Excavator boom design essay

UNIVERSITY OF THE WEST INDIES MENG 2005 – ENGINEERING DESIGN 1 LECTURER – DR. BRIDGE GROUP PROJECT – DESIGN OF A MINI – EXCAVATOR DATE SUBMITTED – 19TH NOVEMBER, 2010 GROUP NAME – THE A-TEAM GROUP MEMBERS ? ? ? ? GERARD MOHAMMED JUSTIN KALLOO SHIV RAJ BALROOP ADRIAN PIRTHYSINGH 809000675 809002400 808000136 809000847 i TABLE OF CONTENTS SECTION

 INTRODUCTION On any construction site, the excavator is one of the most important pieces of equipment. Excavators are designed to move considerable amounts of soil and earth. They consist of a backhoe and cab mounted on a pivot (a rotating platform) atop an undercarriage with tracks or wheels. Excavators are used in several tasks on construction areas, such as: brush cutting (with hydraulic attachments), digging of trenches, holes, foundations, demolition, general grading (landscaping), heavy lifting, mining and river dredging.

Excavators come in a wide range of sizes, spanning from large excavators weighing in at around 30 tonnes to mini excavators weighing around 5 tonnes. This project deals with the design of a mini excavator to be used on a small construction site. To increase the versatility of the machine, the excavator is designed with the capability of attaching different work tools. Our particular design contains four attachments, along with the standard bucket. These accessories include: 1. 2. 3. 4.

A trench/ditch cleaning bucket A hydraulic hammer An auger A compactor This report contains the steps taken in completing the design of a mini excavator. It includes final drawings as well as engineering calculations based on several design specifications to determine suitable dimensions for the finished design. In arriving at the final design, a few alternatives were considered and evaluated. The report also deals with the different alternatives and the methods of evaluation for choosing the final design.

The bulk of this report details the methods undertaken to solving the design problem, which entails an array of design calculations, all aiding in arriving at the final design. Appropriate justifications for choices made are stated throughout the report. Certain assumptions were also made during calculations and these were positioned next to the relevant calculations. 1 LITERATURE REVIEW In our research for the Mini-excavator there were many useful sources that helped in our design and understanding of the Mini-Excavator as they directly pertained to our project.

These sources gave a brief history of the excavator as well as how it operates and functions. There were also sources pertaining to Hydraulic systems and the different types of steels used to construct the excavator itself. The first source used was an excerpt from the book, “ Tall Buildings: From Engineering to Sustainability. ” This was used to attain a basic knowledge of the workings and the history of the hydraulic excavator. This was utilised in tandem with other sources to provide a coherent amount of background knowledge on the hydraulic excavator.

The design of the excavator goes way back to the year 1839 by inventor William Smith Otis, starting with the Otis Steam Shovel. This machine was powered by steam and for many years after, most inventions that were created used the same steam principle. As the age of steam powered excavators died off, new excavator inventions were created. These new excavators were powered by fuel such as diesel and the movements of the machine itself as well as the arms were run using a hydraulic system.

Prior to this, there were even excavators whose arms and its mechanism for digging were run by cables, running along pulley systems. Excavators have a wide variety of uses and can be customized to fit the desired job. The customization of an excavator involves changing of the bucket or adding hydraulic attachments. The standard bucket can be changed with various sizes of different types of buckets (ditch/trench cleaning) and other attachments include augers, hammers, vibratory compactors, dozers, breakers, grapples, thumbs, clamshells, wood clippers, brush cutters and concrete crushers.

Excavators are used in digging of trenches, holes for foundations beams, general construction or even demolition of old buildings, sewage works, road widening works, pipe laying and rock breaking and dragging; resources mining which include overburden, removal in coal mining, gravel collection; forestry works which entails timber handling and prune off work; metal recycling which involves slag handling and scrap handling; brush cutting with hydraulic attachments, general grading and landscaping. To assist in understanding the basis of operation of the excavator, an article on hydraulics was used.

This article gave a general understanding of the principles governing the transmission of pressure due to hydraulics and its widespread application. The basic idea behind any hydraulic system is very simple: Force that is applied at one point is transmitted to another point using an incompressible fluid. The fluid is almost always an oil of some sort. The force is almost always multiplied in the process. 2 The arm of the excavator is attached to the lower part of the frame chassis. This arm has three hydraulic pistons with chromed steel piston arms to reduce corrosion. The arm has two main sections and a bucket loader.

The two main sections are jointed with a hinge. One piston is attached to the underneath side of the first section and one on the top side of the second section. When the first piston extends, the rod pushes against the arm and raises it, extending the section. The second arm contracts or expands, raising and lowering the second section for more reach. An additional hydraulic piston moves the bucket loader forward and backward so the bucket can dig or scoop up material. Articles on interaction with soil were also used to obtain some background information on the material that the excavator would be designed to work with.

Soil type is extremely important when considering the type of excavator to be selected for design for an application. When choosing an excavator, soil density, drainage and ability to withstand differing weights must be on the criteria list determining the selection of the excavator. Fine sands are found in the coastal areas where the use of the mini excavator is limited. Clay soil particles are the smallest of all soil particles being approximately 0. 002mm in size. The bulk density of the soil is defined as the mass per unit volume including the pore space of the soil.

