \cdot S.L. = \forall_p \tag{85}$$

$$1033.34 \cdot A_p = \frac{0.4064}{S.L.} \tag{86}$$

Ultimately it was determined the pump could have a maximum $\forall_p$ of $3.9329x10^{-4}\ m^3$ based on the applied forces and allowed vertical travel of the lever. As previously mentioned, a mass of 0.084 kg must be stored in the air tank to complete one treatment from the nebulizer. The mass displaced by a single pump, m_{pump}, stroke was found in (87), where the density of air, ρ_{air}, at standard temperature and pressure is 1.204 kg/m^3. The number of pumps required to fill the tank was then calculated. Assuming 1.5s per pump, the time required to fill the air tank for one treatment, T, was found to be 4.35 minutes with a total of 178 pumps (88).

$$m_{pump} = \rho_{air} \cdot \forall_p \tag{87}$$

$$T = \frac{m_{pump}}{0.084kg} \cdot \frac{1.5s}{60s/min} \tag{88}$$

C. Load Analysis

The first step taken to design the lever was to determine the required lengths, L_1 and L_2, for the individual lever sections. L_2 was determined to be 101.6 mm based on the required clearance for the piston connecting rod, this will be further elaborated in the connecting rod section of the report. Based on the relations found in (82) and (84), L_1 was found to be 406.4 mm. It was concluded a solid 25.4x50.8x541.03 mm rectangular, 6061-T6 Aluminum stock would be used to make the pump lever [60]. A 6.35x101.6x152.4mm 6061-T6 Aluminum plate would be fastened to the top of bar stock to provide a stable platform to stand on [61]. The shear and bending moment diagrams for the pump lever can be seen in Fig. 54.

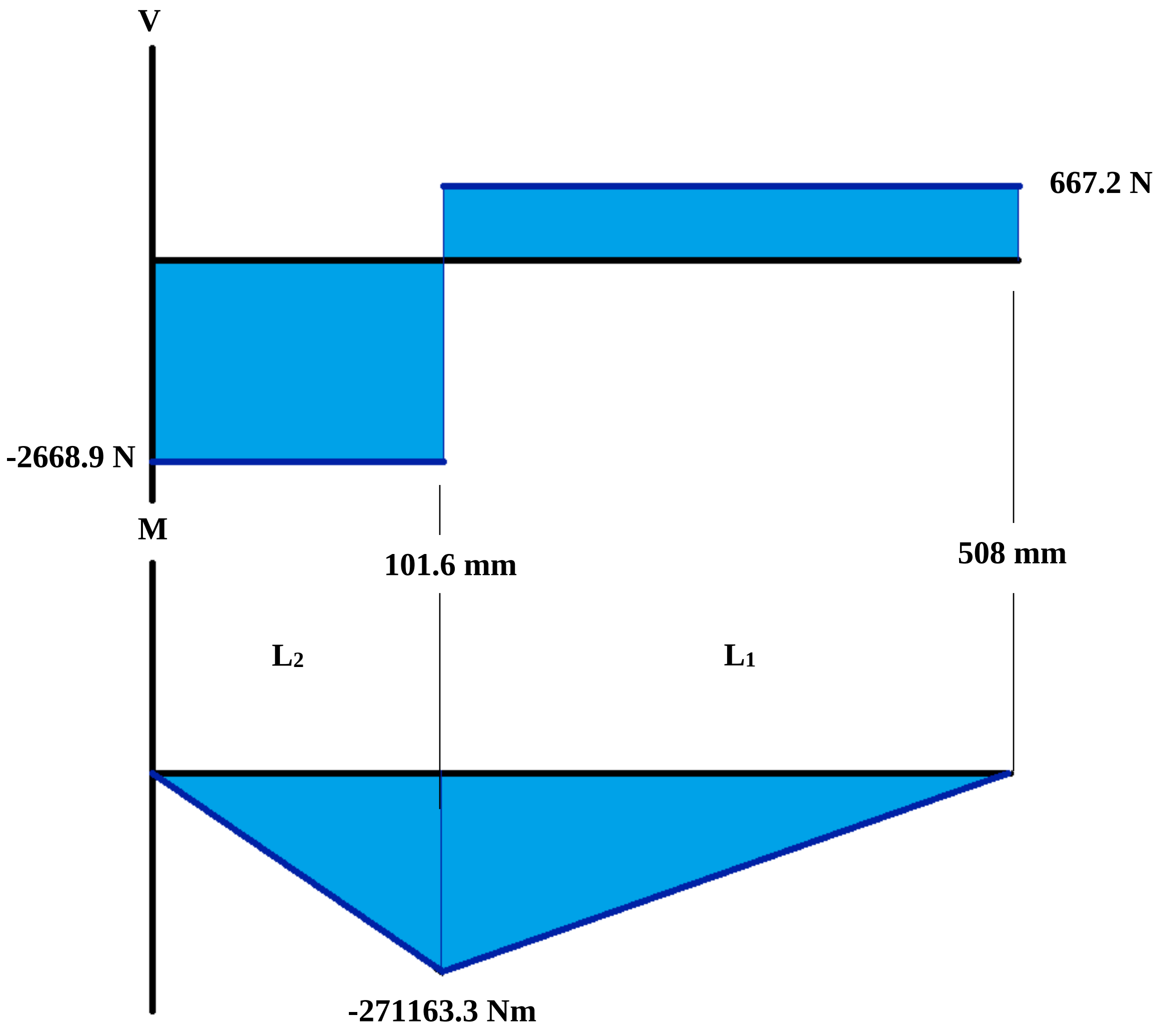

Fig. 54. Shear and Bending Moment Diagram, Static Loading, 100 psi in piston pump

Static loads were then used to find the maximum static stress acting on the lever using (89). The maximum static load was found to be 49.6 MPa. Considering the yield stress of 6061-T6 aluminum is 276 MPa, this gives a static factory of safety of 5.6 [62].

$$\sigma_m = \frac{M}{S} \tag{89}$$

Knowing that the lever cycles from zero stress to 49.6. MPa, the modified Goodman equation was then used to find the fatigue strength of the lever (25). The fatigue stress of the lever was found to be 60.5 MPa. The fatigue strength of 6061-T6 Aluminum at 5.00e8 cycles is 96.5 MPa, giving the lever a factor of safety for dynamic loading of 1.6 [62]. Due to the high number of cycles the lever can withstand, it was concluded to have infinite life.

Deflection of the lever was then found to ensure $\forall_p$ would never be inhibited due to a sizable amount of beam bending. Deflection of the lever was calculated using (90). The maximum deflection was found to be 3.10 mm. Due to the small value of maximum deflection, it was concluded $\forall_p$ would never be inhibited by beam deflection.

$$y = \frac{PL^3}{3EI} \tag{90}$$

D. Lever Manufacturing, Assembly, and Cost

Manufacturing the lever starts with cutting down the 6061-T6 Al rectangular bar stock. The stock is purchased in a 609.6 mm length and will be cut down to 541.03 mm using a marvel band saw. Standard manufacturing tolerancing is used for this operation since minute changes in the length of the lever is not critical to successful part function. After that, the 6061-T6 Aluminum foot plate will have two M7 clearance holes drilled through it. The foot plate is purchased from the manufacturer with the proper dimensions. Next the Al rectangular bar stock will have 6 clearance holes drilled through it. Two M7 clearance fits are to secure the connecting rod pillow block bearing. Two M7 clearance fits are to secure the foot panel to the lever. Lastly, Two M8 clearance fits are to secure the pillow block required for lever pivoting. Both pillow blocks were chosen based on their low friction properties and the high load allowance specified by the manufacturer [63]. Next, the 1023 Carbon Steel rod will be cut down from 304.8 to 152.4 mm. Standard machining tolerances apply here since the width of the pump frame can accommodate slight variances in the rod length. The 1023 Carbon Steel rod is then press fit into the pillow block for lever pivot. The rod pillow block assembly is then fastened to the lever using M8 bolts. Next the connecting rod pillow block and foot plate are fastened to the lever using M7 bolts. The lever assembly is now complete and ready to be implemented into the frame.

E. Closure

A slight press fit was determined to be the best method of inserting the 1023 Carbon Steel rod into the pillow block required for lever pivoting [64]. The Carbon Steel rod was purchased at the desired radius with a tolerance specified by the manufacturer [65]. The pillow block required for lever pivoting was purchased at the desired inner radius with a tolerance specified by the manufacturer [63]. A closure equation was developed in Fig. 55 to ensure the rod supplied would always form a slight interference fit with the pillow block. Looking at the interference results, one can observe there will always be some degree of interference between the shaft and the pillow block. The closure equation was solved using English units since that was the form provided by the manufacturer. Closure equations relating the angle of rod insertion to piston orientation were not developed due to the pillow bearings ability to account for misalignment [63].

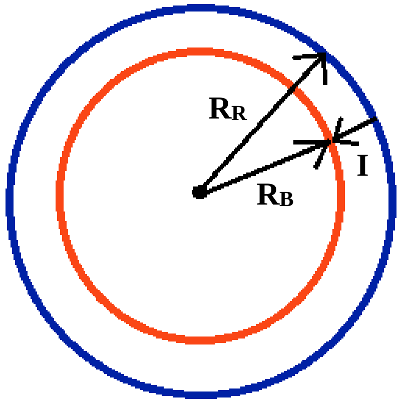

Fig. 55. Closure diagram for steel rod in pillow block

Closure Values for steel rod in pillow block

Feature	Nominal Dimension [in]	Tolerance	Interference, I
Rod Radius, R_R	0.25	+0.0000 -0.0010	Max: 0.0005
Pillow Block Inner Radius, R_B	0.2495	+ 0.000 - 0.0005	Min: 0.0000

Table XXII: Closure values for steel rod in pillow block. Dimensions in English units due to manufacturer specifications.

Dynamic Loading Approximation

To simplify pump calculations the complex dynamic loading profile of the pump was converted to an equivalent loading profile that corresponds to the maximum loading conditions. As pressure is increased in the air receiver the amount of pressure the pump experiences during the stroke increase. At the beginning of the charge cycle the pressure change is close to zero but increases to 689 KPa for the final few strokes.

