# DESIGN OF A DEMOLITION BOOM

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# ABSTRACT

DESIGN OF A DEMOLITION BOOM

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Excavators are used for many purposes. Some of these are digging, drilling, breaking and demolition. A demolition excavator boom consists of 3-piece boom which is different in form and construction from a 2-piece boom used in standard excavator. The aim of this thesis is to design a demolition boom for hydraulic excavator with operation weight of 30 ton. With this construction a higher reach is gained. Design of the demolition boom consists of three stages. Firstly the mechanism design is performed to determine the basic link dimensions. In the second step the structural shape of the boom is estimated to perform static stress analysis. The EXCEL program is chosen due to the ease of repetative calculations and applying the changes in structure parameters. The demolition boom is modeled by PRO-ENGINEER, and consequently the model is analyzed by using a Finite Element Analysis (FEA) in MSC.Marc-Mentat. According to the FEA results the model is revised.

Keywords: Demolition Excavator, Mechanism Design, Structural design

## ÖZ


BİNA YIKIM BOMU TASARIMI



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Yüksek Lisans, Makina Mühendisliği Bölümü

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Ekskavatörler birçok farklı amaç için kullanılır. Bunlardan bazıları kazma, kırma, delme ve bina yıkımı gibi işlerdir. Bu tez çalışmasının amacı 30ton çalışma ağırlığına sahip ekskavatörün bina yıkımında kullanılması için yıkıcı ekskavatör bomu tasarlamaktır. Standart ekskavatörlerin iki parça boma sahip olmasına karşın yıkıcı ekskavatörler üç parça boma sahiptir ve bom formu standart ekskavatör bom formundan farklıdır. Bu farklı bom formu sayesinde ekskavatörün daha yükseğe ulaşması sağlanmış olur. Yıkıcı bom tasarımı üç aşamadan oluşmaktadır. İlk aşama mekanizma tasarımı olup tasarım parametrelerinin belirlenmesi ve mekanizmanın oluşturulmasını kapsar. İkinci aşama ise yapısal tasarım olup tahmini bom ölçüleri için bomların mukavemet hesapları yapıldı. Hesapların tekrarlanma ve değişiklik yapma kolaylığı sebebiyle hesaplar için çoğunlukla Excel programı kullanılmıştır. Elde edilen bom ölçülerine göre bir çizim-tasarım programı olan Pro-Engineer'de bomların katı modeli oluşturuldu. Son olarak bu model bir sonlu elemanlar analizi programı olan MSC.Marc-Mentat ile analizi yapıldı. Analizden elde edilen veriler değerlendirilerek modelde değişiklikler yapıldı ve tasarıma son hali verildi.



Anahtar Kelimeler: Bina Yıkım Ekskavatörü, Mekanizma tasarımı, Yapısal Tasarım

**To My Parents Ayşe and M. Emin ÇETİN**

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# CHAPTER 1

# INTRODUCTION

## 1.1. Aim

Excavator booms are mainly designed for digging. If the excavator is to be used for another type of construction work, then auxiliary parts must be redesigned. The aim of this thesis is to design demolition boom for an excavator.  This thesis is concerned about the redesign the booms of a standard 300LC excavator for demolition purpose. Since all the computer programs are prepared parametric form, this work can easily be extended for the design of different range demolition booms.

## 1.2. Introduction to the Excavator

An excavator is defined as: "A mobile machine, which has an upper structure capable of continuous rotation and which digs, elevates, swings, and dumps material by action of the boom, the arm, or telescoping boom and the bucket. Not included are crane-type machines and their attachments (that is, cable clam, dragline, pile driver, and logging equipment, etc.)" [1]

Standard excavators are classified in two different categories. Firstly with respect to their travel train those are crawler type or wheeled excavators. The other category is operation weight. The varieties are mini excavators, mid range excavators and heavy-duty excavators. For example a crawler excavator (Figure 1.1) with operation weight of 30 ton is named 300LC (Long Crawler). Power source of an excavator is generally a diesel engine. The hydraulic pump uses the torque, from the internal combustion engine to produce hydraulic flow with a high pressure. The main actions are traveling, swinging and excavating. Traveling is provided by two hydraulic motors (travel motors). Swinging of the upper frame relative to the lower frame is provided by a hydraulic motor (swing motor). Excavating is provided by two cylinders for the boom, one cylinder for the arm and one cylinder for the bucket (Figure 1.2).