Since clay soils have the smallest particles they can be held together most compact and a high bulk density is directly related to being more compact. Further compaction of the clay soil can be done however with use of the plate compactor attachment to the excavator. Soil Porosity refers to the amount of pore or open space between particles. Clay soils hold the least amount of water as they are the least porous in nature. If a highly porous soil was considered for being excavated like when digging, holes will have to be present at the sides of the bucket to make ure that excess water is removed from the material and not adding to the weight to be carried therefore decreasing stability. Tracks are ideally suited for use when working in areas when the material is clay soil as it has a high stability and manoeuvrability in this type of condition. For the purpose of this design, the working material is assumed to be a loamy soil of density 1360kg/m3. This influenced our choice of maximum digging force which was chosen to be 25kN. The excavator final design included rubber tracks since the excavator is designed to work at small construction sites and around the home.

Rubber tracks will decrease the amount of damage done to the ground and hence would be much more suitable than steel tracks. A fixed boom was used for simplicity in the design and also for the advantage of producing a much larger digging force than a swing boom. To aid in our design, a few patents and previous mini-excavator designs were looked at. The first patent looked at was from the United States Patent Publication, Patent number US 2002/0174573, published on November 28th, 2002. This patent was a design of a mini excavator, which made it relevant to my design.

However, the main aim of their design was to solve the problem of the hydraulic hoses obscuring the view of the operator. The patent was helpful in that 3 it provided some information on the design of the hydraulic system. My design differed in that my aim was to design a mini-excavator to be used on a small construction site. Therefore, when compared to the referred patent, the excavator I designed had a smaller total weight. This was due to the fact that the excavator would be used around the home and on small construction sites and therefore the weight was minimized to reduce damage to the surface on which the excavator is operating.

Another patent was used for this literature review to aid in determining the final design of the excavator. The patent used was a United States Patent, Patent number Des. 406152, dated February 23rd 1999. This patent is an ornamental design of an excavator. This patent was different to my final design in that their design had greater manoeuvrability of the boom and stick. The boom was designed in two parts, allowing the second part to rotate with respect to the first part. This was also the case for the stick, in that it could rotate in two planes with respect to the boom. My design however, was more conventional in hat the stick could only move with respect to the boom in one plane. Although the design put forward in this patent can prove to be a great innovation, it significantly reduces the magnitude of both the applied digging force and the payload capacity. Also, my design is much simply to manufacture and will contain less points of stress concentration. This is due to each connecting point in the design from the patent providing a point of stress concentration and making the overall machine weaker. Thus, my design would be stronger overall and can also be said to be more durable.

The sources used for this literature provided sufficient background information to allow for accurate completion of the project. Patents were used for comparison so that there would be certainty that my design is actually an improvement to existing designs of its type in terms of performance and customer satisfaction. 4 DEFINITION OF PROBLEM Large excavators are generally needed for heavy duty construction projects, but these excavators have proven to be too cumbersome for smaller size projects. Mini multipurpose excavators have proven to be very practical when it comes to „ do it yourself? (D. I. Y. and home use projects as it still has the same capabilities as the larger excavators such as shallow digging, breaking concrete, digging deep narrow holes, cleaning ditches and compacting the earth and yet small enough to manage easily in small spaces. Objectives ? ? ? ? ? To design a Mini-Multipurpose excavator for the use on small light duty projects. The payload capacity = 0. 5 tonnes Diesel powered Standard bucket size = 0. 3m3 Attachments : A trench/ditch cleaning bucket An Auger for digging deep narrow holes A Hydraulic hammer (for breaking concrete) A Compactor (for compacting the earth) DESIGN SPECIFICATIONS 1. Since excavator will not be needed to make large movements across ground, the top speed is limited to 5km/h. 2. Excavator will be working with soil hence a max digging force of 25kN. 3. The standard bucket size is 0. 3m3 4. Diesel engine chosen for power. 5. Excavator total weight limited to 5. 5 tonnes. 6. The payload capacity of the excavator is limited to 0. 5 tonnes. 6 FUNCTIONAL DECOMPOSITION DIAGRAM INPUT Accept chemical energy from fuel Convert to mechanical energy for hydraulic pump OUTPUT Motor for swivel

Excavator overall motion Manipulation of arm and bucket/attachments SOIL Scoop and Transport Soil DISPLACED SOIL SIGNAL Control Setup VISUAL SIGNAL OF OPERATION Figure 1 7 MORPHOLOGICAL TABLES Table 1: Function Motion Movement of arm Dumping and scooping Swivel A Tracks Hydraulics Hydraulics Rack and Pinion Method B Wheels Cables Cables Belt drives C Hover with air Pneumatics Mechanically Direct Hydraulics Table 2: Function Motion Movement of arm Dumping and scooping Swivel Combination 1 B C C B Combination2 C B B C Combination3 A A A A EVALUATION OF ALTERNATIVES

Table 3: Evaluation Weight (%) Cost 20 Ease of 15 Manufacture Durability 20 Maintenance 12 Ease of operation 8 Power output and 10 Transmission Ease of changing 10 attachments Performance in 5 variety of terrain/environment TOTAL 100 Alternative 1 0 0 0 0 0 0 0 0 Alternative 2 ? ? ? 0 + 0 ? ? Alternative 3 ? ? + + + + + + 0 -62 +30 8 From the above evaluation, combination 3 was chosen as the final design, as it greatly outweighed the other two possible choices. Combination 1 consisted of using wheels for

motion, pneumatics for motion of the arm, mechanical means for dumping and scooping and belt drives for swivelling.

This combination was used as the datum and the other combinations were compared with reference to this one. Combination 2 consisted of motion by hovering with air, using cables for both manipulation of the arm and for dumping and scooping, and swivelling by using hydraulics directly. When compared to combination 1, this combination was seen to be inferior in that its evaluation produced a total of -62. Combination 3 consisted of motion using tracks, using hydraulics for both manipulation of the arm and for dumping and scooping. The rack and pinion method was used for swivelling.