To understand how the pressure in the tank changes with each pump a modified form of the ideal gas law was used. Equation (91) assumes an isentropic compression from the pump to the air receiver. The constants in this equation are a temperature of 293.15 Kelvin, a ideal gas constant of 8.314 L·KPa/m·mol, and the molar mass of air is 28.97 g/mol. The volume of the tank is also fixed at 9.95 liters.

$$P \cdot \forall = m \cdot \frac{R}{M} \cdot T \tag{91}$$

The mass added to the tank depends on the number of cycles. One complete stroke of the pump is considered a real pump cycle. The pump is designed to move 0.4732 grams per stroke of the pump. By multiplying the grams per stroke by the number of real pumps cycles the amount of mass in the air tank can be calculated (Equation (92)).

$$m = RPC \cdot 0.4732 \frac{grams}{Real\ Pump\ Cycle} \tag{92}$$

By inserting equation (91) into (92) figure 56 was created. To fully charge the air receiver 173 real pump cycles are needed

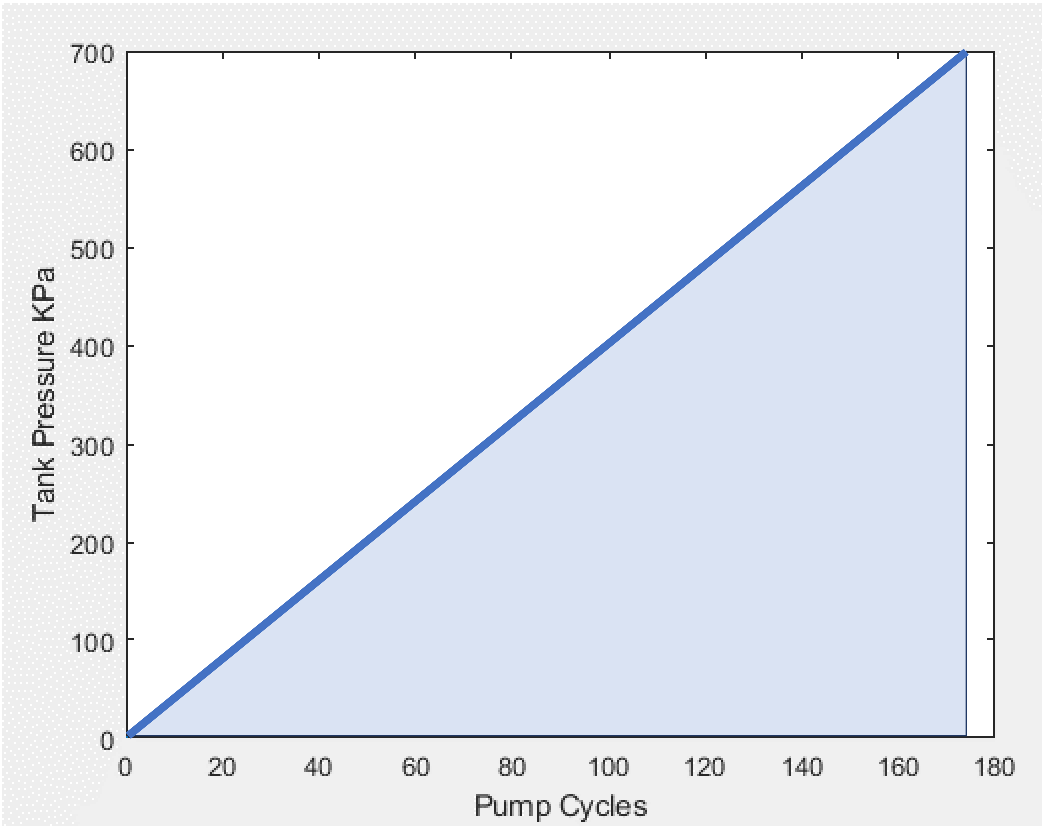

Figure 56: Real loading profile acting on the pump. The pressure in the tank is considered equal to the maximum pressure the pump will experience. The area underneath the curve shaded in light blue is analogous to the total cyclical loading damage sustained by the pump during the charge cycle.

Pressure profile within the tank is analogous to the pressure experienced by the pump. The area underneath the real pump cycle curve is essentially the total damage done to the pump in order to charge the air tank for one use which is known as a charge cycle. To match this damage profile a rectangle always at the maximum loading pressure was used this was considered an equivalent damage profile. The number of cycles required by the equivalent damage profile to have the same area as the real damage profile is are considered the equivalent pump cycles. Figure 57 shows the equivalent damage profile and the real damage profile.

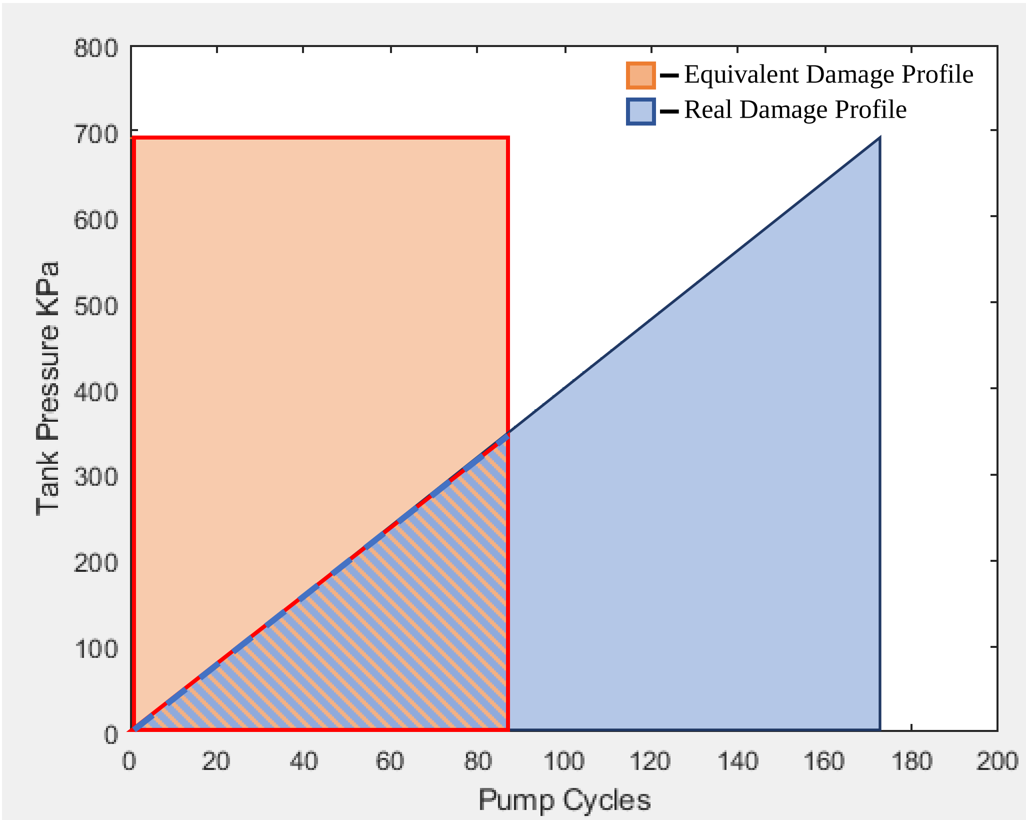

Figure 57: Equivalent loading profile at maximum pressure. The area in red is the equivalent loading profile where the cumulative damage underneath the equivalent load is equal to the static load.

The figure above shows has shown that 86.5 equivalent pump cycles are needed to match the required 137 real pump cycles; This results in a real to equivalent pump ratio (PR) of 2. This allows dynamic loading to be calculated for simple alternate loading at the maximum pressure in the piston. The modified Goodman equation (93) was used to calculate the failure stress for materials based on their yield, ultimate, and fatigue strengths. The number of cycles the fatigue stress is evaluated at is equal to the equivalent pump cycles.

$$S_{Failure} = \left(\frac{2}{n}\right)\left(\frac{1}{\frac{1}{S_f} + \frac{1}{S_{Ult}}}\right) \tag{93}$$

The number of actual cycles experienced by the pump can be determined using equation 59. This equation is based on the ratio between real pump cycles and equivalent pump cycles determined above

$$RPC = RP \cdot EPC \tag{94}$$

At 137 real pump cycles or 86.5 equivalent pump cycles the air receiver is fully charged for one treatment, at this point the pump has undergone one charge cycle at which point the nebulizer is ready to be used.

The approximation made above allows the failure strength to be plugged directly into statically determined equations for maximum loading conditions to analyze the system for dynamic loading at changing values. This allows critical dimensions to be determined.

A. Dynamic Application

The dynamic loading alternates between zero and the maximum loading stress. The average and mean stress are equal and are at half of the maximum stress experienced by the system. The modified Goodman equation can then be used to solve for the average stress on the system which is equal to the stress at which the part will fail for a certain number of cycles. The number of cycles the part will last is the number of cycles the endurance limit was calculated for. This value can then be plugged directly into maximum loading condition equations to get dimensions based on cyclical loading.

For the pump analysis the number of cycles that the pump will last be based on the modified Goodman equation is the equivalent pump cycles the pump is under. To calculate the number of real pump cycles Equation (94) was used with the calculated pump ratio of 2. For aluminum parts the fatigue strength was evaluated at 10^6 cycles which gives $2 \cdot 10^6$ real pump cycles. For steel components the stresses will be under the endurance limit of steel which will allow virtually infinite life.

B. Pump Lifetime Analysis

The lifetime of the pump will be based on the number of charges cycles the pump can produce to calculate the total number of uses per pump. Based on the aluminum components the pump will last for $2 \cdot 10^6$ real pump cycles. Each charge cycle requires 137 real pump cycles. By dividing the maximum number of real pump cycles by the number of real pump cycles per charge the total number of charges was calculated (95).

$$Charges = \frac{Total\ Cycles\ Untill\ Failure}{Cycles\ Per\ Charge\ Cycle} \tag{95}$$

The pump is capable of producing 14599 charge cycles before any component other than the O-Ring needs to be replaced. To estimate the timeframe on which the pump will last the pump was categorized into three categories of use: high, moderate, and low use. High use is three cycles an hour for twelve hours a day, moderate use is two cycles an hour for twelve hours a day, and low use is one cycle an hour for eight hours a day. Assuming the recovery time for the disaster is three months the total number of days is expected to be 90 days. Table XXIV shows the expected number of deployments before failure for usage.