Figure 1.1: Standard hydraulic excavator

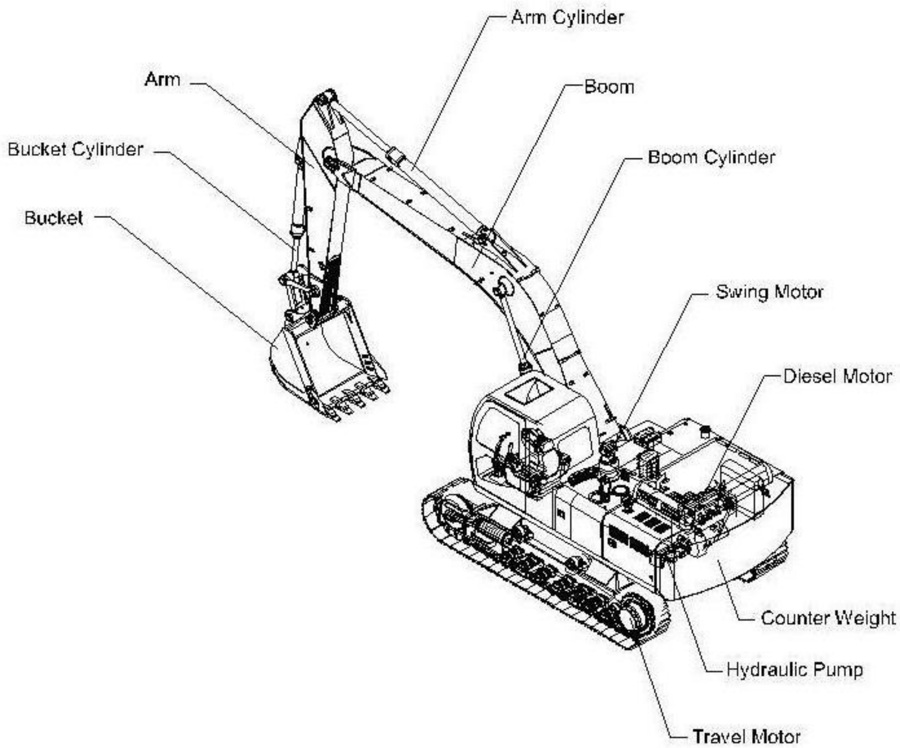



Figure 1.2: Basic parts of a standard hydraulic excavator

## 1.3. The Usage of Excavators

Digging is the main action for excavators, and the standard attachment used for excavation is the bucket. Excavators can as well be used for different purposes by different attachments instead of the bucket. For some purposes changing the attachment is insufficient because long reach is also needed (Table 1.1).

Table 1.1: Different attachments for different purposes

| | PURPOSE | ATTACHMENT |
|---|---|---|
| CHANGE THE ATTACHMENT | Digging | Bucket |
| | Drilling | Driller |
| | Breaking | Breaker |
| CHANGE THE ATTACHMENT & BOOMS | Cutting | Cutter |
| | Material Handling | Rehandling Grab |
| | Demolition Boom | Crusher, Cutter |


Figure1.3: An example for excavator used by changing the attachment only (Breaker)

Figure1.4 An example for excavator designed by changing the attachment and the booms for long reach (Material Handler)

## 1.4. Demolition Excavators

Excavators used for demolition are named as demolition excavators. Standard excavators can be used for demolition but for higher buildings they are insufficient. For higher reach longer boom and arm can be manufactured. Also for better out reach instead of a boom and an arm, a boom and two arms can be used. This system is named 3-piece boom: first piece is the main boom, second piece is the middle arm and the third piece is the end arm (Figure 1.5). Another advantage of 3-piece boom is the transportability of the excavator. Long booms cause difficulties in transportation but when it is more then 2- piece it can be folded to get better transportation positions.

When the out reach is increased by longer 3-piece boom, the tipping moment on the excavator is increased too. Instead of a bucket as the tool (end attachment), cutters (Figure 1.6) or crushers (Figure 1.7) are more effective for demolition. Unlike the bucket no moment is created on the excavator due to the action of these tools. These tools are also actuated by hydraulic power.


Figure 1.5: A demolition excavator


Figure1.6: A cutter

Figure1.7: A crusher

## 1.5. The Study Plan

An algorithm for the design of a demolition boom for an excavator is given in Figure1.8. As a first step the mechanism of the demolition boom will be designed. This step includes decision on the design parameters and design limitations. Taking care of these parameters and limitations the out reach of the excavator will be formed. The second step is to design the structure of it. Structure of the booms will be formed and their strengths will be calculated in this step, also they will be analyzed using finite element analysis method. Prototype is not included within the scope of this thesis.

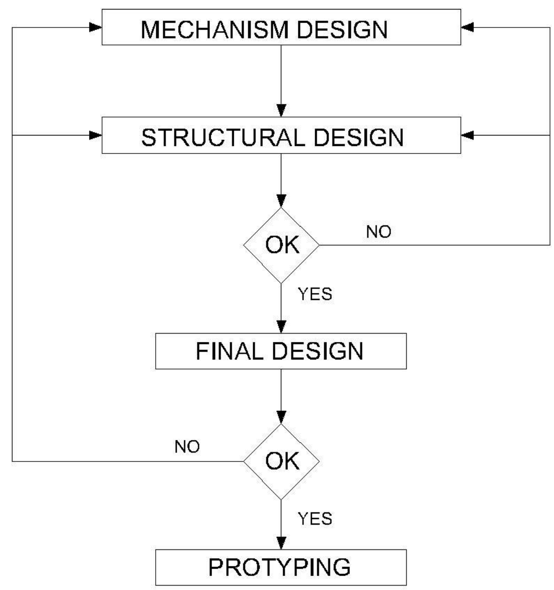



Figure 1.8: An algorithm for the design of a demolition boom

# CHAPTER 2

## LITERATURE SURVEY

In this section the previous works are briefly discussed that are carried out by the researches in the field of Excavator boom design and analysis.

Yener [2] has defined a computer interface called OPTIBOOM, which is a parametric modeler. This interface links the user to commercial Finite Element Analysis program, MSC. Marc-Mentat. Yener has studied the kinematic analysis and digging force calculations of excavator and parametric description of the boom geometry to create the structure of OPTIBOOM.