The evaluation of this combination with respect to combination 1 produced a total of +30. The various alternatives were evaluated based on eight characteristics as seen in the table. The characteristics were weighted by percentage based on their importance to the overall design. 9 FINAL DESIGN The final design therefore consisted of tracks for motion, hydraulics for motion of the arm as well as dumping and scooping, and the rack and pinion method for swivelling. It was also decided that a fixed boom would be used rather than a swing boom.

Although a swing boom presents obvious advantages, the fixed boom was chosen to allow for a greater digging force to be obtained. The material chosen for the structural components was AISI 4340 steel. This material was chosen mainly because of its high tensile strength 830 MN/m2 and its resistance to corrosion. This type of steel is alloyed with nickel, chromium and molybdenum making it extremely corrosion resistant in a variety of environments. Also, this material has good forming and machining characteristics, making it ideal for structural parts.

This material was therefore used for the boom, stick, bucket and the pins at the connecting joints. The hydraulic pistons were made using AISI 6150 steel which has a yield strength of 745MN/m2. As in any design, an appropriate factor of safety must be chosen. The factor of safety is used to account for effects of temperature, uncertainties in the material and also uncertainties in the loading of the component. The overall factor of safety is given by: n = nS ? n L Where nS represents the strength based factor of safety. This was chosen as 2 because ? Representative material test data was readily available for all materials used in construction of components in the design. The components would be used in moderately challenging environments. nL represents the loading based factor of safety and was assigned a value of 2. Therefore the overall factor of safety used was 4. 10 PART LISTING: Table 4 Part Number 1 2 3 4 5 6 7 8 9 10 11 12 13 14 Part Description Rubber tracks Base Engine Hydraulic pump Hydraulic motor Swivel Gears Boom Stick Bucket Connecting pins Hydraulic pistons Hydraulic cylinders Cabin Electronic control setup

Quantity 2 1 1 1 2 1 1 1 1 9 3 3 1 1 11 PROJECT PLANNING Table 5 Weeks Task Problem Definition Conceptual Design Engineering Calculations Solid works drawings Final Analysis Discussion of findings 1 2 Done by: Entire Group Entire Group Gerard Mohammed, Justin Kalloo Gerard Mohammed Gerard Mohammed Gerard Mohammed * * 3 4 5 6 7 8 * – – – * – * * * * * * * * * * * Compactor Buckets Jackhammer Auger Model Justin Kalloo Adrian

Pirthysingh Gerard Mohammed Shiv Raj Balroop Entire Group – * – * * * Key: * Initial designated time frame Actual completion timeframe 12

FAILURE MODES, EFFECTS AND ANALYSIS This was done on each major component of the excavator. 1. Hydraulic Setup Failure mode Failure cause Effects Safeguards Actions Seizure Improper Lubrication Piston sticks and pump would not work causing a total shut down Corrosion of metal fittings Exposure to Moisture It can cause a leakage Design hydraulic setup such that lubrication is made simple and does not need a significant halt in work progress. Use more corrosion resistance materials Use a good lubricant (oil) Ensure enough lubricant is used within reasonable ime intervals Replacement of the rings. Have regular check up to ensure everything is in order Replacement of piston rings Wearing of pistons rings Constant in an out movement Reduction in pump efficiency due to oil leakage and reduction in output pressure Use Stronger materials 13 2. Tracks Failure mode Crack Failure cause Environment in which equipment is working in Effects Equipment would not be able to move properly Equipment would not be able to move properly Safeguards Use a very strong material which can withstand shocks, impacts etc Install a lubrication system.

Use stronger materials in making gears Install a lubrication system Actions Replacement of tracks Wearing of gears Constant turning the tracks movement cause gears to wear Replacement of gears Seizure of tracks Not lubricating the gears Equipment would not be able to move Lubricate gears within reasonable time intervals 3. Cabin Failure Modes Surge Failure Cause Faulty alternator overcharging Effects Injury to operator due to electrical shock and also loss of control of the machine. Safeguards / Backups Contacts made with non corrosive materials.

Wires are made off material that allows it to be flexible. Also, use of PCB? s for hydraulic controls. Actions 14 4. Engine Failure Modes Corrosion Failure Cause Exposure to water flowing through water lines in engine block Effects Damage to water lines in engine block. Also, engine may overheat, malfunction, an eventually fail Safeguards / Backups Actions Use of anticorrosion engine coolant in the radiator Fracture Overheating Design of smoothed shaped water lines in block (i. e. no sharp change in flow direction, and therefore no rate of accumulation).

Also, the use of some corrosion resistant metal in casting of engine block Hair line cracks Introduction of an may develop on onboard engine head due to instrument panel stress with a temperature concentrations. gauge so that the This causes the temperature can engine to lose be monitored. power and eventually fail and overheat again Engine may not start due to lack of current to the starter and/or diesel pump. Electrical hydraulics controls may also malfunction and light may also fail. Starter can? t turn flywheel (flywheel stuck) so therefore engine won? t start Electrical short

Corrosion at electrical contacts, broken or burnt electrical wires, or even a faulty battery To prevent fracture (due t overheating), use engine coolant, and check the level of coolant regularly. Also, check the oil level to make sure it is adequate. Contacts made When electrical with non corrosive fault indicator materials. Wires (battery light) are made off comes on, stop material that usage and get allows it to be electrical system flexible. Also, use checked of PCB? s for hydraulic controls Have onboard water and oil check lights to indicate low level of fluid. Before starting engine, always check oil and water levels.