Estimated Operating Time for the Foot Pump

Operation Intensity	Cycles per hour	Hours per day	Total Deployments
High	3	12	4.5
Moderate	2	10	8.1
Low	1	8	20.3

Table XXIV: The estimated operating time for the foot pump based on intensity of operation. Each deployment corresponds to one application of the foot pump to a disaster relief zone for 90 days. If the pump is deployed to one disaster relief zone per year it is equivalent to years.

Isothermal Compression Assumption

The pumping process that moves air from the pump to the tank assumes isentropic compression of the ideal gas as well as an isothermal process. To check if the process is isothermal a modified form of the ideal gas law was used; Equation 96. For the following equation the subscript one refers to the pump and the subscript two refers to the air receiver.

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \tag{96}$$

The temperature and pressure of the air at the pump is assumed at standard temperature and pressure, which has a pressure of 101.35 KPa and temperature of 293.15 K. The volume at point one is the volume of the pump which is the ideal amount of air compressed, $3.93 \cdot 10^{-4}$ m^3. The known condition of the tank is a pressure of 689.48 KPa. To create a temperature difference of one-degree celcius the tank volume must be smaller than $5.768 \cdot 10^{-5}$ m^3. This corresponds to a volume of 0.057 Liters which is much smaller than the tank therefore the temperature of air can be assumed as constant during compression.

Piston Head

The piston head is what slides within the piston cylinder to pump air into the receiver. The primary concern regarding the piston head is the tolerancing to ensure no contact is made between the piston head and the cylinder wall. The piston head will be made out of 6061 Aluminum bar stock; it will be cut to length using a bandsaw and all major features will be machined by a lathe.

A. Dimensional Considerations

The area of the piston head was calculated based on the geometry of the pedal and required stroke length. The area of the piston head was chosen to be 38.7 cm^2. The piston head needs to fit within a cylinder, the diameter of the piston head was calculated to be 7.01 cm with a tolerance given by Machineries Handbook for an RC 5 clearance fit to be -0.0635 mm to -0.0940 mm [66].

B. O-Ring

To properly seal the pump, an O-ring was sourced to prevent air from escaping between the piston head and piston cylinder. Based on the calculated piston diameter, it was determined an AS-568-146 O-ring would be required for our application [33]. The O-ring chosen for the piston head was a Medical Grade Silicone AS-568-146 O-ring offered by Apple Rubber. This specific O-ring was chosen due to its ability to meet ISO 10993 requirements [67].

C. Piston Head Geometry

The groove in the piston head is designed to fit the O-Ring selected to create the seal between the piston head and piston cylinder. Fig. 58 below shows the dimensions of the trough and was used to derive a set of closure equations to fully define the piston head. The trough is designed to be a square trough where h_2 is negligible. The depth of the groove was chosen to ensure approximately 75% of the O-Ring cross sectional area is contained within the groove. The dimensions of the groove were given for a 146 O-Ring from the Parker O-Ring for hydraulic pistons catalog. The inner diameter of the piston head must be between 6.678 cm to 6.683 cm. The total height of the groove must be between 0.3581 cm to 0.3835 cm [33].

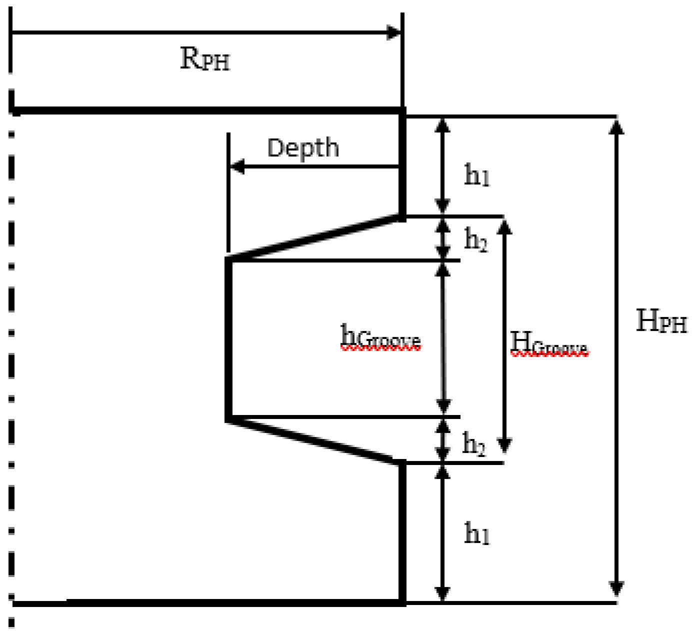

Figure 58: Dimensional Diagram for the side profile of the piston head. The dashed line on the left represents the centerline of the piston head. This profile is revolved around the centerline to create the piston head.

Equation (97) shows calculates the nominal height of the piston head based on the maximum groove height and the required height on each side of the groove in order to prevent O-Ring blowout. The equation solves for the minimum required height of the piston in order to safely hold the O-Ring. A large tolerance can be applied to the height of the piston head as long as it is larger than this critical dimension. The height on each side of the O-Ring h_1 was arbitrarily chosen to be half of a centimeter with a large tolerance of a tenth of a millimeter.

$$H_{PH,MIN} = 2 \cdot h_1(1+\mathcal{P}) + H_G \cdot (1+\mathcal{P}) \tag{97}$$

Using the equation above the minimum required height of the piston head is 1.58 cm. This will be allowed a cutting tolerance of 2.5 mm.

A pillow block is attached to the center of the piston head for the connecting rod to attach to. This provides the force which acts on the bottom of the piston head. The compressive forces were estimated as a distributed load due to the maximum pressure on the interior side of the head. Equation (98) was derived to ensure the piston head remained rigid during operation.

$$\sigma' = n \cdot \frac{{D_{PH}}^2}{{d_{PH}}^2} \cdot P \cdot K_t \tag{98}$$

K_t was approximated using Shigleys Mechanical Engineering Design tables for a grooved bar in tension [68]. This equation was used to ensure the piston cylinder will not fail and deform during operation. The stress was checked for the middle tier of the piston head where the stress concentration is the highest.

The stress concentration on the piston head from the groove was found to be approximately 2.5. Using the following numbers in equation (98) the stress of the loaded aluminum is 1.2 KPa. This is substantially lower than the failure stress approximated by the ASME-eliptic equation for dynamic loading of 2,277 KPa. Since the dimensions of the piston head were determined by geometry this ensures the piston head will not fail.

Two threaded holes will be tapped into the base of the part to allow a pillow block to be attached. The through hole in the pillow block is 7.1 mm in diameter which allows for a M7 screw to be used. Steel screws will be used to attach the pillow block warranting UNC threading as the aluminum piston head is the weaker component. The tap diameter of a M7 UNC threading is 6 mm. The taps will go deep enough to ensure five thread lengths are engaged the tap will be 8 mm with a conical bottom so finishing does not need to be done, this depth will allow eight threads to engage. The pattern is linear with the screws 44.45 mm meters apart diametral. This allows for 2 M5x12 mm screw to be used. The threads will have a negligible stress concentration compared to the groove and are not assumed to affect the stress within the head by enough to create failure.

D. Manufacturing and Costs

Standard 6061 Aluminum bar stock was considered for an aluminum piston head which is approved for medical use with non-oxidizing gases. The aluminum variant of the piston head will be shaped from a 3-inch diameter bar stock which corresponds to a 76.2 mm diameter. Based on the tolerances provided by McMaster-Carr the minimum diameter this bar is manufactured to is 76.11 mm which is above the maximum machining diameter of 70 mm so the bar stock can be turned to length. The bar stock costs $17.29 per 3-foot unit which corresponds to $0.20 per millimeter in height of the piston head [69]. Based on the total height of the piston head the head will cost $3.16. The volume of the piston head can be calculated using equation (99).

$$\forall = \frac{\pi}{2} D_{PH}^2 \cdot h_1 + \frac{\pi}{4} d_{PH}^2 \cdot H_{groove} \tag{99}$$

This volume was calculated to be 51.13 cubic centimeters which was multiplied by the density of 6061 Aluminum which is 2.7 grams per cubic centimeter to yield a mass of 138.05 grams.

To manufacture the piston, head the cylinder must first be cut to length this operation will take approximately 5 minutes by hand. The part then needs to be turned to the proper outer diameter, this operation will take approximately 10 minutes on an engine lathe. The central groove then needs to be turned into the part, this operation is expected to take 15 minutes due to the amount of material that must be removed. The piston head then needs to be transferred to an endmill where two threaded holes will be taped into the cylinder. A cylindrical shape is difficult to work with on and endmill so this operation is expected to take 30 minutes. The total manufacturing time estimation for a human machinist is conservatively estimated as one hour per piston head. The summary of the piston head manufacturing and material costs can be seen in table XXV.

Piston Manufacturing Material and Labor Costs				
Part Name	Part NO.	Labor Cost	Material Cost	Total Cost
O-Ring		N/A	$0.73	$ 0.73
Piston Head		$90.00	$3.16	$ 93.16
M7 x 12 Hex Bolt		N/A	$0.76	$ 0.76

Table XXV: Summary of material and labor costs for the piston head assembly. The labor cost is assumed for a worker who costs $90 per hour.

Pump Cylinder

The piston cylinder is used to contain the air which will be forced into the tank by the piston head. The cylinder consists of a cylindrical piston wall which is modeled as a thin pressure vessel with a flat disk on top with two nozzles, one is an inlet and the other is an outlet. This will be hard mounted to the pump frame and the piston head will move within. This was made out of AISI 4130 steel to prevent material strength loss due to welding [70].

A. Dimensional Calculation

The piston cylinder is modeled as a cylinder with a flat disk top and an open bottom which piston head moves within. The internal diameter of the piston cylinder is determined by the diameter of the piston head and the tolerance for an RC5 fit [71].