Sharma [3] has evaluated an existing design and has redesigned the robotic excavator attachments by an approach of kinematic and Finite Element Analysis of these attachments. The ultimate goal of this project is to reduce the stresses of excavator attachments.

W. Gutkowski, J. Bauer, Z. Iwanow and J. Putresza [4] mentioned the optimization of multi-arm mechanism elements. Mechanism is statically determined. They give two examples of optimization of excavator arms. They avoid consideration of the particular cross-section of each element by taking hinge reaction as an objective, the absolute value of which is to be minimized.

H. Ergin and O. Acaroglu [5]states that “A “road header” excavator has a special set of characteristics among tunnel excavation machines and the determination of the stability states of a road header is important for effective and continuous excavation.

For road headers having equal power, if one is more stable than another, it can respond to higher boom forces." They have developed a computer program to analyze the stability states of road headers by investigating the effects of machine design parameters.

E. Rusinski, J. Czmochowski, P.Moczki [6], discuss digging problems of machines used in underground mining and investigate of its reasons based on cracked boom of underground mine machine. They have used finite element method for numerical simulation. The paper "Numerical and experimental analysis of a mine's loader boom crack" provides information backed by evaluation and test results stating the nexus of causes of the boom failure.

E. Budney, M. Chlosta, W. Gutkowski [7], tried to minimize the time needed for bringing the filled bucket to the discharge place, and back to the digging site. They deal with an optimum problem of positioning an excavator bucket along prescribed trajectory using minimum time in the article. The paper is illustrated with numerical results giving some optimal trajectories.

J. Medanic, M. Yuan, B. Medanic [8], "The design of controllers to achieve stability, regulation and tracking for multi-link excavator is an open control problem due to the many nonlinear effects that have a dominant role within the wide region of link motions. A design model and a nonlinear polar controller are developed for the Caterpillar 325 excavator modeled as a two link system."

H. Iwami [9], "In accordance with advance in materials and hydraulic technology, size of excavators became large. At present, excavators with 700tons in weight and bucket size of 40m3 are being used all over the world. A paper explains severe working conditions for large excavators used in mining, fundamental structure, and production capacity in the field. In order to work continuously and maintain stable production, durability of main frames and reliability of parts become key points, and high accuracy stress analysis by FEM is contributing in the development. Preventive maintenance is important and mounting inspection systems on excavators is becoming usual. Hereafter, intelligent side approach such as installation of self-diagnosis function, trouble foreseeing system, etc. will become important."

S. Frimpong, Y. Li [10], develop dynamic models for real-time stress monitoring using a combination of flexible and rigid body approach. A virtual prototype simulator is developed to simulate the cable shovel excavation in oil sands and to examine the motion, stress and local deformation of the boom. The P&H 4100A cable shovel, deployed in the Athabasca oil sands formation, is used to examine the cable shovel boom durability using stress fields simulation. The results show that high stresses occur at regions around the joint between the upper body and the boom, resulting in large deformations. In hard formations, the results show that the stress fields in this region exceed the Von Mises yield strength of steel used in making the shovel boom components. The study provides a solid foundation for further study of failure life analysis of the cable shovel components.

Yefei Li, Xianghong Xu and Qinying Qiu [11], present "An application of Grid-enabled computing technologies in the field of engineering design optimization using Finite Element Method (FEM). Three essential elements in FEM-based structure optimization problems (CAD modeling, mesh and solution, and optimization) are integrated and automated with Grid-enabled computation Environment, dataset toolkits which is now being developed collect metadata to build a group level to describe the analysis tasks of the whole engineering problem, it allows access to remote computational resources and executing analysis tasks. A FEM-based Optimization is exposed to remote users and is applied to help FEM-based excavator working equipment analysis. A case running in this environment is shown in the end to validate design results."

# CHAPTER 3

# MECHANISM DESIGN

The main questions before beginning the design are:

a) What are the main features of a demolition excavator?
b) What are the design criteria's?
c) What are the design parameters?

This chapter is relevant to motion of the tool (crusher or cutter). The main feature is to get higher out reach, but there are limitations of the motion such as the tipping, the power of the engine, strengths of the bodies.

## 3.1. Main Features of the Demolition Excavator

All excavators used for demolition can be named "demolition excavator" but demolition excavators mostly have a long boom and long arms. In spite of standard excavators, which have 2-piece boom (boom and arm), demolition excavators have generally 3-piece boom (boom, middle arm and end arm) (Figure 3.1). The reason of this selection is that the 3-piece boom has more motion capability and better transportation position than a 2-piece longer boom as mentioned in Chapter 1.4.

The hydraulic actuators are two cylinders for boom motion, one cylinder for middle arm motion, one cylinder for end arm motion and one cylinder for the tool motion. There are also hydraulic actuators on the tool but the tool is not included in this study.