Seizure Corrosion, oil starvation 15 Wear Happens over time with usage. Also, may be cause by lack of oil. Leaking Damaged or faulty seals, gaskets and/or hoses. Loose fitting Vibration, wear Surge Faulty alternator, battery or fuses Oil starvation Oil leakage, bad seals, gaskets, hoses. Wear of rings may lead to engine smoking, back pressuring and eventually failure. Wear on cylinder may cause ridges and this may damage rings. Could lead to lack of oil, which implies loss in performance, overheating and eventually engine failure. Lack of water could lead to overheating, cracking nd then engine failure Can leak fluids, exhaust gas, lose compression. Can overheat and/or starve for oil due to lack of fluid. Could damage electrical controls, and render it and therefore boom and bucket operation, useless. Could damage other electrical systems, like lights, air conditioning, etc, Engine may overheat, begin to lose power and eventually fail. Main bearings may burn and seize. Make replacement parts readily available. Have onboard oil check light to monitor level of oil in engine. Check oil regularly, and change oil when required. To fix wear, the engine would need to be overhauled, or rebuilt.

Check oil and water levels before starting engine, and monitor if fluid is being lost from the fluid reservoirs. Use of appropriate, high quality materials for manufacture of hoses, seals and gaskets. Also, having the onboard oil and water temperature and level check lights. Implement vibration dampers on the engine in the design. Proper sizing and usage of fuses, and voltage, current regulators and filters. Equip cabin with an onboard electrical check light. Observe onboard electrical light and replace any faulty parts. Equip with an onboard oil check light in a very visible location

Check oil levels regularly and change oil when required. If oil starvation occurs, engine may need to be rebuilt or overhauled 16 5. Arm (Boom and Stick) Failure Mode Failure Cause Mechanical Component Crack or fracture exposed to excessive stress (compressive or tensile). Component exposed to impact loading exceeding maximum force it can withstand. Corrode The component may rust or corrode upon exposure to rain or corrosive chemicals Effects Causes an immediate halt in work progress. May cause load to fall to ground, damaging other equipment as well as injuring personnel.

Actions Carry out check after selected time periods to ensure no cracking or yielding develops. Do not subject component to loads beyond its rated load. Causes a Consider working Do not weakening of the environment when operate component, choosing a material machine in making it more for component. Also environment vulnerable to carry out proper for which it mechanical treatment to make was not failure. component more designed. corrosion resistant. Periodically ensure paint is fully applied for protection. Causes a Consider a suitable Carry out complete material and diameter routine checks dissembling of for bolts and rivets to ensure no arm.

May cause based on force cracks workload falling analysis. Reinforce develop at to the ground joints to account for joints or bolts. resulting in stress concentrating Ensure joints injury to effect. are well personnel. greased at all Damage to joints times. may transfer to damage to hydraulic system and hoses. Safeguards/Backups Ensure thorough force analysis is done to ensure suitable material as well as dimensions are chosen. Include appropriate F. O. S. Include a range of loads to be used in operation. Failure of Connecting Joints Cracks developed in bolts or rivets at joints.

Also, due to stress concentration at joints. 17 6. Electronic Control System Failure Mode Damage to connecting wires Failure Cause General wear of wires or being subjected to excessive current. Effects Causes operator to have no control of machine. Can result in fire or electrical shock, injuring operator. This can cause a total shutdown of the excavator due to the fact that operator cannot perform any processes. Safeguards/Backups Ensure a means of disconnecting or shutting off power in case of malfunction. Design electrical system so that wires and other components are easy to replace.

Contacts made with non corrosive materials. Wires are made off material that allows it to be flexible. Also, use of PCB? s for hydraulic controls. Actions Carry out periodic inspections of battery and wires to ensure no deterioration is occurring. Electrical short Faulty alternator, fuses, contacts, short circuiting, corrosion Monitor the electrical fault indicator, and get it checked by an electrical when it comes on. 7. Swivel Mechanism Failure mode Wear of Gear teeth Failure cause Direct contact of pinion and internally meshing gear from rotating cabin can cause the gear teeth to gradually wear.

Hazards Can cause user to lose control of the rotation of the cabin. Can cause over swing where arm can swing out of control and can damage buildings, nearby equipment and injure individuals Not lubricated Can cause an Overheating properly with instant halt in between oil and rotation of the bearings and grease. cabin or eventual sticking. failure. Effects Can cause the swivel movement of the cabin to stick or even halt when rotating Safeguards Use stronger material which is more resistant to fatigue and creep. Use material with high hardness As this would be better against deformation.

Ensure proper lubrication of bearings. Actions Have routine maintenance checks pertaining to the gear system. Make sure to grease gears after certain hours of use. Bearings Fail Have routine preventative maintenance checks dealing with lubrication of parts. 18 CALCULATIONS The following were required: ? ? ? ? ? ? ? ? Cross-sectional dimensions of boom and stick Diameter of pins at connecting joints A, B and C Diameter of hydraulic pistons Inspection of pistons for buckling Inspection of entire machine for tipping Pressure required from hydraulic pump Power capacity of engine Angular velocity of gear driving tracks . To determine cross-sectional dimensions of boom and stick. Figure 2: Sketch of Force analysis for Digging 19 Dimensions chosen: BC (vertical) = 1100mm AB (direct) = 2600mm AA? (direct) = 1500mm A? B (direct) = 1500mm Angle of curve on boom = 125° Width at curve = 400mm Width at A and B = 150mm AB (vertical) = 1500cos35° = 1228. 7mm AB (horizontal) = v (26002-1228. 72) = 2291. 35mm CF (direct) = 914mm CF (vertical) = 643mm CF (horizontal) = 650mm 20 Using equations of equilibrium: (Fx, Fy, Moments about a point) On boom, Ax + Bx – 0. 5 FH1 + FH2 = 0 ——— (1) Ay + By + 0. 66 FH1 = 0 ———- (2) -1228. 7 Bx – 2291. 35 By -117. 6FH1 – 1433. 5FH2 = 0 ——- (3) (Moments about A) On stick, -Bx + Cx – FH2 = 0 ——— (4) -By + Cy + FH3 = 0 ———- (5) 1100Cx + 200FH2 – 300FH3 = 0 ———- (6) (Moments about B) On

