

# AN EVOLUTION IN THE DESIGN OF THE DEEPEST MINING DREDGE IN THE WESTERN HEMISPHERE

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## ABSTRACT

This paper will take us through the steps involved in the design and manufacture of the deepest mining dredge in the Western Hemisphere. The paper will detail the original equipment types reviewed during the feasibility study through the final choice of equipment selected to meet the project goals. Details will be provided that led to the decision process to justify the dredge including; for sale aggregate inventory, future permitting problems, economical factors, simplified equipment, etc. The paper will bring us through the life cycle from a paper sketch to an engineering challenge that evolved into an 18+ month design and manufacturing process to build a mining dredge that spans the length of a football field. The paper will go into detail on the design challenges that needed to be considered to accommodate the end user including; dredge platform stability as center of gravity shifted with dredging depth, incorporation of standardized major components, challenges to overcome in the event of a major cave-in on the ladder, emergency back-up power, ladder hoisting redundancy, transportation, assembly, erection and commissioning. The paper will also discuss some challenges encountered with the required long lifespan for the dredge that would be working in a very corrosive environment.

**Keywords:** Gantry, structure, ladder, winch, hull.

## INTRODUCTION

This paper describes the history of a sand deposit Greenfield site established in 1997, that would end up using the deepest mining dredge in the Western Hemisphere for extraction of forty years of sand inventories. Florida Rock Industries was the operating company during of the establishment of this plant in 1997. The location of this facility, Witherspoon Sand Plant, is near La Belle, FL – or South Central Florida. Through a series of geological testing procedures it was determined that usable sand inventories were available 61m (200') below the water surface. Based on the increasing difficulty in permitting new dredging sites there is a strong movement for aggregate producers to mine deeper. The available inventories are used for golf course sand, masonry sand and concrete sand. An existing dredge, manufactured in 1973, would be used to begin mining the 32 hectare (80-acre) pond but was limited to maximum dredging depth of approximately 21m (70 feet). Knowing the start-up dredge would eventually become limited by dredging depth capability, Florida Rock began researching options to eventually replace this dredge to maximize the available and mine-able inventories. The methods considered for long-term mining equipment included; airlift pump, cable suspended submersible pump, clamshell dredge, hydraulic dredge with suction jet assist and hydraulic dredge with submersible pump. Florida Rock delayed the capital investment until 2007 when they decided to move forward with the purchase of a dual pump hydraulic dredge.

During the lifecycle of the design and manufacturing process for the dredge, Florida Rock would be acquired by Vulcan Materials and the project would continue on as planned with full support from Vulcan Materials. Vulcan had a few basic requirements in their new dredge option that included maximizing the available sand inventories by meeting a 61m (200 ft) dredging depth, a simple low tech design, low variable operating cost for the equipment and a long service life (40 years).

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## MAJOR EQUIPMENT

The dredge pumps were manufactured by GIW with a 350mm suction x 300mm discharge (14 in x 12 in) configuration with a 914mm (36 in) impeller. Due to the sand characteristics the dredge would be piped with a 400mm suction and a 400mm discharge (16 in x 16in) HDPE pipe, still allowing for the dredge pumps to perform near the BEP. The dredge pumps would be supplied with standard base configurations. Both dredge pumps would be powered by 447 kW (600 hp) AC electric motors through a belt and sheave drive arrangement for fixed speed operations, which is a typical configuration for the Vulcan dredges. Because the sand compaction rating was soft a digging jet system would prove to be adequate for the excavating device for the dredge. The jet water would be supplied by two (2) vertical shaft turbine pumps, each powered by a 149 kW (200hp) electric motor. Each digging jet assembly is capable of supplying a flow rate of 69 l/sec (1100 GPM) and pressure of 21 bar (300 psi). The control system is a PLC based operating system capable of monitoring digital and analog inputs, controlling digital and analog outputs, performing automatic loop control, displaying equipment operating information, safety related features and recording historical data. The dredge was equipped with a dual winch hoisting mechanism for the ladder consisting of two (2) 3334 kN (75,000 lb) planetary winches. The forward positing winches were 53 kN (12,000 lb) planetary winches and the stern planetary winch was rated 111 kN (25,000 lb).