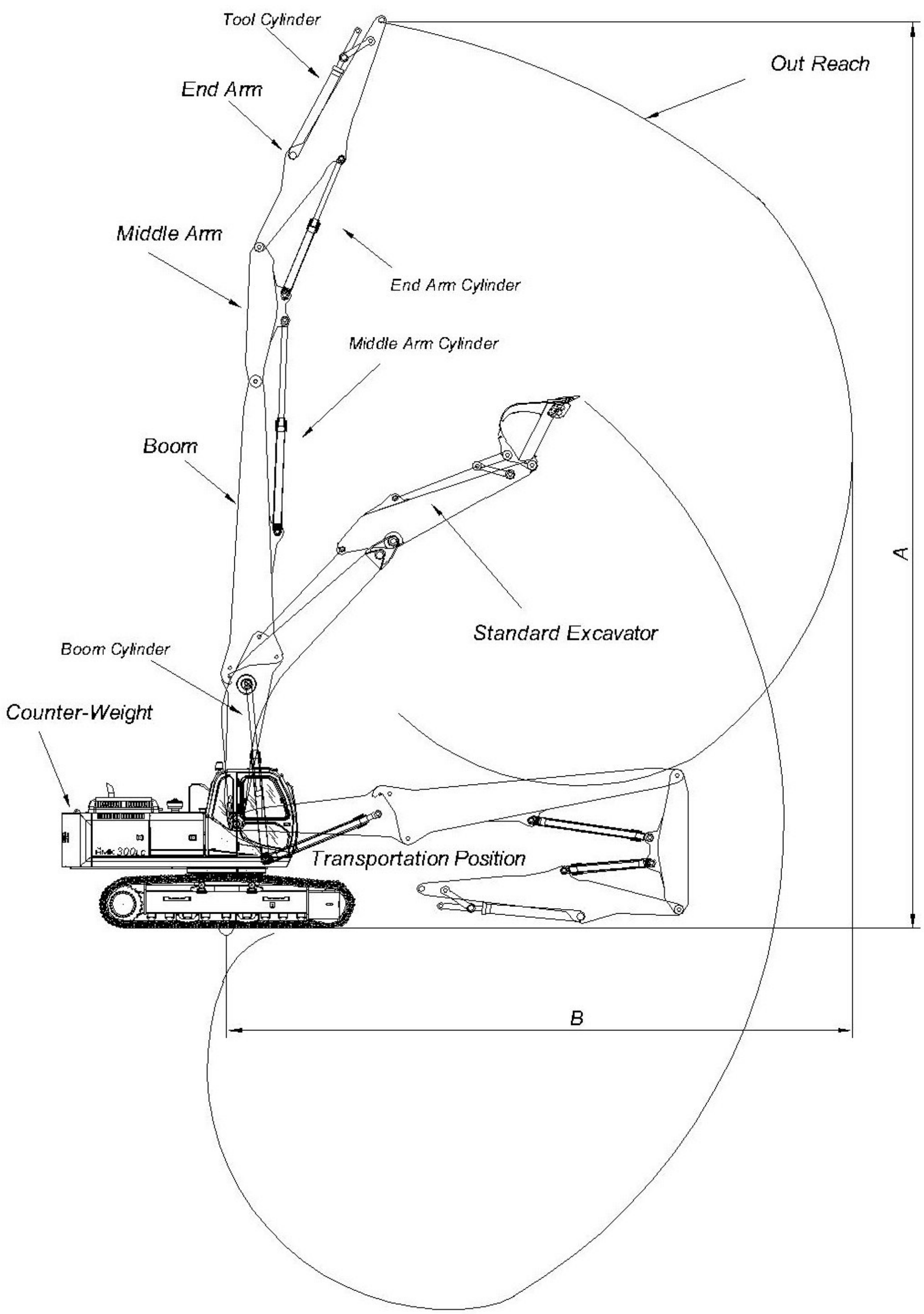



Figure 3.1: Main parts and out reach of a demolition excavator

### 3.2. Design Criteria

What is required from a demolition excavator? The same question can be asked as "what are the necessary properties of a marketable demolition excavator?". Of course the response is maximum high reach and maximum front reach. To determine these criteria the products of the other companies are searched [13], [14], [15], [16], [17], [18], [19].

The main sales criteria of demolition type excavators have been found as:

a) Operation weight: The weight of the whole excavator
b) Maximum pin height (A): Maximum high reach of the end arm-tool joint (Figure 3.1).
c) Maximum front reach (B): Maximum front reach of the end arm-tool joint (Figure 3.1).
d) Tool weight: The maximum tool weight which is allowed by the manufacturer
e) Ease of transportability: The transportation position should be considered during the design (Figure 3.1).

The important design criteria for mechanism design are maximizing the pin height, maximizing the front reach and meeting the need of transportation position while keeping the balancing weight within a reasonable magnitude. To design the mechanism, formulas of out reach drawing will be designed.

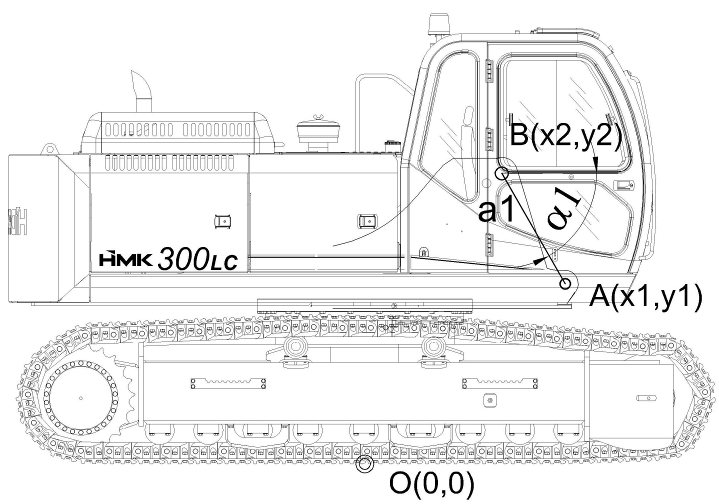
### 3.3. Design Parameters

Researching on the out reach, firstly design parameters must be determined. These parameters are the geometrical parameters of the bodies and the cylinders. Cylinders are the actuators which move the booms.