Bucket, -Cx = 0 ————- (7) -Cy – FH3 = -25kN ———- (8) 150FH3 = 16250kNm ——- (9) (Moments about C) These equations were used to formulate the following matrix equation: 1 0 0 0 0 0 0 0 0 1 0 0 0 0 0 0 0 1 0 -1228. 7 -1 0 0 0 0 0 0 1 -2291. 35 0 -1 0 0 0 0 0 0 1 0 1100 -1 0 0 0 0 0 0 1 0 0 -1 0 -0. 5 0. 866 -117. 6 0 0 0 0 0 0 1 0 -1433. -1 0 200 0 0 0 0 0 0 0 1 -300 0 -1 150 Ax Ay Bx By Cx Cy FH1 FH2 FH3 0 0 0 0 0 0 0 -25kN 16250kNm ? = 21 Solving the matrix yielded: Force Ax Ay Bx By Cx Cy FH1 FH2 FH3 Value (kN) -385. 05 641. 9 -162. 5 25 0 -83. 33 -770. 1 -162. 5 108. 33 NOTE: A similar force analysis was also done for lifting and the forces obtained were significantly less than those obtained for digging. This would translate into smaller stress values and hence would be irrelevant in these calculations since the stresses developed during digging would be used as the determining factor when deciding on the dimensions of the cross sections, pins, and hydraulic pistons.

Forces obtained for lifting: Force Ax Ay Bx By Cx Cy FH1 FH2 FH3 Value (kN) -0. 0000275 -15. 046 -11. 25 3. 875. 84 0 13. 34 19. 05 13. 73 9. 34 22 Boom Calculations: The cross section chosen for both the boom and the stick was a hollow rectangular cross section: Figure 3: Sketch for Boom Calculations Thickness , t chosen to be 15mm 23 Resultant of Ax and Ay = 748. 53kN at 59. 04° to horizontal Boom at angle of 55° to horizontal Resolving Force at A to components along and perpendicular to the member: Along the member: 748. 53 cos(59. 04 – 55) = 746. 67kN Perpendicular to the member: 748. 53 sin(59. 04 – 55) = 52. kN Maximum bending moment occurs at the curve which is called point A? = 52. 7kN ? 1. 5m = 79. 05kNm Maximum bending stress = My/I y = 0. 4/2 = 0. 2m I, using hollow rectangular cross section, =

bd3/12 (outer) – bd3/12 (inner) b = 0. 25m d = 0. 4m Using choice thickness, t of 0. 015m, I = 0. 25 ? 0. 43 /12 – (0. 25 – 2t)(0. 4- 2t)3/12 = 4. 074 ? 10-4 m4 Therefore Bending Stress = 39. 0MN/m2 Considering Axial Stress at point of interest, Stress = Force Along Member/Area of Cross Section Area of cross section for hollow rectangle = Outer Area – Inner Area Cross sectional Area = (0. 25)(0. 4) – (0. 25 – 2t)(0. – 2t) = 0. 0186m2 24 Axial Stress = 746. 67 ? 103N/ 0. 0186m2 = 40. 14MN/m2 Therefore total stress at point under consideration = Axial Stress + Bending Stress = 79. 14 MN/m2 NOTE: Stress analysis taken from left hand side yielded the same result so this value was used. Material chosen for boom and stick – AISI 4340 Steel of yield Strength 830 MN/m2 Factor of Safety = nS ? nL = 2? 2= 4 Therefore design stress, = Yield Stress/ F. O. S = 207. 5 MN/m2 Hence it can be clearly seen that the cross sectional dimensions chosen for the boom are safe, since the design stress exceeds the maximum stress in the component. 5 Stick Calculations: Figure 4: Sketch for stick calculations For the stick, a hollow rectangular cross section of thickness, t = 0. 015m was also used. However, the width, b was chosen as 0. 2m. Forces at joints B and C and hydraulic forces are already directed perpendicular to and along the member. Maximum bending stress occurs midway between B and point of application of FH2. This point under consideration has width b, 0. 2m, d = 0. 15m hence y = 0. 075m. I is calculated using the same method as in the previous section, = 3. 175 ? 10-5m4 Max. Bending stress = (162. 5 ? 103N)(0. 1m)(0. 75m) / 3. 175 ? 10-5m4 = 38. 386MN/m2 26 Axial Stress at this particular point = 0. Therefore we consider point at which Axial stress is a maximum, and compare the magnitudes of the axial and bending stresses, choosing the higher value is the determining value for our design. Cross sectional area is determined using the same method as in the previous section. Cross Sectional Area = (0. 15)(0. 2) – (0. 15 – 2t)(0. 2 – 2t) = 0. 006 m2 Maximum Axial Stress = 83. 33 ? 103N / 0. 006m2 Therefore, the maximum bending stress is used in determining whether the component is safe. Since maximum stress = 38. 86MN/m2 is less than the design stress of 207. 5 MN/m2 the cross sectional dimensions of the stick are safe. 2. Determination of Pin diameters at Joints A, B and C Design Stress = 207. 5 MN/m2 Therefore Design stress in shear = 207. 5 MN/m2 /2 = 103. 75MN/m2 Resultant force at joint A = 748. 53kN Since Pins are considered to be in double shear, Shearing force is 748. 53kN / 2 = 374. 265kN Shearing Force / Cross Sectional Area of Pin, must not exceed Design stress in shear. Hence Critical value of cross sectional area = Force/? design Acritical = 374. 265 ? 103 N / 103. 75 ? 106 Nm-2 Acritical = 0. 036m2 Critical Diameter = 0. 0677m = 67. 7mm Chosen diameter = 70mm to the nearest centimeter. Resultant Force at joint B = 164. 4kN Using the same method as for obtaining pin diameter at joint A, the pin diameter at joint B was found to be 40mm. Resultant Force at joint C = 83. 3kN Using the same method as above, the pin diameter at joint C was found to be 30mm. 27 3. Determination of diameter of hydraulic pistons The maximum hydraulic force on any piston was found to be 770kN. Material of choice was AISI 6150 steel of yield stress 745MN/m2. Using same factor of safety of 4, Design stress = 186. 25MN/m2.