## ENGINEERING CONSIDERATIONS

While Vulcan already operated many dredges, this new dredge would prove to be physically four times larger than their standard dredges as shown in Figure 1 and Figure 2. Designing and building a 61m (200 ft) digging depth suction dredge was similar in process to a typical underwater pump, catamaran style dredge, just bigger. It required a ladder of sufficient length to reach the material and support pumping equipment, a floating platform to support the ladder, control room, other deck equipment, a gantry to raise and lower the ladder, and it needed to be modular in order to be constructed and tested at the manufacturing facility in Louisiana and then shipped and reassembled at the sand plant in Florida. Portability would play a key role in the design process as it would influence the overall cross section and length of hull and ladder sections for highway transport. In all, it would take 23 truckloads to move the dredge from the manufacturing site to operating site in Florida. The hull would measure approximately 76m (250 ft) long x 15m (50 ft) wide, the 72m (236 ft) long ladder would extend the overall length to 91m (297 ft), making it roughly the size of an American football field. The gantry would reach a height of 15m (50 ft) above the waterline and the mass of the completed machine would displace approximately 454 t (1,000,000 lb) of water.

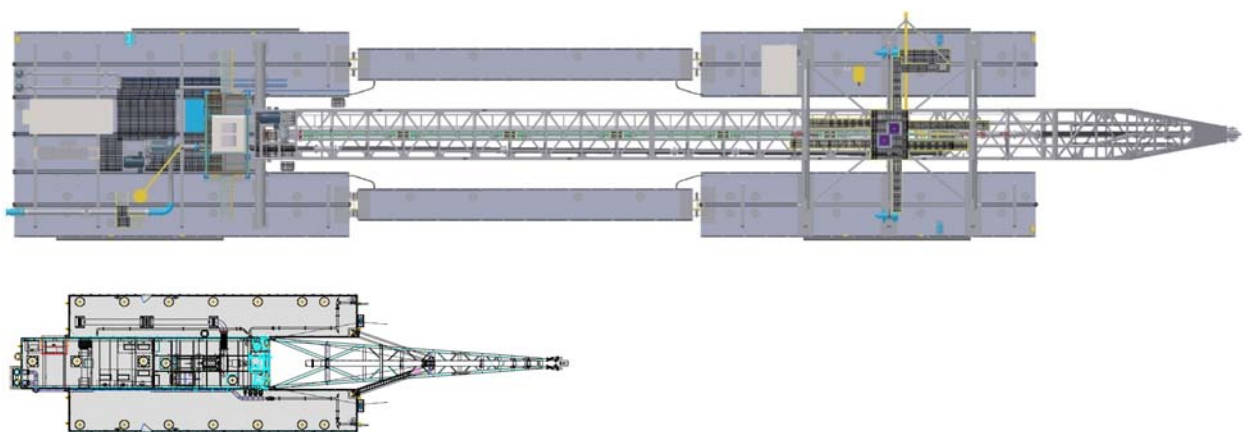
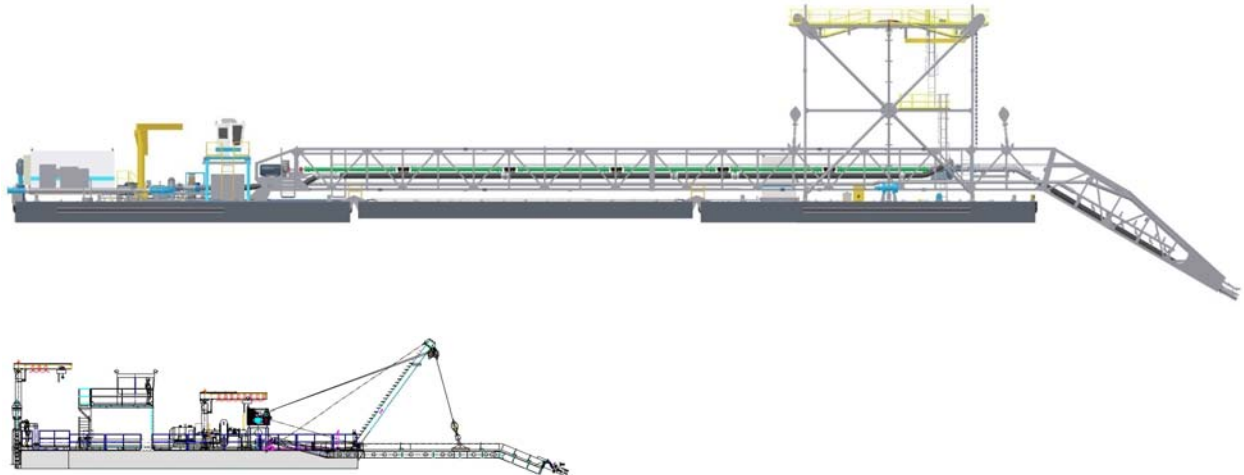


Figure 1. Plan view of Witherspoon dredge (top) and standard Vulcan dredge (bottom).