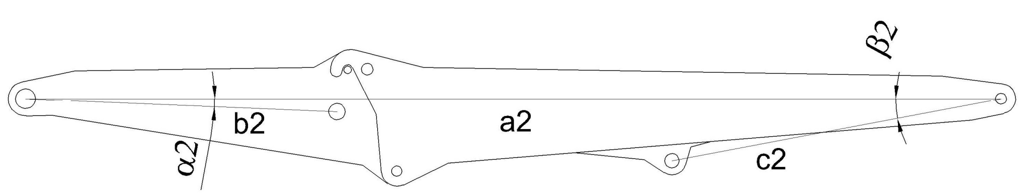
Four parts that make up the demolition boom are as (Figure 3.2):

a) Body-1: Rotating upper frame
b) Body-2: Boom
c) Body-3: Middle arm
d) Body-4: End arm

These parameters directly determine maximum working out reach, maximum pin height, and ease of transportability. Also they indirectly affect the dead weight.

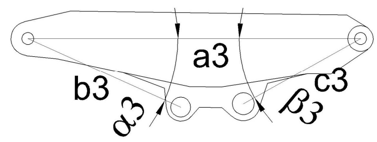



a) Upper frame

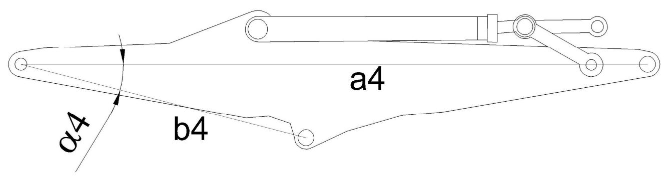



b) Boom

Figure 3.2: Main bodies




c) Middle arm




d) End arm

Figure 3.2 (continued): Main bodies

Definitions of main bodies' geometrical parameters are:

- The Origin is given as O(0,0) which is at the intersection point of ground and rotation axis of upper frame.
- Points A, B, C, D and E are the axis points of the revolute joints between the parts
- Upper Frame (Body-1):

  $a_1$= length between A and B
- Boom (Body-2):

  $a_2$= length of the boom

  $b_2$= distance from body joint to boom cylinder ($s_2$) joint on the boom

  $c_2$= distance from middle arm joint to middle arm cylinder joint on the boom
- Middle Arm (Body-3):

  $a_3$= length of the middle arm

$b_3$= distance from boom joint to middle arm cylinder ($s_3$) joint on the middle arm

$c_3$= distance from end arm joint to end arm cylinder ($s_4$) joint on the middle arm

- End Arm (Body-4):

  $a_4$= length of the end arm

  $b_4$= distance from middle arm joint to boom cylinder joint on the end arm

- The angles $\alpha_1$, $\alpha_2$, $\beta_2$, $\alpha_3$, $\beta_3$, $\alpha_4$ are given on Figure 3.2.

Demolition excavator is a three-degree of freedom system, and consists of five simple mechanisms, which work independently from each other. Hydraulic cylinders form inverted slider-crank mechanisms to actuate the three booms and the end tool. The fifth mechanism is a four bar mechanism which is not included in this study. The first objective is to determine the out reach of the system by considering the motion of the three bodies (boom, middle arm and end arm) actuated by the cylinders.

The parameters of the cylinders are:

$s_2$: length of the boom cylinder

$s_3$: length of the middle arm cylinder

$s_4$: length of the end arm cylinder

The angles $\theta_2$, $\theta_2^*$, $\theta_3$, $\theta_3^*$, $\theta_4$, $\theta_4^*$ are given in Figure 3.3 These parameters will describe the position of the booms, consequently the position of the tool.