Fig 59 below shows how the outer diameter of the piston head and inner diameter of the cylinder influence the gap between the cylinder.

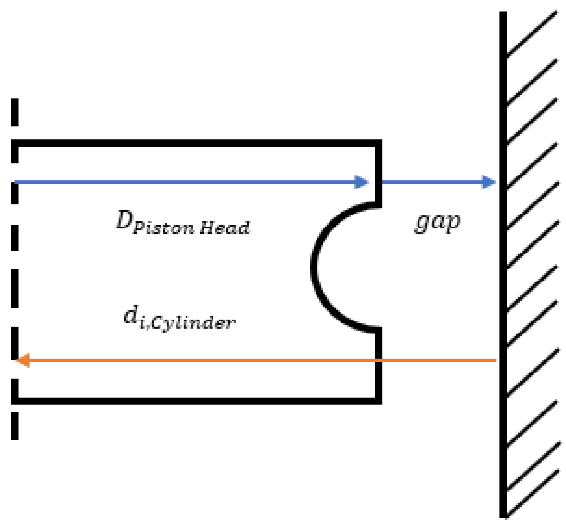

Figure 59. Closure diagram for the piston cylinder running fit.

Equations (100) and (101) define the maximum and minimum gap distance. The minimum gap distance is determined for a running fit. For a cylinder head nominal diameter of 7.0104 cm, an RC-5 clearance has a maximum of 0.0635 mm and a minimum distance of 0.0940 mm [71].

$$D(1+\mathcal{P}) + 2 \cdot g_{min} = d_i(1-\mathcal{P}) \tag{100}$$
$$D(1-\mathcal{P}) + 2 \cdot g_{max} = d_i(1+\mathcal{P}) \tag{101}$$

Based on the closure equations done above the minimum inner diameter of the piston cylinder has a nominal size of 7.01 cm with a tolerance of 0 mm to + 0.046 mm. The Parker O-

Ring catalog specifies the internal diameter average roughness must be between 0.5 and 0.25 mm [33].

The thickness of the piston cylinder is decided by the stresses that the cylinder is expected to see. The pump was approximated as a thin walled pressure vessel with the maximum hoop stress and longitudinal stress determined by equations (102) and (103)

$$\sigma_H = \frac{P \cdot d_i}{2 \cdot t} \tag{102}$$

$$\sigma_l = \frac{P \cdot d_i}{4 \cdot t} \tag{103}$$

A force is applied to the top of the cylinder that is also determined by the pressure. The force was calculated by multiplying the pressure within the cylinder by the interior area of the cylinder top; which is just the area of a circle with a diameter equal to the inner diameter of the cylinder (104).

$$F_{Load} = P \cdot \frac{\pi}{4} \cdot d_i^2 \tag{104}$$

This force is applied over the area of the cylinder to calculate the stress on the piston cylinder due to pressure acting on the top part of the cylinder (105). This load is assumed to be an axial load along the length of the cylinder.

$$\sigma_{Load} = \frac{P \cdot d_i^2}{4 \cdot d_i^2 \cdot t + 4 \cdot t^2} \tag{105}$$

The distortion energy failure criterion was used to determine the static failure conditions for the pump. Where σ_1 is the hoop stress and σ_2 is the longitudinal stress and the loading stress added together. These were used with the distortion energy criterion to determine the Von-Meises stress for the cylinder at any given thickness (106).

$$\sigma' = n \cdot \left[\frac{\left(\frac{P \cdot d_i}{2t} \left(\frac{1}{2} + \frac{d_i}{2d_i^2 + 2t} \right) \right)^2 + \left(\frac{P \cdot d_i}{4t} \left(1 + \frac{d_i}{d_i^2 + t} \right) \right)^2 + \left(\frac{P \cdot d_i}{2t} \right)^2}{2} \right]^{\frac{1}{2}} \tag{106}$$

The minimum required thickness can be calculated by solving equation (106) for the thickness where the Von-Meises stress is equal to the yield stress. Dynamic loading conditions were analyzed using the ASME-Elliptic curve for aluminum failure at $5 \cdot 10^8$ cycles. Dynamic loading was the only loading analyzed as it would require a much greater thickness than static loading. The required thickness of the cylinder is 0.8 cm with a factor of safety of four. For a 4130 Steel alloy the required thickness of the wall is 0.48 cm, this is for the infinite life condition of steel.

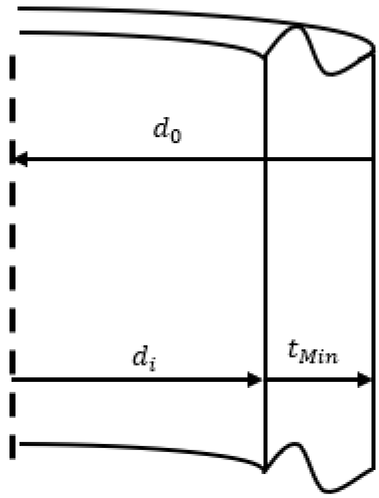

Figure 60. Closure diagram for the cylinder thickness

Equation (107) was derived from fig. 60 to calculate the minimum required outer diameter of the cylinder to ensure the thickness criteria is met.

$$d_i + \Delta d_i + t_{Min} = d_{0,Min} \tag{107}$$

For the steel cylinder the outer diameter must be a minimum of 7.4946 cm to ensure the part will not fail. A steel hollow bar stock was selected to ensure this requirement the Otter diameter of the bar stock is 7.62 cm with a tolerance of 0.023 cm. The inner diameter of this tube stock is 6.67 cm which is much smaller than the inner diameter of the piston cylinder allowing it to be machined to the proper size [72].

B. Cylinder Top

The cylinder top was modelled as a flat disk under a distributed load. The static load was evaluated at the maximum pump pressure. Equation (108) was used to calculate the minimum required thickness in order to prevent bursting in the lid. The radius of the top was assumed to be the inner diameter of the cylinder. This equation assumes small deflections in the top of the cylinder.

$$t_{Min} = n \cdot \sqrt{\frac{3 \cdot P \cdot r^2 \cdot (3 + v)}{8 \cdot \sigma_{yeild}}} \tag{108}$$

The yield stress is the von-misses stress acting on the surface of the plate. Under the dynamic loading assumed for the cylinder the required thickness was calculated using ASME-Elliptic. The for the aluminum alloy the required thickness of the cylinder top is 0.62 cm. For the steel alloy the required thickness is 0.42 cm.

The inlet and outlet to the cylinder will be achieved by having ¼ inch NPT threads screwed into the top of the cylinder. To be fully engaged a ¼ inch NPT thread needs 15.875 mm of material to thread into [73]. A bar stock of diameter equal to the outer diameter of the piston will be cut to a height of 16 mm with a tolerance of 0.1 mm.

C. Cylinder Height

The stroke length of the piston head is 10.16 centimeters. The cylinder needs to be tall enough to allow the piston head to complete one stroke. An additional length was added to the height of the cylinder to account for the height of the piston head. This ensures the seal is not broken at the bottom of the stroke. At the top of the stroke a gap between the top of the piston cylinder and the piston head is calculated based on an RC 9 fit which has a required clearance between 0.02286 cm and 0.05207 cm. the figure below shows the piston assembly at the bottom of the stroke.

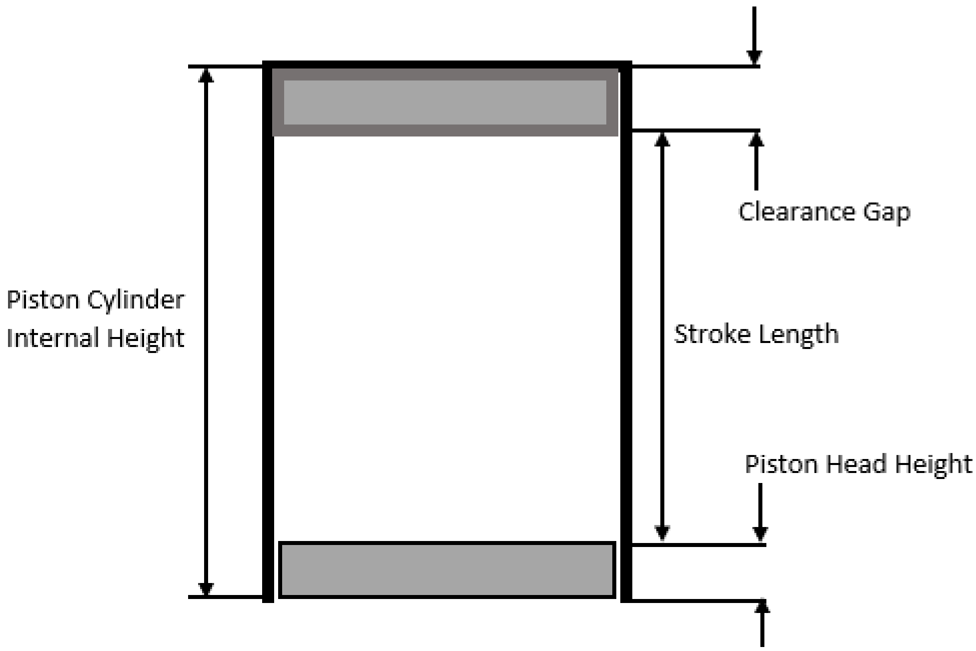

Figure 61: Dimensional Diagram for the internal height of the piston cylinder

Fig. 61 was used to derive a set of closure equations for the internal height of the cylinder equation (109) solves for the maximum internal height and equation (110) solves for the minimum internal height of the piston cylinder. The uncertainty in stroke length was assumed to be the same uncertainty of the connecting rod Assumed 1 mm.

$$h_{max} = gap_{Max} + SL + \Delta SL + PH + \Delta PH \tag{109}$$

$$h_{Min} = gap_{Min} + SL - \Delta SL + PH - \Delta PH \tag{110}$$

Using the above equations, the internal cylinder height must be between 11.61 cm and 12.14 cm. The maximum outer height of the cylinder has no functional restraints and does not need to be calculated.