**Figure 2. Profile view of Witherspoon dredge (top) and standard Vulcan dredge (bottom).**

With the dredging being totally dependent on electrical power provided from shore side service, another consideration was to accommodate the hoisting of the ladder in the event of a power outage. This was accomplished by installing a 60 kw (80 hp) John Deere diesel engine that would drive a hydraulic pump to feed the ladder hoisting mechanism in the event of electrical power loss. This feature was a very important consideration because of the possibility of a cave in situation while the dredge was powerless.

Maintenance of onboard dredge components was also a major consideration throughout the design process. Simply based on the overall size of the dredge and the inability to economically dry-dock the dredge for maintenance repairs, the dredge needed to incorporate as many features as possible to allow for maintenance procedures to be performed while the dredge was floating. An example of the maintenance features includes a rail and trolley mounted inside the ladder structure to accommodate adjustments to the torque tube driveline or slurry piping. A stern jib crane was also supplied to handle lifting capabilities required for the stern deck equipment.

Another interesting concept used on the ladder was HDPE pipe spools instead of the standard steel pipe. This concept was used for a multitude of factors including; better wear characteristics, better adapted to the corrosive pond environment, weight considerations and the fact that the HDPE pipe is more maintenance friendly than steel pipe.

### **THE LADDER**

The ladder needed to be long enough to reach 61m (200 ft) at an angle of 60 degrees as shown in Figure 3. The ladder would support an underwater pump, the pump's drive system, slurry piping, jet water piping, etc. The ladder structure needed to be rigid enough to support itself and all of the components incorporated within the ladder structure. A major design consideration would be to accommodate the working safety of the driveline connecting the electric motor to the ladder mounted underwater pump. The ladder design would be required to minimize its normal operating deflection to reduce the amount of stress of this driveline. At the same time the ladder needed to be light enough to keep the equipment, structure, and power required to raise and lower it, at a reasonable level.