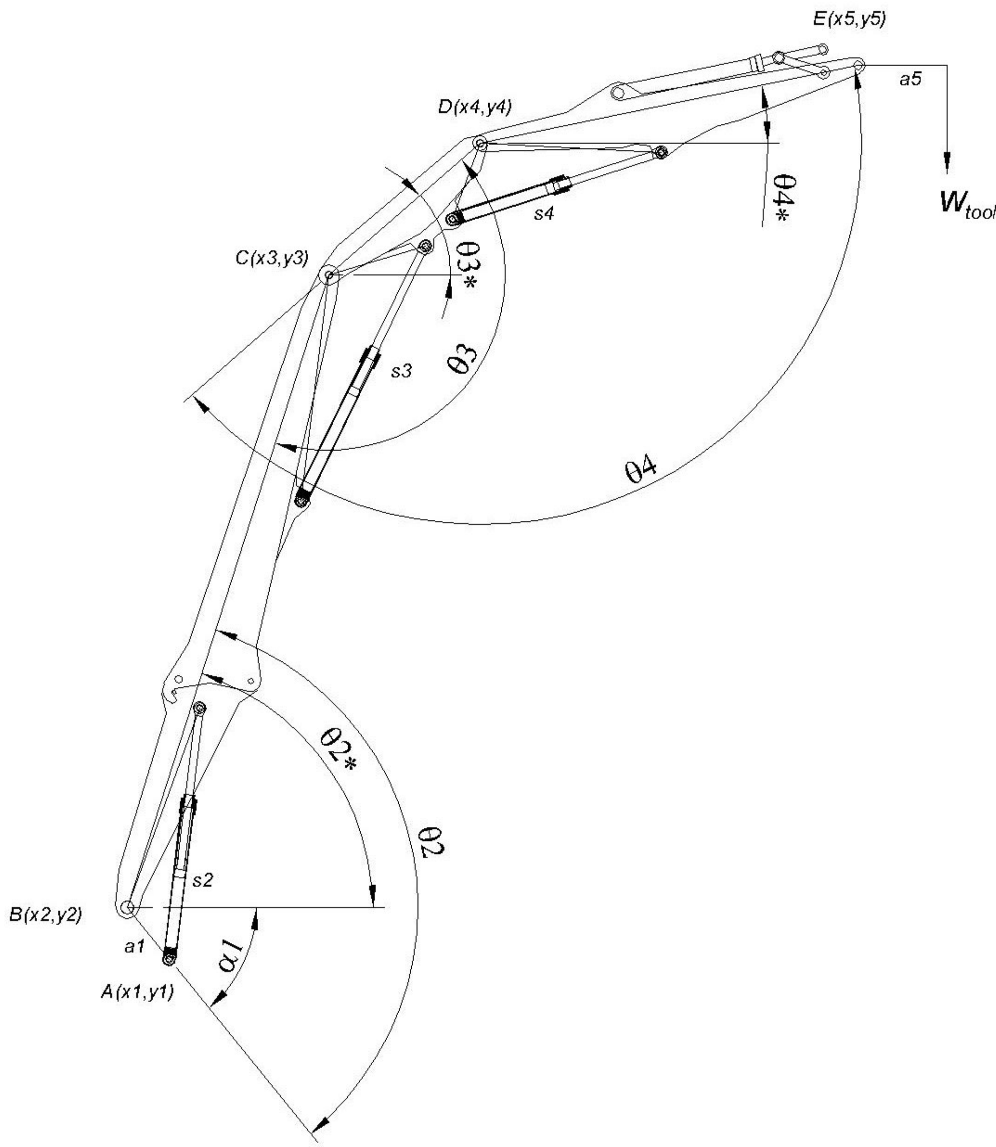



Figure 3.3: The Kinematic Parameters

### 3.4. The Out Reach

Out reach is the boundary curve of the working area of demolition attachment while the excavator lower and upper frames are fixed (Figure 3.4). The out reach is a function of the link dimensions and the stroke of the cylinders moving the booms. To determine the out reach firstly the parameters are studied and then the drawing algorithm and the formulas are formed.

