Innovative Alternatives to Lifting Overturned Military Vehicles

Team Members: C1C Rebecca Bauman, C1C Jason Hodges, C1C Will Fleming, C1C Max Jones, C1C Kevin Tanous, C1C Mark Jensen, C1C Alex Pecci, C1C Jake Risma, C2C Kenner Roberts
Team Advisor: Capt Michael Knauf
United States Air Force Academy

Introduction

Air Force Pararescue is tasked with rescuing the lives of downed soldiers, sailors, airmen, and marines. In many situations, the victim is trapped underneath large vehicles and other heavy objects. For instance, Mine-Resistant Ambush Protected (MRAP) vehicles, which can weigh up to 55,000 lbs, have in the past overturned, trapping people and sensitive equipment underneath. The current means of lifting the large weights is to use large and bulky pneumatic lifting bags. In an emergency situation when victims are often in need of immediate medical care, rescue personnel may not have time to use current lifting bag technology to lift the vehicles off of the victims. Instead of using these slow bags however, in many cases, the swiftest and most effective course of action is to amputate appendages in order to transport victims to medical care. Identifying the necessity to develop a better solution, the Air Force Research Laboratory solicited ideas from multiple universities to create a device that will lift a 45,000-55,000 lb object up to 24" while being packaged in a volume no greater than 12"x12"x6". The USAFA Service Academy Challenge team has developed a means of solving the problem of quickly lifting the required weight to an acceptable height from a portable device.


Figure 1: MRAPs in Combat Environment

The team has developed two solutions to the problem. The first design is a rigid chain composed of several interlocking chains pushed together by a ball screw/worm gear combination. The second design is composed of nine telescoping segments lifted by a hydraulic pump.

Technical Analysis

Each design requires comprehensive analyses through the engineering design process. Both designs share multiple common elements, including the area of the base required to ensure the design can distribute the loading on the ground evenly as well as ensuring the top plate which attaches to the top of each lift can support the required weight. Through a soils analysis, it was determined that an average Afghan soil should be able to support 30,000 lbs if a surface area of approximately 4.4 square feet were provided.

Report Documentation Page

Form Approved
OMB No. 0704-0188

1. REPORT DATE **2014**	2. REPORT TYPE	3. DATES COVERED **00-00-2014 to 00-00-2014**

4. TITLE AND SUBTITLE **Innovative Alternatives to Lifting Overturned Military Vehicles**	5a. CONTRACT NUMBER
	5b. GRANT NUMBER
	5c. PROGRAM ELEMENT NUMBER
6. AUTHOR(S)	5d. PROJECT NUMBER
	5e. TASK NUMBER
	5f. WORK UNIT NUMBER
7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES) **United States Air Force Academy,Washington,DC,20301**	8. PERFORMING ORGANIZATION REPORT NUMBER
9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)	10. SPONSOR/MONITOR’S ACRONYM(S)
	11. SPONSOR/MONITOR’S REPORT NUMBER(S)

12. DISTRIBUTION/AVAILABILITY STATEMENT
Approved for public release; distribution unlimited

13. SUPPLEMENTARY NOTES
The 2014 National Security Innovation Competition,April 25, 2014, Colorado Springs, CO

14. ABSTRACT

15. SUBJECT TERMS

16. SECURITY CLASSIFICATION OF:			17. LIMITATION OF ABSTRACT	18. NUMBER OF PAGES	19a. NAME OF RESPONSIBLE PERSON
a. REPORT **unclassified**	b. ABSTRACT **unclassified**	c. THIS PAGE **unclassified**	**Same as Report (SAR)**	**5**	

Standard Form 298 (Rev. 8-98)
Prescribed by ANSI Std Z39-18

Following this, the worst case distributed load for the top plate assumes a "wedge" distribution. In this scenario, assuming the top plate is welded to the top of the lifting device, only half of the weld will feel the load. Analyzing this weld, the max force calculated came to be 4,840lbs. Because of this, the top plate interface is being redesigned to incorporate a ball and socket type connection versus the originally-planned welded connection.