**Figure 3. Autodesk Inventor model showing 60 degree down angle.**

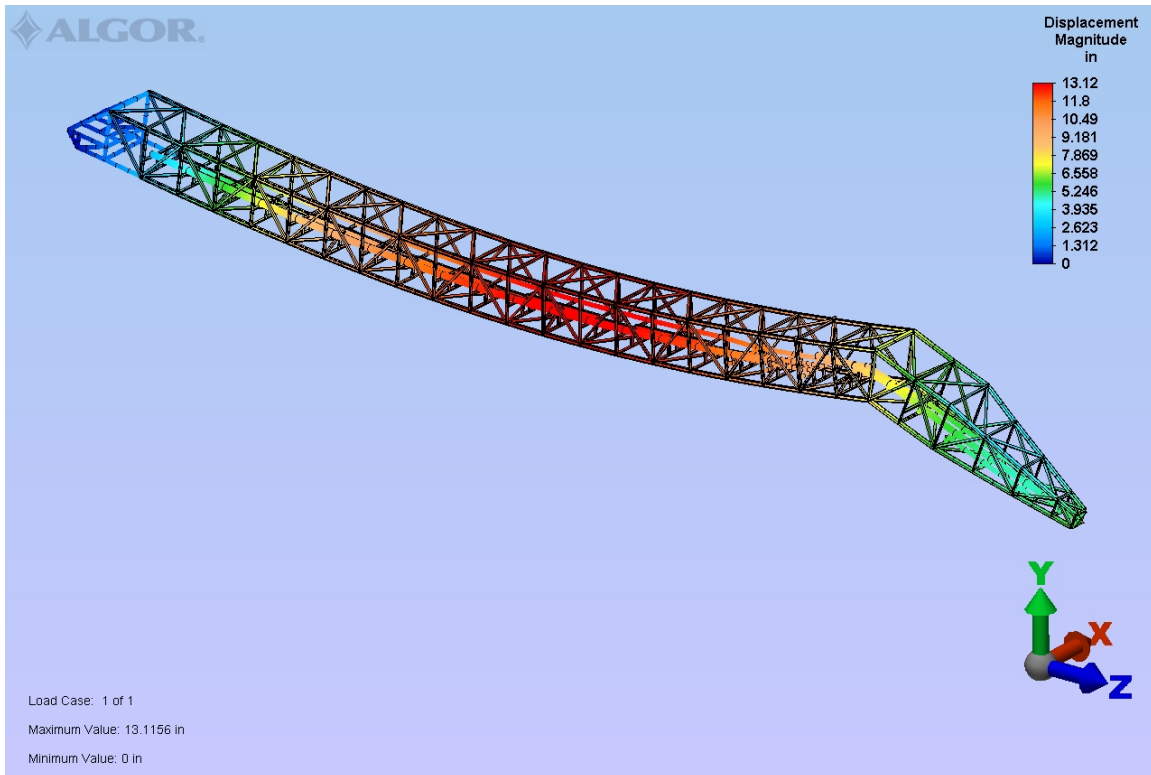
### **Ladder Structure Design**

The solution came in the form of a warren truss structure, constructed from steel beam and angle. While more labor intensive to fabricate, it offered the strength to weight ratio that was required for this application. While the initial design was based around a tubular steel structural member, this design concept was eventually abandoned due to the possibility of water intrusion into the tubular members causing corrosion from the internal sections to the exterior/outside. With an electric motor located on the ladder over the pivot and the pump located at 75% of the ladder's length, it would take six torque tubes and five sealed bearing assemblies to couple the motor and pump together. This would represent the longest in-series of drive shafts to date supplied by the manufacturer T.B. Woods. The driveline was located along the centerline of the truss structure to reduce the amount of axial stress it would be subject to when the ladder deflects in any direction. However, this does not eliminate the bending that the driveline would experience during deflection, so the coupling components would still need to be flexible. The torque tubes selected are a dynamically balanced hollow tube shaft, coupled to the bearing shafts with flexible stainless steel disc packs. These disc pack couplings allow for an angular misalignment of 0.5 degrees per disc pack. This capability applied over a long span through multiple coupled sections results in a driven component, which can be well off the original centerline. The mid point deflection of the drive train could approach 51 cm (20 in) off center and still be within the manufacturers' constraints. In this particular application, the allowable deflection of the driveline is greater than that of the ladder structure under extreme conditions. This flexibility allows the driveline to follow the ladder's natural deflections during operation without consequence.