Critical Cross Sectional Area = Force/Design Stress = 770 ? 103 N/ 186. 25MNm-2 Critical Cross Sectional Area = 0. 00413m2 Critical Diameter = 0. 073m = 73mm Chosen diameter = 80mm to nearest centimetre. 4. Inspection of hydraulic pistons for buckling Material used to make hydraulic pistons – AISI 6150 steel which has a Modulus of Elasticity, E = 200 ? 109 N/m2. Using Buckling Equation for long slender columns: Pcr = ? 2EI / Le2 Where I = ? d4 / 64 (Using d as 0. 08m as determined in previous calculation) = 2. 01 ? 10-6m4 Le = Effective length = L = 0. 85m (for both ends pinned) Therefore Pcr = (?)(200 ? 109 N/m2)(2. 01 ? 10-6m4) / (0. 85m)2 = 5. 46 ? 106 N Applying Factor of safety to this load, Pcr = 1. 365 ? 106 N Largest Compressive Force is any hydraulic piston is 770 ? 103 N. Therefore, pistons will not buckle. 28 5. Tipping Figure 5: Sketch for Tipping Calculations To ensure that excavator does not tip, reaction must always be concentrated at some point which lies within the span of the base (tracks). Total unloaded Excavator weight = 5. 5tonnes, inclusive of counter weight. For a bucket size of 0. 3m3, and using soil of density 1360kg/m3, maximum load carried by excavator is 408kg = 0. 08 tonnes. Force due to excavator weight = mg = 53955N Force due to load, = mg = 4002. 48N (use 4500N for extra safety) Total normal reaction, R = 53955N + 4500N = 58455N 29 Considering worst case scenario, when arm is fully outstretched and bucket loaded to full capacity, taking Moments about the centre of gravity: Distance of load to centre of gravity: 1030 + 860. 36 + 1500 + 1033. 66 – 360mm = 4064mm 4500N ? 4. 064m ? 53955N ? d = 0 Solving for d, which is distance of normal reaction from centre of gravity, d = 0. 339m Now, total span of tracks is 2. 03m.

If centre of gravity lies at 1m from the right end of tracks, then distance d must be less than 2. 03 – 1m = 1. 03m. Since d is 0. 339m at maximum load at the furthest reach, which is less than 1. 03m, excavator will not topple. 6. Pressure requirement from hydraulic pump Largest force required by piston is 770kN. Area of application is taken as twice that of the actual piston area. Hence area of application is = 2 ? ? d2/4 = 0. 01m2 Therefore pressure requirement is given by Force/Area = 77MN/m2. To Account for efficiency of hydraulic pump, and pressure losses in hoses, Pump capacity is chosen to be 1. ? 77MN/m2 = 93MN/ m2. (for piston diameter of 80mm) 7. The Power of the engine was chosen to be 40kW, in order to drive the hydraulic pump and all other functions of the excavator. This power was chosen by using the equation Power = Force ? Velocity The maximum hydraulic force required was 770kN. To allow the piston to move at a velocity of 0. 05m/s, a power of 38. 5W would be required. To account for the efficiency of the engine not being 100%, an engine of capacity 40kW was chosen. 30 8. The maximum speed of the excavator along ground was chosen to be 5km/hr.

Therefore, to attain this speed, angular velocity of the gear driving the tracks was determined using the equation: V = ? ? radius Where V = speed in metres/second = 1. 389m/s And radius of gear = 0. 275m Hence, ? = 5. 05 rad/s 31 SUMMARY OF CALCULATIONS. Table 6 1. Cross Sectional Dimensions of Boom Hollow rectangle, of width 250mm and thickness 15mm. Depth at critical point is 400mm. Hollow rectangle of width 200mm and thickness 15mm. Depth at critical point is 150mm. 70mm 40mm 30mm 80mm 1. 365? 106 N 339mm 93MN/ m2 40kW 5. 05rad/s 2. Cross Sectional Dimensions of Stick 3.

Diameter of Connecting pin at Joint A (connecting boom to main body) 4. Diameter of Connecting pin at joint B (connecting stick to boom) 5. Diameter of Connecting pin at joint C (connecting tool to stick) 6. Diameter of Hydraulic pistons 7. Critical buckling load of pistons 8. Distance of Equivalent concentrated normal reaction from Centre of Gravity 9. Hydraulic Pump Capacity 10. Engine Capacity 11. Angular Velocity of gear driving tracks 32 DISCUSSION AND INTERPRETATION OF FINDINGS As outlined in the previous section, there were eight major calculations to be carried out.