D. Mounting Plate

The mounting plate was made from the same material used for the top of the piston cylinder. The mounting plate is a square with a circle cut out of the center for the piston cylinder to fit through. Four through holes are cut at the corners of the mounting plate these will have standard drilling tolerances. The mounting plate will be welded to the open base of the piston cylinder. The hole cut in the middle of the piston cylinder will be water jetted to a diameter of the maximum outer cylinder wall diameter of with a tolerance of an RC9 fit which is 0.1 cm [71]. The maximum outer diameter of the piston cylinder is 7.643 cm the hole will be cut to

7.743 cm. An aquajet precision of 0.05 cm will be more than adequate clearance remains for this operation. The circle cut out can then be welded to the cylinder as the top and the remaining portion can be welded to the bottom as the mounting plate.

E. Manufacturing and Costs

Four raw components will be used to manufacture the pump cylinder: two NPT barbed tube fittings (Part number 5058K41), one 1/16-inch-thick six by five plate (Part number 8910K591), and a hollow bar stock (Part number 7767T68) from McMaster-Carr.

The NPT barbed tube fittings will be bought directly from McMaster-Carr and will cost $9.84 and will take approximately 20 minutes to thread into the cylinder. Teflon tape will be used to prevent leakage [74].

The hollow bar stock will be used for cylinder walls. The bar stock will need to be cut to a length of 11.875 cm with a tolerance of 0.265 cm [75]. The inner diameter of the hollow bar stock is 6.67 cm. The interior of the bar stock will need to be cored to a diameter of 7.1 cm. The external diameter of the bar stock is 7.62 cm. This satisfies the failure criteria of the cylinder and does not need to be processed. The bar stock costs $48.25 per foot which corresponds to a price of $1.58 per centimeter. This means the wall of the cylinder will cost $18.80 in raw material. This process is expected to take a machinist an hour to complete.

The five-inch by six-inch plate will have a 7.743 cm diameter hole cut in the middle of it with an Aqua jet. The thickness of the plate is 6.35 mm which is thick enough to satisfy the loading conditions of the top of the cylinder with the tolerance provided of -0.254 mm [76]. The external geometry will be tapped with through holes in the four corners to allow for mounting to the frame. The external geometry will then be welded to the base of the cylinder. The plate costs $13.96. The estimated time for all of these operations combined is one hour for machining and welding.

The top of the piston was manufactured by cutting a steel bar tock with outer diameter 7.62 cm to a height of 16 mm. Two NPT threads will be tapped into the material 25.4 mm apart along the same axis. The thickness of the cap. A 7.62 cm diameter steel bar stock was cited from McMaster Car [77]. The bar will cost $4.58. A cutting operation will be done to cut the stock to 16 mm cutting will take five minutes. A tap hole of 11.11 mm is used for the NPT Threads. The Threads will then be tapped into the piston head and the head will be welded to the rest of the piston assembly. The taping process is expected to take twenty minutes and the welding process will take ten minutes. The manufacturing time for the piston head is thirty-five minutes and will cost $4.59 in raw material

The total cost of the piston cylinder is $48.24, and will take three hours to machine and assemble. Table XXVI summarizes the material and labor costs of the piston cylinder.

Table. XXVI. Summary of the material and labor costs of the piston cylinder.

Pump Inlet and Outlet Hardware

The top of the piston cylinder has two quarter inch NPT tapped holes in it. In one of these holes an Aluminum low pressure barbed fitting will be screwed in with Teflon tape to ensure the seal. An Aluminum low-pressure barbed tube fitting with a quarter inch NPT male thread that connects to a quarter inch inner diameter tube was chosen [78]. The part was sourced from

McMaster-Carr and will cost $4.84. And will take five seconds to screw into the piston assembly.

To attach the outlet hose to the air receiver a female barbed NPT tube fitting was used. The fitting cited from McMaster Carr is for a quarter inch inner tube diameter and a quarter inch female NPT threading [79]. This will be assembled to the inlet fitting with teflon tape to improve the seal. The part costs $9.31 and is estimated to take five seconds to screw into the piston assembly.

Silicone braided tubing was selected to connect the piston outlet to the air receiver inlet. The tubing selected is a medical grade quarter inch inner diameter tube that is FDA approved for use with ingested components. Shore A:65 silicon braided tubing from Granger is rated to a maximum pressure of 1434 Kpa [80]. The tubing costs $26 per meter and one meter will be provided with the pump for a cost of $26.

The tube will be attached to the barbed tube fitting with a press fit and secured by a hose clamp. The hose clamp chosen is for diameters of tubing less than one quarter of an inch [81]. These cost $6.26 for a pack of ten and two will be used for the pump assembly, one at the male barbed fitting and one at the female barbed fitting. The cost of using two of these worm drive clams is $1.32.

The same type of male to male check vale used in the air tank assembly will be used to allow air into the cylinder. The check valve has quarter inch NPT male threads on both sides. It will thread directly into the cylinder. This will leave a male end exposed for the filter cap to thread onto.

A. Pump Filter Cap

To prevent the inhalation of bacteria and particulates, an air filter was placed into the nebulizer. The cylindrical shaped filter is made of a white colored polyurethane foam. At a porosity of 40 pours per centimeter, the foam is effective in preventing the passing of harmful agents while simultaneously adding virtually no resistance to airflow [82].

The filter will, however, accumulate particles over time. To maintain efficient airflow and prevent health hazards, the filter must be replaced when it begins to noticeably discolor which should happen at about 100 treatments cycles [83].

A flat sheet of polyurethane foam will be bought and punched into disks of the correct size. A 304.8 mm by 304.8 mm sheet of foam costs $6.70, this sheet can be punched into approximately 900 disks of 10 mm [84]. The total number of disks will be reduced by ten percent to account for spacing between punches resulting in 810 disks.

The filter cap will be machined into a 5/8 inch Hex stock made of FDA approved Nylon [85]. The hex cap is 18 mm tall and has a 10 mm slot where the filter fits. The filter cap was made of plastic to ensure it fails before the aluminum check valve. The bar stock costs $0.04 in raw material for the filter cap. The filter cap will take approximately 25 minutes to cut and machine.

Table XXVI summarizes the raw material and labor costs of the inlet and outlet hardware.

Table XXVI. Summary of parts and labor for the hardware inlet and outlet fittings.

Pump Frame

The frame provides the structure to the mechanical function of the pump system. The frame was designed to give the pump stability during operation and keep the piston cylinder aligned during the pumping process. The frame was designed from 6061 Aluminum hollow bar stock and was designed as a two-dimensional truss system. For the final assembly a ½ inch square hollow bar was selected from McMaster Carr with a part number 6546K51 [86].

To determine the required dimensions of bar stock two calculations were preformed the first was related to the required thickness of the bolts that attach the pump cylinder and the second pertained to the material strength of the frame itself.

At maximum loading conditions the piston exerts a 2668.93 N. with four bolts used to attach the force acting on each bolt is 677 N. Grade 5 strength steel bolts were used for this calculation. The endurance limit of the bolts was assumed to be half of the yield strength of the bolts, this is a conservative estimate as half of the ultimate tensile strength is typical for steel alloys. Equation (111) was derived to calculate the required bolt diameter to prevent failure.

$$d = \sqrt{\frac{4 \cdot P}{\pi \cdot \sigma}} \tag{111}$$

The required diameter of the bolt is 2.844 mm. A ¼-20 UNC bolt has a minor diameter of 4.8 mm. With this bolt diameter a half inch bar stock will allow approximately one bolt diameter on each side. The mounting screws used to attach the piston head to the frame is ¼-20 Zinc Plated Grade Steel Hex Bolt with a part number of 91247A544 [87]. These correspond to a ¼-20 Zinc Plated Grade Steel Hex nut with a part number 95462A029 [88]. The bolt will be assembled with a ¼ inch polycarbonate plastic washer [89].

The maximum load this system is ever expected to experience is produced by the pivot connection. The maximum possible load can be 3336 N, the load will be distributed evenly to the two frame walls which creates s force of 1668 N. Analyzing bar for failure modeled by the dynamic loading approximation the area of the bar was required to be 45.29 mm. The hollow bar stock with a half inch outer dimension and 3/8-inch inner dimension has an area of 70.52 mm^2 which is more than enough to ensure the frame will last for the 10^8 equivalent pump cycles based on the Goodman equation.

The frame was evaluated as a two-dimensional truss system under half of the maximum load that the entire truss system is expected to undergo. Fig. 62 below shows the loading profile on the system as a free body diagram and was used to calculate the reaction forces on the system. The reaction point at the bottom left was assumed to be zero as the device will rotate towards the third reaction point.

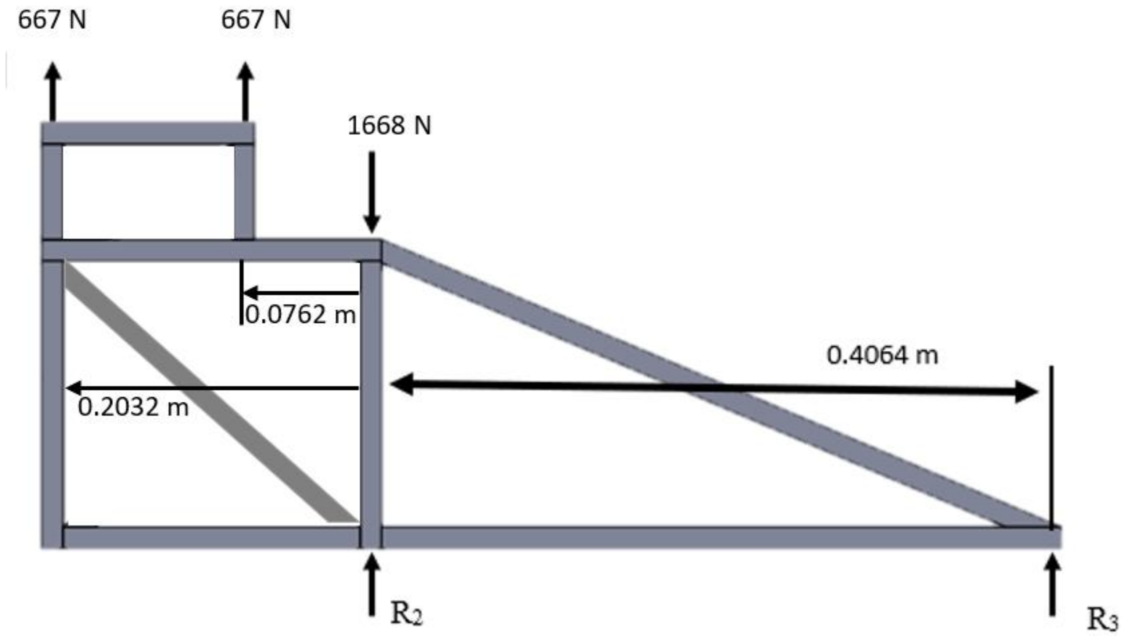

Figure 62: Free body diagram of the 2-D truss system under maximum load conditions.