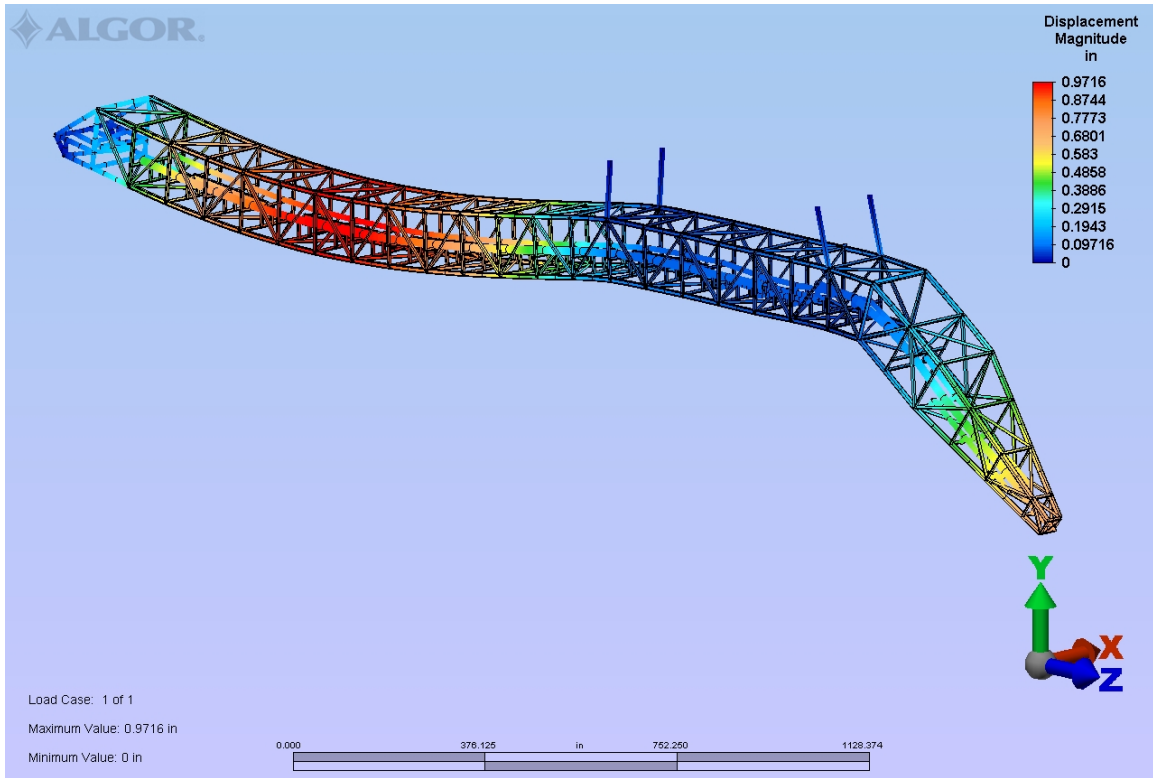
### **Ladder Hoisting Concept**

To further reduce the amount of operational deflection and stress in the ladder structure, the ladder needed to be supported and hoisted from two points rather than just the typical single point hoisting location. While the maximum deflection calculated and observed is around 30.5 cm (12 in) when the ladder is bridged from its pivot to the tip of its nose (Figure 4), which is necessary when the nose is set on the bank for maintenance and cable replacement, this deflection and subsequent stress is reduced to less than 2.5 cm

(1 in) when supported by two hoisting points located forward and aft of the underwater pump (Figure 5). Hoisting the ladder from two points over 15m (50 ft) apart presented its own challenge. The initial thought was to use two winches, each one lifting its own point. This had the disadvantage of controlling relative winch speeds and pressures as the forward hoisting point would move faster than the aft hoisting point as the ladder rotates around its pivot. This arrangement also had the disadvantage of elevating the stress levels in the ladder in the event that one winch failed or lost a cable and the other would have to lift the entire ladder from its point alone. It was decided to use one winch to lift the entire ladder weight through both hoisting points. This would not only ensure that the ladder would be fully supported but would allow a second winch to run in parallel without having to worry about differential speed controls. Both winches will be pulling or paying out the same amount of cable on the same layer at all times. The second winch adds a level of redundancy to the system by reducing the load on the winch and hydraulic system by half. In the event that one winch or cable fails, the other winch is capable of lifting the entire ladder from both hoisting points.



**Figure 4. ALGOR computer generated stress analysis. Ladder deflection bridged condition.**



**Figure 5. ALGOR computer generated stress analysis. Ladder deflection fully supported condition.**

### THE GANTRY

The winch and hoisting arrangement would lead to the way in which the gantry frame was designed. Not only would the gantry have to support the weight of the ladder, the maximum pull of the winches, and the forces imposed by the necessary reeving, but it would also have to do this while holding the hull tanks together when they naturally want to spread apart under the ladder's weight and the pull of the dredge's positioning winches. Other design considerations were that the frame would have to be tall enough to lift the entire ladder above the water line, which resulted in a taller than expected structure caused by a permanently angled nose to aid in "shallow" digging depths, and the hoisting winches were to be located on the deck. The hoisting winches were located on deck to aid in maintenance, cable changes, and safety.

The gantry would take a shape similar to a hanger, as shown in Figure 6, in order to support two loads 15m (50 ft) apart on the same cable. Each cable would originate from its winch located at the base of the center columns of the frame and pass through five directional sheaves on the frame and two sheaves on the ladder, one at each traveling block forming a four-part system. Each sheave would impose a load on the frame that was accounted for during the structural analysis. Unlike a hanger with a rigid foundation, the base of the columns would also be free to move in translation, resisted only by the stiffness of the hull and the frame itself. Upon completion of the gantry's design and computer stress analysis it was calculated that the forward hull tanks would spread 3.8 cm (1.5 in) apart with the gantry supporting the ladder completely removed from the water. The forward hull tank spreading condition and value were easily measured in the field and verified after commissioning the dredge and proven to be in line with design calculations. The stress modeling as detailed in Figure 7 was performed in a 3 dimensional finite element analysis software known as ALGOR.