Firstly a force analysis was done on a component level to obtain the maximum stresses in any component at the critical point. This was done for the chosen dimensions and compared to the design stress values to ensure that at no point was the actual stress exceeding the design stress. From this analysis it was concluded that the cross sectional dimensions chosen for the boom and the stick were appropriate, taking into account the material used and the relevant stresses. The diameter of the pins at each connecting joint was required and this was also obtained from the component force analysis.

The resultant force at each pin was calculated, and the cross sectional area and hence diameter of each pin was calculated using the design stress as the upper critical value. The diameter of the hydraulic pistons was also calculated using the force analysis. From the force analysis, the forces along the pistons were obtained, and using the largest force and the design stress, a suitable cross sectional area was found and hence diameter. Calculations were also done to ensure that the hydraulic pistons would not buckle under compression.

This made use of the basic buckling equation to obtain a critical buckling load, and then compare this to the largest force in any hydraulic piston. It was clearly seen that the pistons would not buckle since the actual load carried was less than the buckling load. To ensure that the excavator would not tip (topple), calculations were carried out to determine the position of the equivalent concentrated normal reaction when the bucket was filled to capacity at maximum reach. This involved taking moments about the centre of gravity and it was found that the reaction lay within the span of the tracks and hence the unit would not tip.

To determine the hydraulic pump capacity, the pressure required by them pump was calculated using the largest force and the area of application of the pressure to exert this force. To account for the pump not being 100% efficient, an extra 20% of pressure was added to give the final pump capacity in terms of pressure. The engine capacity was found by using the velocity at which the piston under the maximum force would be required to move. Similarly to the pump calculations, inefficiency of the engine was accounted for. To attain a maximum speed of 5km/hr as chosen in the design specifications, the angular velocity of the gear riving the tracks was calculated. 33 INDIVIDUAL ATTACHMENTS 1. Jackhammer – Gerard Mohammed The hydraulic hammer is used to break up concrete, asphalt or rock by producing repeating impacts to the surface. The mechanism behind the hydraulic hammer consists of two valves, which alternate to apply pressure to opposite ends of the piston. This causes reciprocation of the piston which transfers this motion to the tool and hence destructive blows are applied to the desired surface. A tool length of 525mm was chosen. The material chosen for the tool was AISI 1080 steel which is a high carbon steel known for hardness and resistance to wear.

The tool diameter was chosen as 75mm. Since the hammer will be designed to break concrete, a maximum breaking force of 30kN is used. To obtain safe working impact force that the tool can generate without buckling, the following formula was used: Pcr = ? 2EI / Le2 Where E: 190 ? 109 N/m2 I = ? d4/64 = 1. 553 ? 10-6 m4 Effective length, Le = 2 ? L = 1. 05m (One end fixed and one end free) Therefore: Pcr = 2. 6MN For sudden impact load, Force P = 2 ? Pstatic Maximum static force tool can withstand without buckling is 2. 6MN. Hence maximum impact force tool can withstand is 2. 6/2 = 1. 3MN.

Applying Factor of safety of 4 to this load, critical value = 325kN. Since maximum breaking force (30kN) is less than critical value, jackhammer tool will not buckle. 34 2. Bucket – Adrian Pirthysingh To find force acting in the bucket we use the density of soil and the volume of the bucket Density of soil is 1360Kg/m3 Volume of bucket is 0. 3m3 Density x volume = mass 1360 x 0. 3 = 408Kg F = mg 408 x 0. 3 = 4002. 5N For AISI 4340 steel the yield strength is 830MPa And we are using a factor of safety of 4 therefore the design stress is 207. 5Mpa In order to find the thickness we use the formula ? es = My/I Where M = WL/2 = (4002. 5 x 1. 4)/ 2 = 2801. 8 (for simplified calculations the bucket was assumed to be a cantilever beam and the length from the pin to the other end, when rolled out was found to be 1. 4m) y = t/2 = 0. 5t I = bt3/12 = 0. 6 t3 /12 = 0. 05t3 Therefore 207. 5 x 106 = (2801. 8 x 0. 5t)/0. 05t3 = 28018/t2 t2 = 28018/207. 5 x 106 = 0. 000135 t = 0. 0116m Hence thickness of material used to make the bucket was 12mm. 35 3. Compactor – Justin Kalloo A+ B= – 2 Base Plate dimensions: 12” x 18” or 305” x 457” Where F is the total Impulse Force. FA and FB are the forces at the bearings. The „ me? s the mass-eccentricity. ‘ W ‘ is the angular velocity of the unbalanced mass-eccentricity. For the design of the compactor: The total Impulse Force = 3000lbs = (3000 / 2. 2) Kg = 1363. 64Kg x 9. 81ms-2 = 13377. 3N The reaction at each bearing = (13377. 3N / 2) = 6688. 64N The reaction at each Rubber Mount = (13377. 3N / 4) = 3344. 32N W= 2 Where ‘ f ‘ is the frequency. Frequency „ f ‘ of this compactor = 2600rpm = 43. 333Hz Therefore W = 2 = 2 x ? x 43. 3333 = 272. 3rads-2 ? W2 = 0. 1804Kgm ‘ me’ mass-eccentricity = The „ e’ or eccentricity = 0. 025m Therefore the ‘ m’ of the rotating unbalanced is = 7. 2166Kg 6 Description of compactor: The type of compactor used is a hydraulic vibratory plate compactor. These types of compactors vibrate due to a rotating unbalance. As the hydraulics flow, it powers the motor in compactor itself. The motor rotates the mass-eccentricity about the shaft. Because of the mass-eccentricity that is, the offset in weight along the shaft, as the unbalance spins, it causes a vibration at the bearings and thus the whole plate vibrates. The magnitude of the vibration depends on the angular velocity of the rotating unbalance. As the angular velocity increases, the vibration also increases.