By summing the forces in the y direction and taking the moment about reaction point 2 equations (112) and (113) were derived.

$$\sum M_2 = 0 = -185.7595\ N \cdot m + 0.4064\ m \cdot R_3 \tag{112}$$

$$\sum F_Y = 0 = 667\ N + 667\ N - 1668\ N + R_2 + R_3 \tag{113}$$

The equations above were used to solve for the reaction force at point three and point two. The reaction at point three is 457 N and the reaction at point two is 123 N. With this loading profile the truss system was analyzed to determine the maximum load on each element which was then converted to stress and the failure criteria was checked for each element. Figure (63) shows the element labels used for the truss.

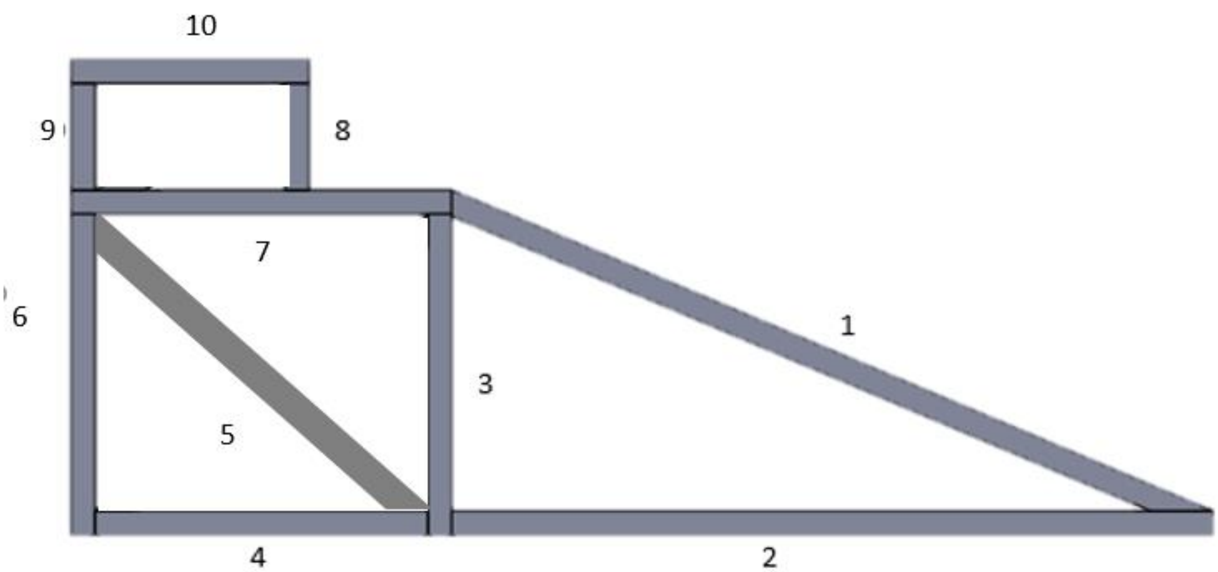

Figure 63: Node Label for the 2-D truss system.

Table XXVII shows the calculated loading and stresses acting on each element within the system.

Nodal Truss Analysis of 2-D Frame

Element	Load	Element Stress
1	1207 N	17.1 MPa
2	1111 N	15.7 MPa
3	123 N	1.7 MPa
4	1111 N	15.7 MPa
5	1545 N	21.9 MPa
6	667 N	9.4 MPa
7	143 N	2.0 MPa
8	667 N	9.4 MPa
9	667 N	9.4 MPa
10	0 N	0 MPa

Table XXVII The element analysis for the 2-D truss system.

The load stress in each element is much less than the dynamically calculated load stress. Therefore, the bar stock used for the frame assembly is adequate to ensure the rigidity of the structure.

Through slots were machined into the aluminum bar stock for mounting of both the piston and the U-Clamps which will secure the pivot rod. Through slots were used instead of through holes to allow calibrations to be made for uncertainties in the assembly process. The diameter of the slot is equal to a close fit for a through hole for a quarter inch bolt which is 6.53 mm. The total height of the slot is two times the diameter of the hole to allow for the parts to slide in order to be aligned for proper use. The total height of the slot is 13.06 mm. The same slot was cut into element seven to allow the pivot rod to be adjusted to account for discrepancies in angle. The rotational bearing used corrects for errors up to five degrees as well. For all of these slots' washers are used to reduce the effects of stresses at the interface of hex nuts a quarter inch washer was cited from McMaster Carr [90].

To clamp the pivot rod to the assembly two U-Bolts were purchased from McMaster Carr. The U-Bolds selected come with a mounting plate and have a part number of 3201T48 [91]. These U-Bolts have 1/4 – 20 UNC threading and mount to element seven to ensure pump operation.

Bar Stock Initial Cuts List

Length	Number of Parts	Post Operations Needed
57.15 mm	4	N/A
127 mm	2	Through Slots for Piston
127 mm	12	N/A
116.56 mm	2	Angles for Cross Beam
441.08 mm	2	Angles for Cross Beam
171.45 mm	4	N/A
406.4 mm	2	N/A
177.8 mm	2	N/A
203.2 mm	2	Through Slots for U-Bolt
246.88 mm	2	Angles For Cross Beam

Table XXVIII: Primary cuts needed for the truss system.

A. Manufacturing

The first step in the frame assembly is cutting the bar stock to size. This will be done with a bandsaw to obtain the specified number of each length of aluminum bar stock. After the raw lengths are cut two of the 127 mm long bars will be slotted on an endmill for the connection slot for the piston, and the 203.2 mm long pieces will be slotted for the U-Bolt connection. The angles on the cross-beam sections then need to be cut. After the cuts and slots are made the parts are ready to be assembled.

The estimated time to cut an individual piece of bar stock to length is three minutes per cut. The total time spent making raw cuts is expected to be 34 minutes. The slots are expected to take 10 minutes per bar. With four bars being slotted the slotting operation is expected to take 40 minutes. The angle cuts are expected to take 10 minutes per bar as well with six bars being angled this operation will take one hour. At this point the bars are ready to be brazed together. The actual parts used for each assembly are discussed in the assembly drawing. The estimated time to braze one joint is five minutes including time it takes to locate the parts. There are 36 joining operations that need to be performed which will take three hours. The total estimated time it will take to assemble the frame is five hours and fifteen minutes.

The total length of the bar stock used is 5876.24 mm. It may be purchased from Mcmaster Carr [86] at a price of $2.16 per foot the total raw material cost of the bar stock is $41.64. The cost of the U-Bolt used to clamp the pivot to the frame costs $1.09 for two of them the total cost is $2.18.

Connecting Rod

A. Geometric Constraints

When the pump lever is oriented horizontally at mid-stroke, the connecting rod is vertically aligned with the piston cylinder centerline. As the pump lever rotates, however, the connecting rod base will experience a horizontal displacement, introducing the potential for interference with the inner diameter of the piston cylinder. Therefore, it is necessary to develop a relationship between the required stroke length (*SL*), the short end of the pump lever (L_2), and the horizontal x-displacement (*x*). At the highest point in the lever's stroke, the following right triangle is formed by the aforementioned dimensions:

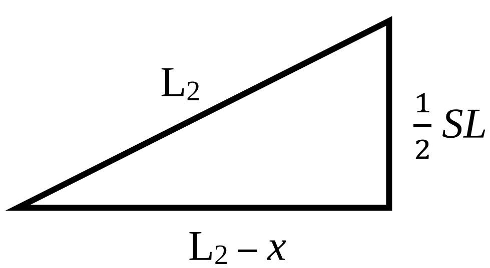

Fig. 63: Geometry relating x-displacement, stroke length, and partial lever length.

Once the triangle is constructed, the Pythagorean theorem is used to equate these dimensions and iteratively solve for *x*:

$$L_2^2 = (L_2 - x)^2 + \left(\frac{1}{2}SL\right)^2 \qquad (114)$$

When rearranged and multiplied out, the L_2^2 terms cancel, and the form of the relation becomes a quadratic equation:

$$x^2 - 2L_2x + \frac{1}{4}SL^2 = 0 \qquad (115)$$

Notably, the stroke length is determined by the volume and diameter of the piston cylinder, so the primary parameter to be manipulated is the partial lever length L_2. Using a length of 10.16 cm for both L_2 and *SL*, the roots of the quadratic equation are found, and the smaller term is taken as our *x* horizontal displacement. In this case, *x* is determined to be 13.77 mm.

In order to evaluate for the possibility of interference between the connecting rod and the piston cylinder, the sum of our derived *x* value plus half the side length of the connecting rod cross section should be less than cylinder's inner radius. Because the side length is also dependent on our 11.11 mm bearing width, a square connecting rod with a 25.4 mm by 25.4 mm profile (The standard size of 1 inch by 1 inch in English units) is an appropriate selection, leaving 7.15 mm of rod material on each side of the bearing while also allowing for 8.58 mm of horizontal clearance, given a 70.1 mm inner diameter for the cylinder.