Figure 6. Gantry.

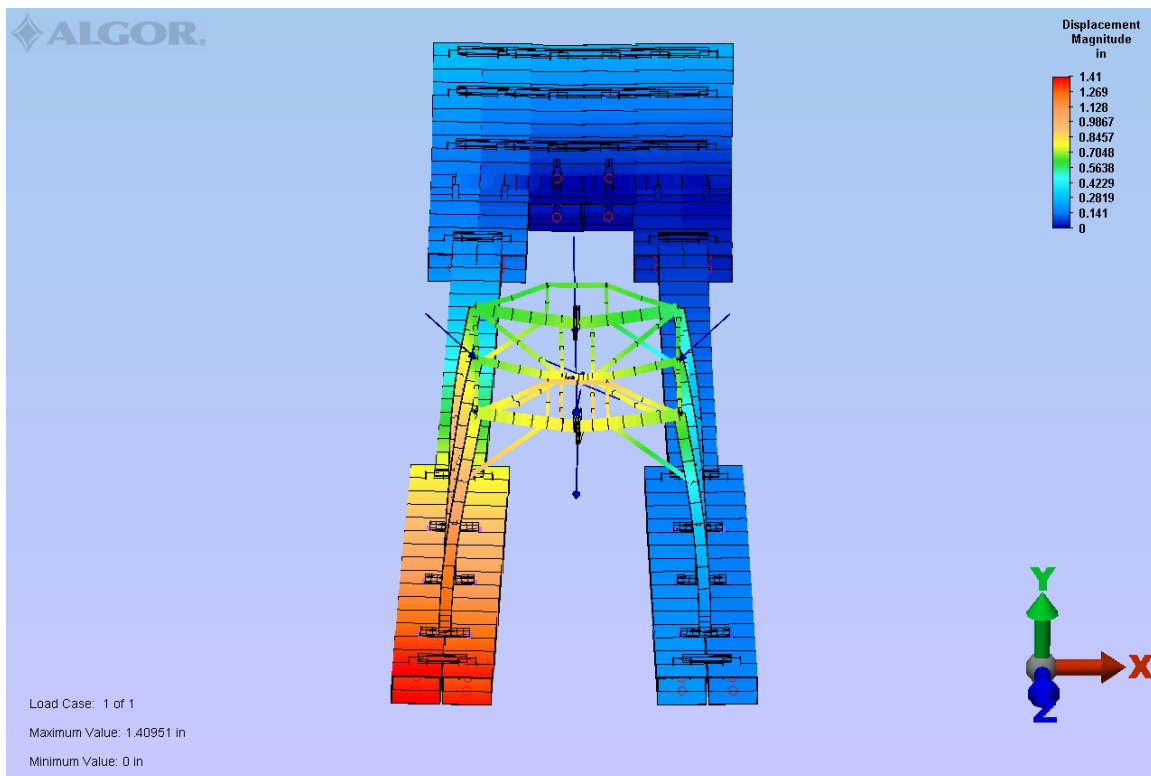


Figure 7. ALGOR computer generated stress analysis. Hull bow deflection – gantry fully loaded.

## THE HULL

With the ladder structure and its gantry designed and weighed, the design of the hull floating platform could progress. A typical hull arrangement for a mining dredge of this type digging half the depth would consist of four rectangular steel tank sections joined in pairs, forming a port and starboard hull similar to a catamaran, and supporting the ladder between them. For ease of fabrication and transportation, it was decided to construct the hull with several tank sections bolted together as had been done successfully in the past; however, instead of four tank sections it would take twelve (12) to provide the necessary displacement and stability required. From the start, it was known that the hull could not be one 'monolithic' rigid structure. The vast distance between the distribution of the ladder weight at the pivot and gantry would impose forces on each local section of the hull, causing it to want to trim in its own way. As the ladder is raised or lowered, the center of mass shifts a considerable amount further contributing to differences in local trimming moments. The magnitude and distance between these moments would result in failure for a rigid hull of the proposed cross section. Without any other need or want for a larger cross-section, the hull was divided into three sections; an aft section, mid section, and forward section. As depicted in figure 8, each hull section would be pinned to the next allowing for release of the moments between the aft and forward sections and allowing them to trim independent of each another.