The rotating unbalance system and the plate are connected to four rubber mounts which in reality absorbs some of the energy of the system but due to the fact that the materials is rubber, this gives the plate an extra „ bounce? and also gives the system more freedom to actually move up and down thus enhancing the vibratory compactor? s output. 37 4. Auger – Shiv Raj Balroop A hydraulic auger is basically a drill that uses the physics behind hydraulics to obtain its power. It was first introduced in 1953 as a hydraulic drill but has improved in technology as time passed by and is now what it is today.

Augers usually consists of two major pieces, the auger bit, which does the actual cutting into of materials, and the hydraulic head, which is the hydraulic gerator motor that drives the auger bit. The workings of the hydraulic auger head obey the principles put forward by Blaise Pascal. Pascal said that the pressure is transmitted everywhere undiminished in an enclosed static fluid this just means that if there were two pistons, of different sizes, in a sealed completely filled hydraulic system, a large movement in the smaller piston would induce a small movement in the larger piston.

This also follows the Law of conservation of energy as what is put into the system is equal to what comes out of the system. This multiplication of force is a very useful property as now it wouldn? t be required to use such a large initial force, but instead have a very small initial force that would be multiplied by the use of hydraulics and then have a relatively huge output force. The auger works on this hydraulic principle which allows it to possess lots of power and torque required for the task it is required and also allows for the variation of speed and direction of motion.

Augers are used for drilling holes for footings, signs, fencings etc, in construction applications and for trees and shrubs in agricultural and landscaping applications. In this design project, it was required to design a mini excavator that could be used for small domestic applications. One additional requirement was the design of an auger attachment for the excavator. The addition of the auger to the excavator? s already existing features greatly improves the versatility of the excavator especially in terms of domestic uses.

Augers are used domestically for drilling holes for footings, fencings and in construction of foundations, etc, in construction applications and for planting of trees and shrubs in agricultural and landscaping applications. In this design, the auger was designed with the right amount of power, speed and torque for maximum productivity in the various soil types. It is desirable to have the proper amount of torque for the application so that when the auger bit enters the material it does not stick or stall or strain.

The right amount of speed is very important in creating the flow of the cut material up the auger bit and out of the hole being dug. The calculations for the auger can be seen on the next section. 38 CALCULATIONS FOR THE AUGER ATTACHMENT The design specifications of the largest auger bit and hydraulic motor are as follows: ? ? ? ? ? ? ? Maximum hydraulic flow = 5. 5 m3 per hour Maximum hydraulic pressure = 2100 kPa Driveshaft torque at maximum pressure = 2300 Nm Rpm at max pressure = 125 rpm Maximum mass of auger bit = 45 kg Maximum diameter of auger bit = 0. 3 m Maximum length of auger bit = 1. 2m The dimensions of the auger bit were 0. m diameter and 1. 2m in length. These dimensions were the most suitable dimensions for domestic applications such as the small construction application and the agricultural and landscaping application and such the bit was designed with this in mind. In the calculations of the power requirement, the specifications of the largest auger bit were used as this bit would have required the large power. To calculate the power requirements, the following formula was used: P= Txw Where, P = power required T = torque (driveshaft torque and torque on the shaft were approximately equal as they were both approximately the same size) w = angular velocity

Therefore, Ptheo = 2300 x (2? x 125)/60 Ptheo = 30106. 9296 W Ptheo, max = 30. 11 kW 39 But the value calculated above did not take into consideration the mass of the auger bit and the radius of the auger, so the following calculation accommodates for such: P= Txw But, T= Fxr And, F = mw2r Therefore, P = mw3r2 P = 45 x [2? x 125/60]3 x 0. 32 P = 9. 1 kW Therefore, the actual power required by this largest auger bit was calculated to be 9. 1 kW. This implies that if a slightly larger bit was designed for the auger, the hydraulic head would still be able to power it as efficiently as this smaller bit but with slightly more effort.

BUCKLING CALCULATIONS FOR THE AUGER Buckling is a failure mode that occurs in long slender columns. It usually occurs on these columns when they are loaded with a compressive force that surpasses their critical load. Therefore, the Buckling equation basically gives the maximum vertical load or critical load (compressive load) that a column can withstand without failure. The formula for Buckling is given below: 40 Pcr = [? 2EI] / Le2 Where, E = Young? s Modulus of material used for auger bit. I = second moment of area of auger bit Le = 0. 707L since one end of auger bit is clamped and the other end pinned.

In the auger design, the material used for the designing of the auger bit was medium carbon steel that was heat treated and tempered. The Young? s modulus of this material was found to be 2. 1 x 1011 N/m2. For the auger, the load acts on the centre shaft alone and not the entire shaft with the spiral. For this reason the second moment of area was taken for that of a cylindrical shaft. I for a cylindrical shaft is [? x d4]/64, where d was the diameter of the shaft. The diameter of the centre shaft was designed to be 0.

8m. Now, the last factor to be considered is the length of the centre shaft. The length of the centre shaft was taken as 1. m. By introducing these new details to the Buckling formula, the result is: Pcr = [? 2 x(2. 1 x 1011) x (? x 0. 84/64)] / (0. 707 x 1. 2)2 Pcr = 57. 90 MN The critical load of the centre shaft and therefore the auger was calculated to be 57. 90 MN, and can be interpreted to mean that if a compressive load greater than this value is exerted on the centre shaft of the auger; it can be expected to fail. 41

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