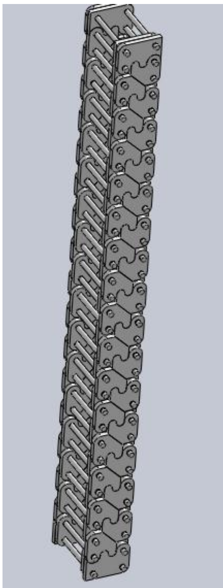

Figure 3: SolidWorks Rigid Chain Design

The design process and technical analysis of the rigid chain began by modeling the system as parallel beams with no connecting pins, assuming the top connecting point is free while the bottom is fixed. This provided the ability to perform an Euler buckling analysis and design an optimized geometry for the rigid chain, ensuring the lowest possible weight of approximately 11.6 lbs for the chain itself. High-strength steels, aluminums, and titaniums were all investigated, though steel was the optimal material because of its high modulus. Two Finite Element Analyses of this design were conducted to provide corroborate hand calculations and give an estimate of the buckling and weight carrying capacity of the design. Determining the size of the pins was required through a shear stress analysis with the assumptions that pins are not load- bearing and the load is divided into eight points on the pins. This gave a requirement for high-strength steel pins 4" long and ¼" in diameter. After testing and overwhelming an aluminum mockup of this design, however, it was determined that the pins are indeed loaded significantly and at only two points in the design. Therefore, 3/8" high-strength steel pins were redesigned into the system to allow for adequate load carrying capacity.

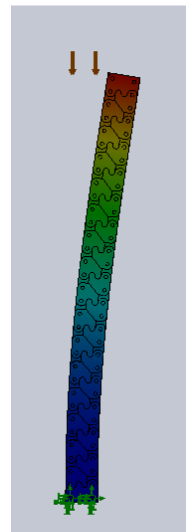

Figure 2: SolidWorks Rigid Chain FEA

Several systems must be joined together for either design to be functional. Initially the chain was going to be raised through the use of specially-designed sprockets, for which a stress analysis was required. Ultimately, however, the idea of using sprockets was replaced with the idea of using ball screws because sprockets required massive torques and adequate speed reduction far exceeding plausible commercial systems. It was determined around this time that no commercial system could provide the torque required within an acceptable weight range. At this point the team decided to attempt to use this system to lift 10,000 lbs, keeping all other constraints the same. This afforded the team some flexibility, but the sprocket concept still required far too much torque to operate. Therefore ball screws will be used to "push" the chain from their back ends and the speed reducers to turn the screws for the gearing of the chain will be of a more manageable design with specifications that will meet necessary capabilities. The system still requires 2500 in-lbs of torque, and a commercial worm gear system is being procured which can provide the required torque/power combination. Assuming fifty percent efficiency from the overall system, to raise this load the desired height within three minutes (a self-imposed timeline), 0.8 HP is required for each motor, equivalent to 600 watts of power. To produce this, high-powered remote-controlled aircraft motors were procured to supply the required power input. These motors are controlled

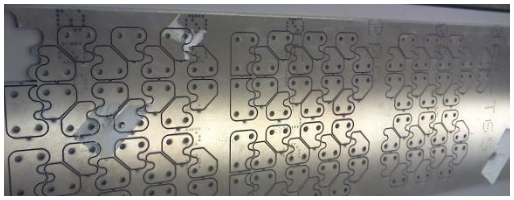

Figure 4: Rigid Chain Pins Waterjet

through the use of a microcontroller providing the motor controller the necessary pulse-width modulation signals through the use of a simple potentiometer. This allows the user to have a safe stand-back distance through long, small wires, and gives the user great control over the lift. The motors are powered by high performance 6.0 amp-hour lithium-polymer batteries capable of a discharge rate of 25C. The entire system has been meticulously designed for efficiency, size, weight, and performance.

Initial testing of the prototype involved using a SATEC load frame to apply a load. As previously stated, during the first test the design failed because of the pins shearing at 2600lbs. After recalculating the failure with the corrected assumption, it was determined the pins failed at 45.7ksi, while the shear strength of the pins was calculated to be 45 ksi. This close correlation confirmed the pins are indeed loaded. Future testing is still necessary as the team needs to test the design in multiple scenarios. The efficiency of the gearing as well as strength of the entire assembly will need to be determined as the team transitions from using Aluminum prototypes towards a final product designed from steel.