**Figure 8. Photos showing hull pin sections that allow for proper trim conditions.**

### Hull Corrosion Protection

Other design considerations included longevity and corrosion protection. In order to attain a long life in a pond with a pH equivalent to tomato juice, the owner requested the entire hull to be protected with a system that would last the design life of 40 years with no maintenance. Coal tar epoxy systems that are typically used offer good protection so long as the coating is not damaged and the metal beneath it is not exposed to the elements. When damaged, the exposed metal begins to rust. The formation of iron oxide will expand beneath the coating causing it to spall or flake off, exposing more of the base metal. This unavoidable situation results in the need for periodic maintenance of the coating in order to preserve the integrity of the hull. Sacrificial zinc or aluminum anodes can be used with a traditional paint system to improve the corrosion protection; however they must also be maintained by replacing them periodically. Anodes must be installed beneath the water surface, which adds a level of difficulty for a machine that will never be pulled out of the water. A coating system that offers cathodic protection in the same manner as anodes comes from a process known as thermal arc spray. This process melts a protecting metal, such as aluminum, and atomizes it with a stream of compressed air depositing it on the base metal. The application process is shown in Figure 9. The aluminum becomes a part of the steel substrate and will protect it the same way galvanizing steel works. Unlike traditional coatings, if the aluminum barrier is damaged and the base metal is exposed, the coating will not flake off and expose more area. Furthermore, it was found that this coating used in aggressive marine environments could experience a life of up to 40 years (USACE 1999). This system was chosen to coat all sides of the tanks submerged in water.





**Figure 9. Thermal arc spray application.**

### **Hull Connections**

It was also requested that the hull have no bolted connections underwater as well as the hull sections not contact one another to prevent friction between tanks from rubbing their coatings away. This was achieved by leaving a 2.5 cm (1 in) gap between each tank and bolting the tanks to several W-beams that spanned the hull's width. The W-beams bolted to T-beams located over each tank's internal bulkheads. Connection tabs were fitted at the ends of each tank above the water line to prevent the tanks from moving forward and aft relative to one another. The connection beams were sized to resist the differential forces from concentrated tank loads and buoyancy. The results included a stable hull with no bolted parts beneath the water that could flex with the different ladder positions.

### **LADDER RECOVERY SYSTEM**

After protecting the hull the ladder needed to be protected from something more dynamic. Operational intent of the dredge is to sit in one spot and dig. The digging jet water suspends the material and the pump suction sucks it up. As the material is removed a cone begins to form with a natural repose angle. As more material is removed the cone gets bigger, the ladder lowers and the dredge digs deeper. Boring data reveals lenses of clay mixed in with the sand. This clay can prevent sand from forming its natural angle of repose thus causing instability in the underwater bank. The sand could eventually release under its own weight and depending on its volume; potentially bury the nose section of the ladder, plug off the suction or totally trap the dredge. Several ideas and designs were discussed for systems to prevent or alleviate this condition and make the ladder retrievable for when such a scenario presents itself. The prevailing idea was to use the existing jet water by positioning twelve (12) nozzles throughout the nose section of the ladder. Valves on the deck would divert jet water from the digging nozzles located at the tip of the nose to the recovery nozzles located within the nose as shown in Figure 10. At the same time valves behind the plugged suction inlet would open up two chutes attached to the suction line that a majority of the recovery nozzles would feed slurry into. The idea is that by saturating the ladder nose with enough high pressure water the material will become suspended and over time conveyed down the chutes into the suction line. This design was implemented and further developed to be initiated via automation from the dredges onboard PLC based operating system.



**Figure 10. Ladder recovery jets during testing.**

#### **SITE ERECTION AND LAUNCHING**

The site erection and launching procedures were very different from standard dredges because of the complexity of launching a 91m (300 ft) long machine via the “push in” method. The dredge would be erected from the stern sections then moving towards the bow of the dredge and adding in the ladder section through the process. As the sections were assembled they would be launched allowing for a seamless water based and land based erection process. The final push would require two (2) dozers and two (2) front-end loaders as shown in Figure 11. The re-coating of the hull sections with the aluminum process was required to allow for the long-term protection required for overall project success. It took approximately 8 weeks for the erection and launching process, which was affected by Christmas and New Year’s holidays.