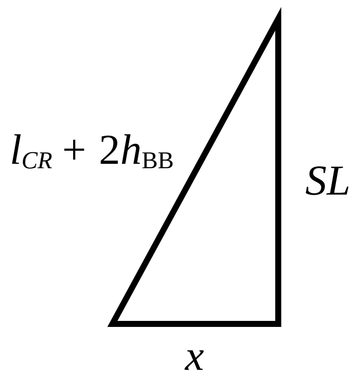

Fig. 64: Geometry relating x-displacement, stroke length, rod length, and bearing base height.

Having established that the relationship between L_2, stroke length *SL*, and the horizontal *x* displacement will not result in interference, a second triangle can be constructed in order to determine the required connecting rod length to meet these requirements:

Notably, the length of the rod is effectively extended at both ends by the offsetting height of the bearing base h_{BB} (approximately 5.95 mm each). The same Pythagorean identity is then used to solve for the only unknown parameter, the connecting rod length l_{CR}, which yields a dimension of 90.63 mm.

B. Static Loading

In designing the rod extending from the shorter end of our pump lever to the base of the piston, it is required that a diameter appropriate for the magnitude of stress acting on the rod is selected, so a relationship between yield stress and area can be developed:

$$\frac{\sigma_y}{N} = \frac{F}{A} \qquad (116)$$

Where σ_y is yield stress, N is a factor of safety, F is the force expected to act downwards on the rod, and A is the cross-sectional area of the rod. A factor of safety between 2-3 is commonly used for piston design in the automobile industry, and the connecting rod is a relatively non-critical component, so a factor of safety of 2 can be selected [P3]. In the interest of consistency in material selection, we use Aluminum 6061-T6, which has a yield strength of 276 MPa [P4]. We can rearrange this relationship and plug in this yield strength to solve for cross-sectional area A as:

$$A = \frac{FN}{\sigma_y} = \frac{(2668.9\text{ N})(2)}{(276\text{ MPa})} = 1.933 \cdot 10^{-5}\, m^2 = 19.33\, mm^2$$

Which, in the case of a square-shaped aluminum bar stock, implies a minimum cross-sectional side length of 4.40 mm, considerably lower than the 25.4 mm width of our selected bar size.

It is also important to ensure that relationship between the rod diameter and its length is such that buckling will not occur. Columns fail by buckling when their critical load is reached, and this load can be approximated by using the Euler column formula:

$$F_{cr} = \frac{n\pi^2 EI}{L^2} \qquad (117)$$

Where F is the allowable load in Newtons, E is the modulus of elasticity, L is the length of the column, I is the moment of inertia in m^4, and n is a integer factor accounting for the end conditions of the column [P5]. In this case, because both ends of the connecting rod rotate freely, the conservative condition of n = 1 is selected.

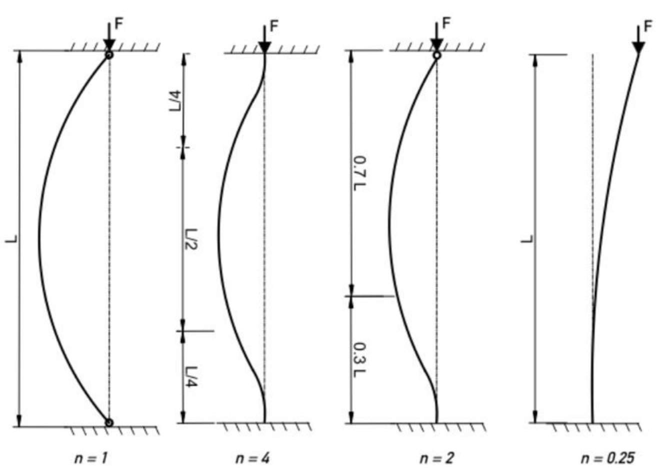

Fig. 65: End conditions determining n factor for an Euler column [P5].

The area moment of inertia of a solid square bar is determined with respect to the geometry of the rod's cross section [P6]. As a result, it is found as a function of side length a:

$$I_x = \frac{a^4}{12} \quad (118)$$

Plugging in the 25.4 mm side length of our bar stock, the area moment of inertia is found to be $3.469 \cdot 10^{-8}\ m^4$, which can then be used to check for the critical load of buckling:

$$F_{cr} = \frac{n\pi^2 EI}{L^2} = \frac{(1)\pi^2(68.9\ \mathrm{GPa})(3.469\ \cdot 10^{-8}\ \mathrm{m}^4)}{(0.09063\ \mathrm{m})^2} = 2871966\ N = 2871.97\ kN$$

Compared to the expected load of 2668.9 N, the critical load of buckling is several orders of magnitude above this threshold, indicating that geometric constraints related to the bearings— rather than failure conditions— are more important in driving the connecting rod dimensions.

C. Dynamic Loading

In addition to evaluating the critical load under static conditions, the cyclically loaded nature of this component warrants the calculation of dynamically equivalent stresses using the modified Goodman equation (26) for 10^6 cycles. Aluminum 6061-T6 has a fatigue strength of 96.5 MPa and an ultimate tensile strength of 310 MPa, so this equation yields an alternating stress of 36.8 MPa [P4]. Plugging this back into our previous equation for minimum cross-sectional area, we find:

$$A = \frac{FN}{\sigma_y} = \frac{(2668.9\ \mathrm{N})(2)}{(36.8\ \mathrm{MPa})} = 1.450 \cdot 10^{-4}\ m^2 = 145\ mm^2$$

Which implies a minimum side length of 12.04 mm, still significantly below our 25.4 mm side length for the rod.

E. Connecting Rod Bearings

Because we desire the force transmitted from the piston head to the connecting rod to act directly through the centerline of each component, the bearings at each end must fit within the internal profile of our connecting rod's top and bottom cross-section. Furthermore, the bearing will be used to mount the connecting rod to the face of the piston head, so the longest dimension of the bearing should be no larger than the diameter of the piston head. Therefore, the selection of a suitably small bearing size will serve as the primary driving force in designing the corresponding cross-section of our connecting rod.

In addition to selecting a set of sleeve bearings with the smallest possible length dimension, it is also necessary to ensure that the chosen bearing can withstand a radial load equivalent to the force exerted by the piston head. With a piston head area of 38.71 cm^2, the maximum possible tangential force applied to the radial portion of the bearing will be approximately 2668.93 N.

With these two constraints in mind, a dry-running mounted sleeve bearing with part number 2820T3 is an appropriate selection [P1]. According to the manufacturer's specifications listed on McMaster-Carr, the dynamic radial load capacity at 120 rpm is rated for 690 lbs., equivalent to 3069.27 N. Additionally, the longest dimension of the sleeve bearing is 57.15 mm, which sets a reasonable minimum diameter for the piston head diameter. Finally, the width of the bearing through which our rod will be fastened is 11.11 mm, which implies the connecting rod must be substantially larger than this size in terms of cross-sectional side length.

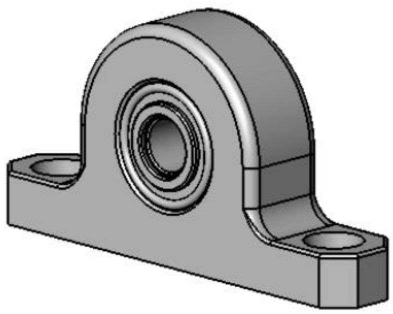

Fig. 66: Isometric view of dry-running sleeve bearing for connecting rod ends [P1].

D. Connecting Rod Bearing Fasteners

The sleeve bearings require fasteners in order to be mounted on both the pump lever and the piston head. The manufacturer specifies holes with a diameter of 9/32 inch, equivalent to 7.14 mm in metric units, so an M7 screw can be used. Furthermore, the height of the bearing's base is 5.95 mm, so the length of the threaded portion of the screw must be greater than that length, in addition to at least 5 times the screw's thread pitch to ensure proper engagement. The nut material being threaded into is aluminum on both the top and bottom bearings, and coarse threads are generally stronger when the nut material is weak relative to the bolt, so with the selection of a steel bolt coarse threads are the most appropriate. A set of Medium-Strength Class 8.8 Steel Hex Head Screws will fulfill all of the aforementioned specifications, and are available with a 16 mm threaded length, which will adequately exceed the recommended threshold for thread engagement [P2].

E. Connecting Rod Fasteners

In fastening the connecting rod to each sleeve bearing at both ends, a bolt must be chosen that can safely withstand the radial force exerted in the condition of double shear.

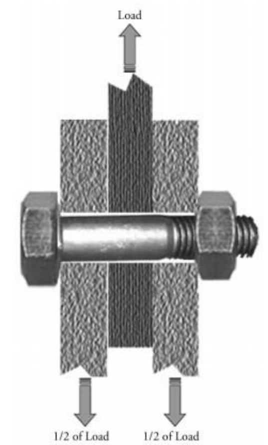

Fig. 67: Diagram depicting bolt loaded in double shear [P7].

The force will act across the cross-sectional area of the bolt in the form of a shear stress [P8]. This shear stress can be calculated with the following relation:

$$\tau_{ave} = \frac{F}{2A} = \frac{2F}{\pi d^2} \quad (119)$$

The bearing hole diameter is 6.35 mm, which implies the largest metric bolt diameter that can be selected is M6. Furthermore, the unthreaded shank length of the bolt must span nearly the entire depth of the connecting rod, indicating this dimension should closely match the rod's side length of 25.4 mm. With respect to these constraints, a stainless steel shoulder screw with a 6 mm shoulder diameter and 25 mm shoulder length (McMaster-Carr part number 90833A116) is an appropriate selection [P9].

Fig. 68: Isometric view of stainless steel shoulder screw [P9].

Performing the shear stress calculation with the inclusion of the selected shoulder screw's diameter, the expected shear stress is found:

$$\tau_{ave} = \frac{2(2668.9\ \mathrm{N})}{\pi(0.006\ \mathrm{m})^2} = 47.20\ \mathrm{MPa}$$

According to the manufacturer's specifications listed on McMaster-Carr, the minimum shear strength of this fastener is 35,000 psi, equivalent to 241.32 MPa, so this fastener will easily withstand the amount of shear stress expected [P9].