Figure 5: Rigid Chain Initial Prototype

The second design, the Hydraulic Lift, is also composed of several subsystems. The initial "cup" design required calculating the cross-sectional area which in turn provided the outer radius of the topmost, internal cup. The cross-sectional area was determined by dividing the internal working pressure by the load. The first cup is sealed at the bottom while cups two through nine are hollow. The thickness of the cups is designed according to operating pressure requirements for lifting 30,000 lbs [2]. Current bottle-jacks have a working pressure of approximately 10,000psi. Optimized for weight, the Hydraulic Lift has a working pressure of just 3,000psi. Calculating thickness involved using an equation for hoop stress, but the ASME 2013 Boil & Pressure Vessel code also provided guidance instead of the normal hoop stress equation [1]. For example, in the ninth tier, the equation for hoop stress yielded a required thickness (using high-strength steel) of 0.100", while the ASME equation yielded a very similar thickness of 0.109", a difference of 8%. To ensure a conservative design, the ASME code's thickness calculations were therefore used. After these analyses were conducted, buckling was investigated assuming the cylinder is simulated to be cantilevered on one side and free on the other, and that the full weight is acting in line with the system. Euler buckling analysis was completed and the analysis determined a very low modulus, attainable easily even with aluminum, was required to ensure even the top-most geometry would suffice. Moving from the main system, the first subsystem analyzed for the lift was the pump. To ensure the lift occurs within the required three minutes, the pump requires a threshold

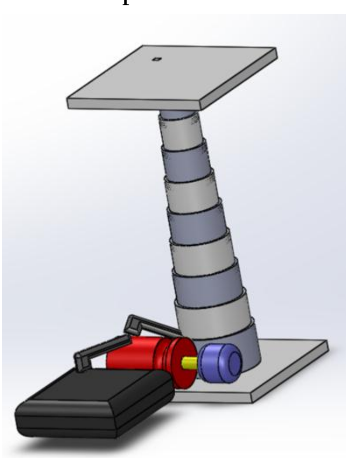

Figure 6: Hydraulic Lift Concept

flow of 0.5 gpm and max operating pressure of 3000psi. A commercial pump was identified allowing for a flow of 0.69gpm and max pressure of 8000psi. This reciprocating pump will lift the full height in 2.86 min. While the motor supplies 7445rpm, the required pump input is 1800 rpm; this requires a gear ratio of approximately 4:1. A planetary gear set has been identified which can easily provide the required speed reduction/torque multiplication. In order to control the system, a similar motor/microcontroller/battery system to the rigid chain will be used.

Figure 7: Hydraulic Lift Initial Prototype

Implementation Readiness Analysis

In order to implement either one of the designs several factors must be considered. Packaging either of these designs requires a focus on equipping the rescue personnel. Air Force Pararescue Jumpers in particular are already outfitted with a plethora of gear, so any extra equipment must be as lightweight and compact as possible. One idea for the implementation of these designs is to include them with any large military vehicle; this would ensure that the equipment would be on hand when needed, but will not add gear for rescue personnel. One innovative idea involves supplying the hydraulic lift with vegetable oil, which is both cost-effective and safe to use in any situation.

Currently, pneumatic lifting bags are used to lift heavy objects such as MRAPs, and an advantage is that the bags provide plenty of surface area for the load to be distributed more evenly, while also providing marginal portability when the bags are deflated. The two new innovative designs are meant to provide greater portability while still being an economical option.

Recommendations

Analysis of the two prototypes is still in an early phase and further testing will highlight each design's strengths and weaknesses. Air Force Research Labs (AFRL) is the primary sponsor for the project, having contributed over $30,000 to the team's budget. While the team has developed a robust budget that will allow for project completion, the intent of the project is to demonstrate feasibility. Should AFRL select the Air Force Academy's designs, further research will need to be conducted to develop a system that is deployable. The final product presented by the Air Force Academy will likely need to be ruggedized to combat the harsh environments in which the product is expected to operate.

References

[1] The American Society of Mechanical Engineers. *ASME Boiler and Pressure Vessel Code*. Tech. New York: ASME International, 2013. Print.
[2] Roylance, David. *Pressure Vessels*. Tech. N.p.: Massachusetts Institute of Technology, 2001. Print.