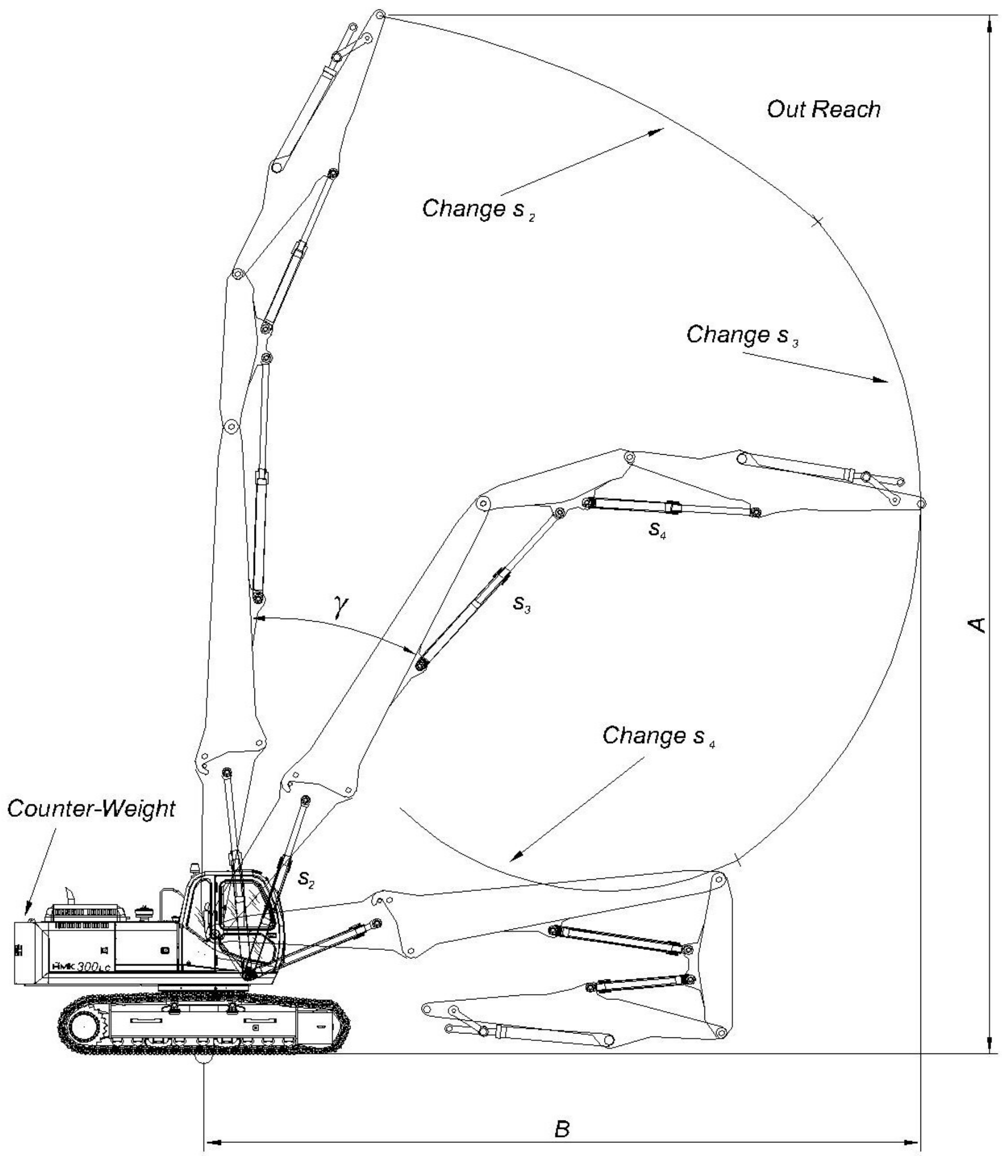



Figure 3.4: Schematic view of out reach

**Parameters:**

There are 25 parameters used in the formulation of the out reach drawing which are the coordinates of point E ($x_5$,$y_5$) with respect to the origin, dimensions of the main bodies ($a_1$, $a_2$, $b_2$, $c_2$, $a_3$, $b_3$, $c_3$, $a_4$, $b_4$, $a_5$, $\alpha_1$, $\alpha_2$, $\beta_2$, $\alpha_3$, $\beta_3$, $\alpha_4$), minimum lengths of the cylinders ($s_{2min}$, $\Delta s_2$, $s_{3min}$, $\Delta s_3$, $s_{4min}$, $\Delta s_4$), maximum allowed angular position of the boom ($\gamma$) (Figure3.3). Minimum length of boom cylinder ($s_2$) must be limited during working; because the excavator can tip down at the maximum front reach position considering the limited counter weight. To prevent the tipping; the motion is limited while working and this limitation is shown by an angular position $\gamma$ (Figure 3.4).

As given in chapter 3.3. the cylinder parameters are $s_2$, $s_3$ and $s_4$. And the limits of these parameters are minimum cylinder lengths and their strokes ($s_{2min} \leq s_2 \leq s_{2min} + \Delta s_2$, $s_{3min} \leq s_3 \leq s_{3min} + \Delta s_3$ and $s_{4min} \leq s_2 \leq s_{4min} + \Delta s_4$).

Out reach is the curve which is formed by connecting the furthermost points of the pin E($x_5$,$y_5$) (Figure 3.4). The algorithm of the out reach drawing is given in Figure 3.5. The initial conditions are taken for maximum cylinder strokes. Secondly the boom cylinder length is decreased with a small step size of $\Delta s_s$ until the minimum cylinder length is reacted. Then the middle arm and the end arm cylinder lengths are also decreased in that order with the same step size $\Delta s_s$. In every decreasing stage the position of the point E is calculated. The successive position of point E($x_5$,$y_5$) describes a curve which gives the limit for the working area.

The formulation of E is given in Appendix A. Maximum pin height (A in Figure 3.1) is the maximum value of vertical coordinate of point E and maximum front reach (B in Figure 3.1) is the maximum value of horizontal coordinate of point E.