**Figure 11. Final push during launching.**

## CONCLUSION-LESSONS LEARNED

At onset of this project being a one of a kind dredge and the buyer not really being interested in serial number one of any piece of equipment, there were many challenges in getting a hand drawn concept as show in Figure 12 to a commissioned vessel. While the cost estimating process was fairly straight forward for the structural items and major equipment, estimating man-hours to design and fabricate the dredge was a challenge. Simply based on the size of the dredge a legitimate cost center would be estimating the walking time it would take to board the vessel and get to the work area to perform daily tasks. Another factor was the level of involvement from Vulcan's engineering department and the effect it had on time to engineer and manufacture the dredge. While their involvement was welcomed and appreciated, it was a little more involved than what was originally expected from the onset of the project. Looking back, Vulcan's engineering involvement is considered a key component in the success of the project and the continued strong relationship between manufacturer and buyer. The biggest challenge in the entire project was the thermal arc spay application process. The application process is very intense, requires continual quality control checks and is very weather dependent on when you can apply the coating. This would be the one item that we would either decline from doing in the future or to make sure you have adequate time and money for this process. With the dredge being commissioned in March 2010 you must ask yourself if you would take on a project of this magnitude again. The simple answer is yes. The design would be almost identical with a reproduction of this same dredge with a few minor changes possibly with structural shapes and sizes for more efficient fabrication processes. The dredge is producing at well above the estimated production rate and on an average at approximately 50% of full capacity due to a processing plant limitation.

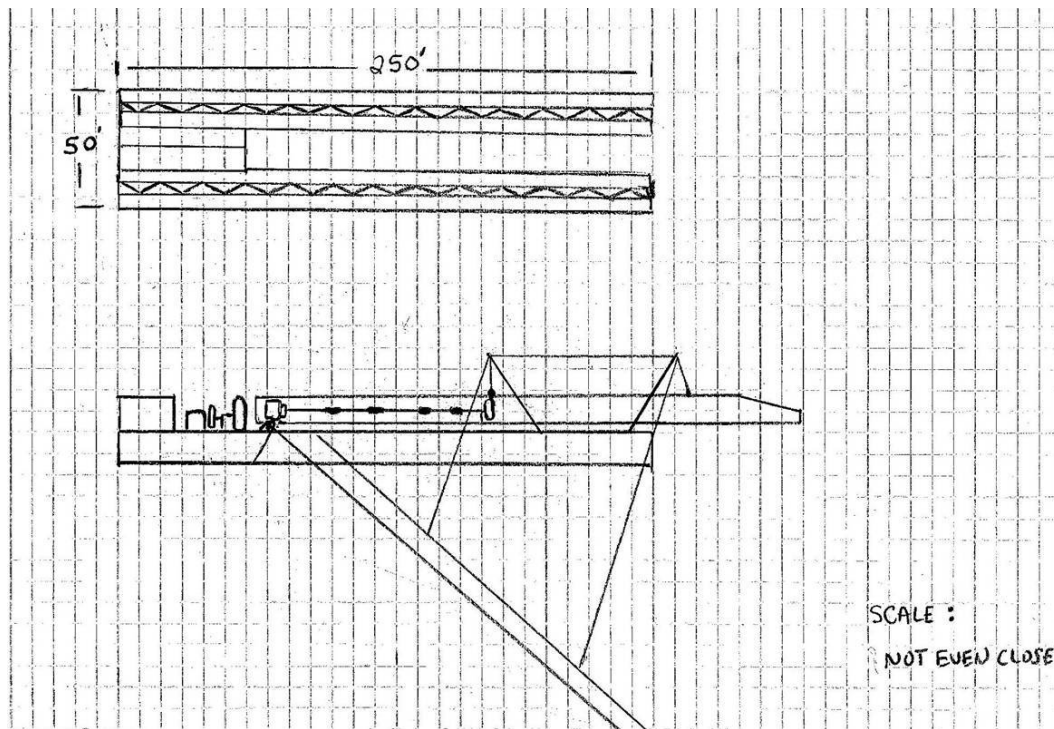


Figure 12. Starting sketch to estimate hull size and configuration.

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