The threaded portion of the bolt is secured using a nut of the same diameter. Unlike the unthreaded shoulder portion of the bolt, the threaded diameter is only 5 mm. The total length of the threaded portion is 8.8 mm, and as a rule of thumb, the number of threads engaged should be at least 5. Therefore, at a 0.8 mm thread pitch, the height of the nut should be at least 4 mm.

Because of this threading requirement, a M5 × 0.8 mm "high" hex nut is chosen with a 50% taller gripping area for increased thread engagement, as well as ease of installation. The height of this nut is 5 mm [P10].

Fig. 69: Isometric view of steel high hex nut with M5 × 0.8 mm size [P10].

F. Manufacturing

The connecting rod is manufactured from a standard 1 inch by 1 inch square bar stock, which must be cut to the 90.63 mm size in length. Although cutting processes have relatively poor tolerances, the dimensions of the piston cylinder are designed to be large enough to ensure the rod will not interfere with its inner diameter if it is slightly too large, and charge time for the pump will only be affected by a small amount if the rod is too short, so loose tolerances are allowable. The expected time for an experienced machinist to mark and cut this part in a bandsaw is approximately 5 minutes.

In order to mount the connecting rod to the sleeve bearings at each end of its length, two thru holes must be drilled to allow the insertion of an M6 (6 mm) shoulder screw. This shoulder portion of the screw will also serve as a pivot point, which means the clearance desired for this hole is that of a free fit. Referencing the tap and drill chart, the corresponding hole size is 6.60 mm, with the closest American drill being size G. The process of correctly positioning and drilling these thru holes is expected to take up to 12 minutes.

Once the bar stock is cut to size and thru holes are drilled at each end, the final step in manufacturing the connecting rod is using a mill to create the two internal slots in which the bearings will sit. The bearings are both self-aligning up to 5 degrees, so typical tolerances associated with milling processes are also allowable in positioning the slots relative to one another. Chamfered edges are also desirable in order to reduce the number of sharp edges and to minimize the possibility of interference with the bearings upon rotation. The expected time spent on positioning and milling the slots is expected to be approximately 20 minutes, with an additional 15 minutes spent chamfering and deburring the rod edge.

Assembly Procedure

Pump

For the formation of this nebulizer, the air tank and pump will be considered two separate assemblies that come together for operation in the end. The first section of data quantifies the required time to handle and assemble the pump assembly. The pump assembly is further partitioned into three sub-assemblies of its own: the frame assembly, the lever assembly, and the piston assembly. The frame is welded during the manufacturing process and is therefore treated as a single part for the purpose of this section. The same is true for the welded piston components.

Once the frame assembly is welded, the lever assembly is mounted to it using two U-bolts, and the piston assembly is attached using standard bolts and screws. The connecting rod, which acts as an intermediary component between the piston and lever for the purpose of force transmission, is mounted to each surface with sleeve bearings. The sleeve bearings allow the free rotation of the connecting rod to account for the relative horizontal displacement that occurs as the lever rotates.

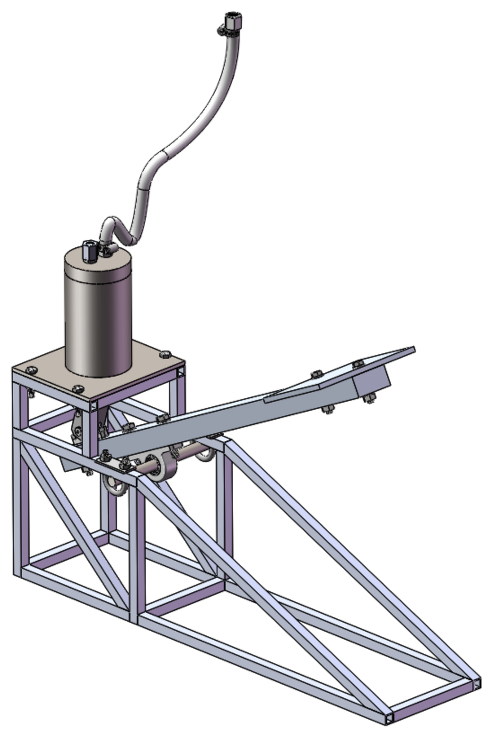

Fig. 70: Isometric view of pump frame and piston assembly.

Air Tank

For the air tank, the inlet piece will be welded on the hemisphere and checked for leaks. The inside weld must be cleaned. Next the hemispheres are welded to the cylinder as specified in the manufacturing section. Assembly of the piping and valves will take place separately and is outlined in the assembly and handling time table in the Handling Time and Assembly Time section of this report.

Due to the nature of NPT threads the exact dimension and position of the components is uncertain. For this reason the sub-assembly is assembled and installed on to the air tank prior to the legs being welded, as to ensure the piping is horizontal as shown in all assembly drawings. The valves and pipes are assembled as shown in drawing TANK-A02. All threads must be securely fastened and checked for leaks. The tank stand and pipe stands are then welded in place depending on the final position of the valves and pipes.

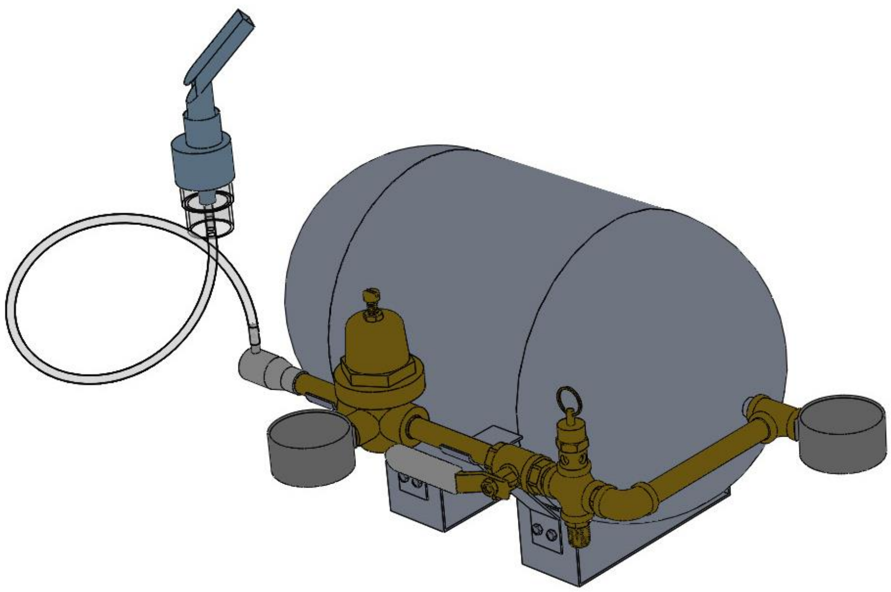

Fig. 70: Isometric view of tank and piping assembly.

Table (XXIIX). Values used to calculate the total assembly/handling time of the tank subassembly. [1] Part cannot be manipulated easily, not easy to align and screw into position. [2] Assuming 0.85 mm/s (2 in/min) weld time. [3] Part can be easily located, holding down requires for subsequent processes, not easy to align or position. [4] Part not easily located, not easy to align

Manufacturing

Assuming $90 an hour for labor costs and manufacturing times based off of Michael Braddock's time estimations for manufacturing. Material costs are based off of the smallest necessary amount one can buy of the needed stock to make their part, then divided by the needed mass. The diffuser manufacturing cost was assumed to be 3D printed in Marston Library with a cost of $0.15/gram. Other manufacturing processes

AIR TANK MATERIALS AND LABOR

Part Names	Part #	Manufacturing Cost ($)	Material Cost ($)	Total ($)
TANK BODY	TANK-P-001	112.50	39.78	152.28
TANK DOME 1	TANK-P-002	0.00	62.75	62.75
TANK DOME 2 (has the hole)	TANK-P-003	22.50	62.75	85.25
TANK INLET	TANK-P-004	79.50	2.26	81.76
TANK STAND	TANK-P-005	37.5	6.63	44.13
PIPE STAND	TANK-P-006	45.00	6.63	51.63
HOLLOW ROD	TANK-P-007	25.00	0.58	25.58
DIFFUSER	TANK-P-008	1.50	3.00	4.50
Total				504.88

Pump

FOOT PUMP MATERIALS AND LABOR

Part	Part Number	Manufacturing Cost	Material Cost	Total
CONNECTING ROD	PUMP-P004	78.00	5.28	83.28
CONNECTING ROD SLEEVE BEARNG	PUMP-OTSO1	n/a	24.86	24.86
SHOULDER SCREWS	PUMP-OTS12	n/a	11.50	11.50
M5 HEX NUTS	PUMP-OTS 13	n/a	7.51	7.51
M7 HEX HEAD BOLT	PUMP-OTS11	n/a	6.45	6.45
6061-T6 Al Rectangular Stock	PUMP-P-001	60.00	35.68	95.68
6061-T6 Al Sheet	PUMP-P-003	30.00	6.74	36.74
1023 Carbon Steel Rod	PUMP-P-002	10.50	3.70	14.20
Pillow Block for Lever Pivot	PUMP-OTS02	n/a	23.19	23.19
M7 Bolt	PUMP-0TS11	n/a	1.42	1.42
Washer	PUMP-OTS08	n/a	0.59	0.59
NPT Barbed Tube Fitting	PUMP-OTS03	n/a	9.84	$9.84
1/16” 6”x5” Steel Plate	PUMP-P-012	90.00	13.96	103.96
PISTON HEAD	PUMP-P-005	52.50	4.58	57.08
PISTON CYLINDER	PUMP-P-14	90.00	18.80	108.80
Silicone Braided Tubing	PUMP-OTS06	n/a	26.00	26.00
Hose Clamp	PUMP-OTS-O5	n/a	1.32	1.32
FILTER	PUMP-P-015	n/a	6.70	6.70
FILTER CAP	PUMP-P-016	37.50	0.04	37.54
TOTAL				656.3

Total Cost of the Nebulizer

Sub-assembly	Assembly and Handling Costs	OTS Part Cost Total	Manufactured Part Cost Total	Total
Air Tank	13.89	253.77	504.88	772.54
Pump	10.94	368.91	287.4	298.34
Total				1070.88

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