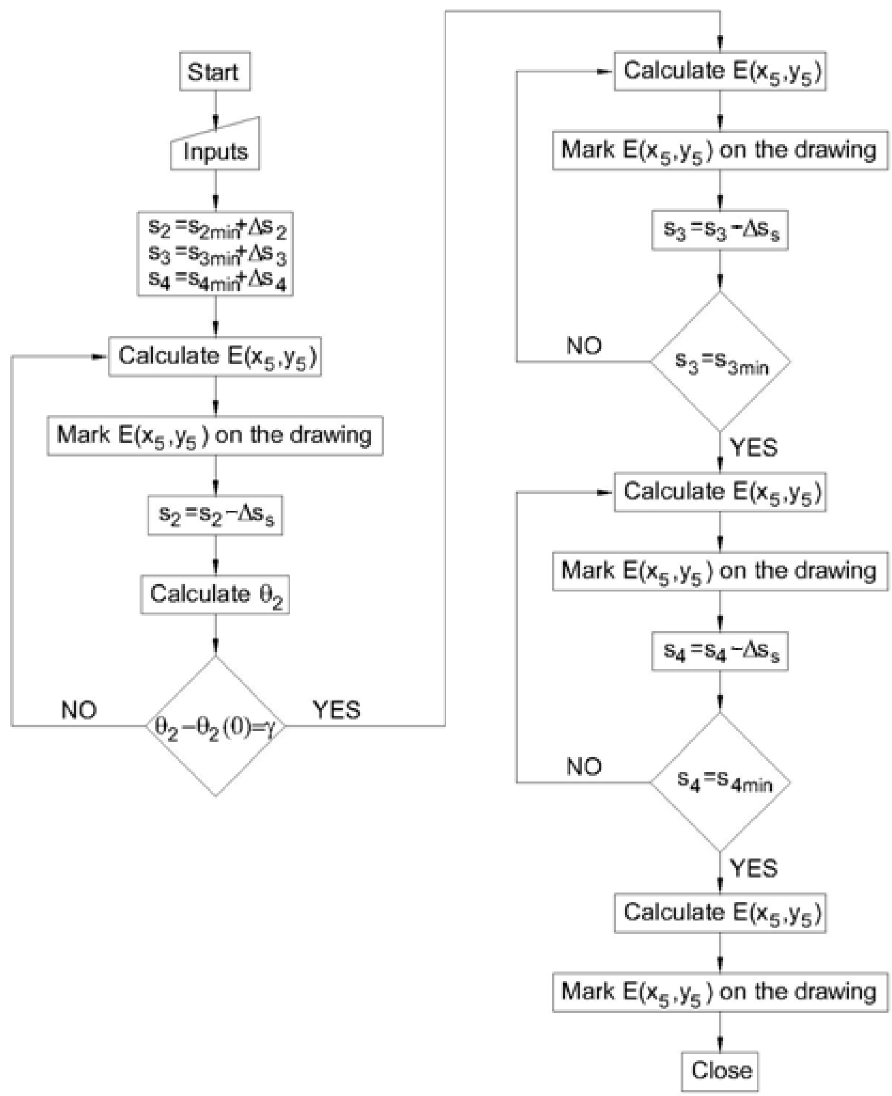



Figure 3.5: The algorithm of drawing the out reach

### 3.5. Basic Limitations of The Mechanism Design

After drawing the out reach, the values A and B are compared with the values of the competitors. The aim is to get better values. The inputs are changed and the calculations are repeated until adequate A and B values are obtained. Although these values can be increased, the increase in the counterweight and the engine capacity limits the out reach. The design criteria of the demolition excavator, which are given in chapter 3.2., will be compared with the available demolition excavators in the market. The design result must be equal with them or better than them.

There are two basic limitations in mechanism design; stability and transportation position. These two limitations should also be checked for acceptable values. Otherwise the values of the parameters must be changed and the process must be repeated until an acceptable solution is reached.

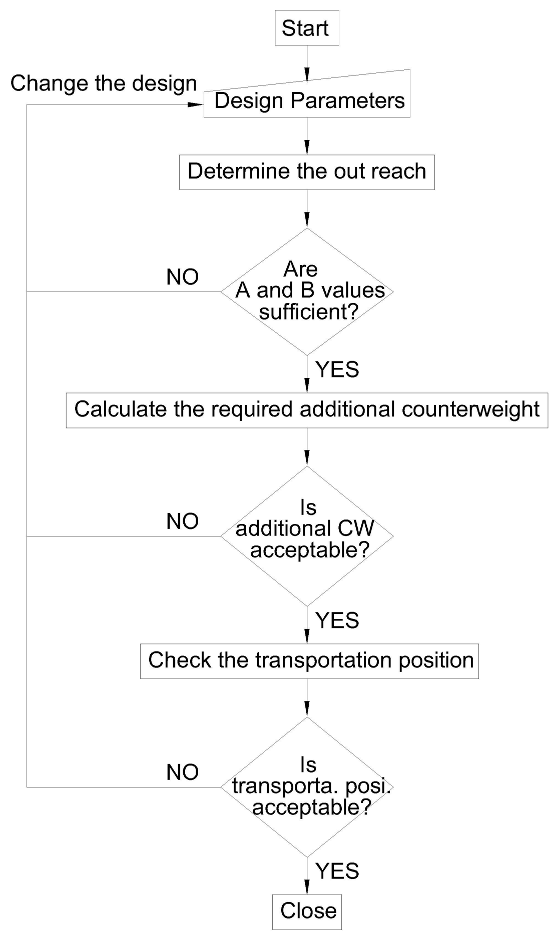



Figure 3.6: The algorithm of mechanism design

### 3.5.1. Stability Calculations

One of the main limitations of the mechanism design is the stability of the excavator. The excavator is said to be statically stable if the center of gravity of the total system is always within the limits of the palettes or wheels of the excavator when the boom is moving within the working range (Figure 3.7). Increasing the boom lengths will increase the weight of the front side and the standard counterweight to balance the boom mass will not be sufficient. Additional counterweight is required otherwise the excavator will tip over when the boom is moving with in the working range.

Additional counterweight will be calculated for the mechanism designed. If the additional counterweight is too large to be implemented than mechanism must be redesigned changing the parameters (Figure 3.6). These steps will be repeated until an acceptable out reach and an acceptable additional counterweight is obtained.

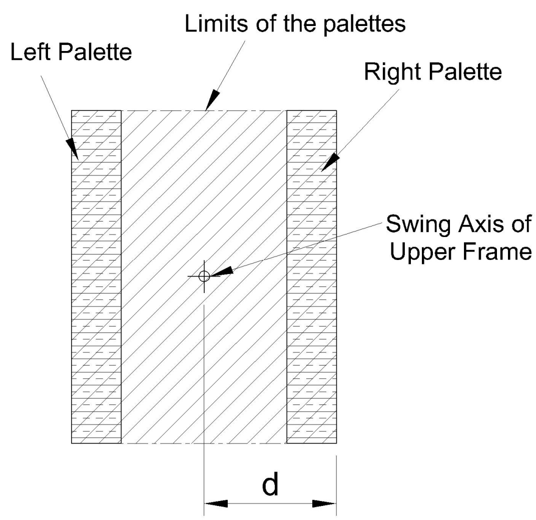



Figure 3.7: Top view of limits of the palettes

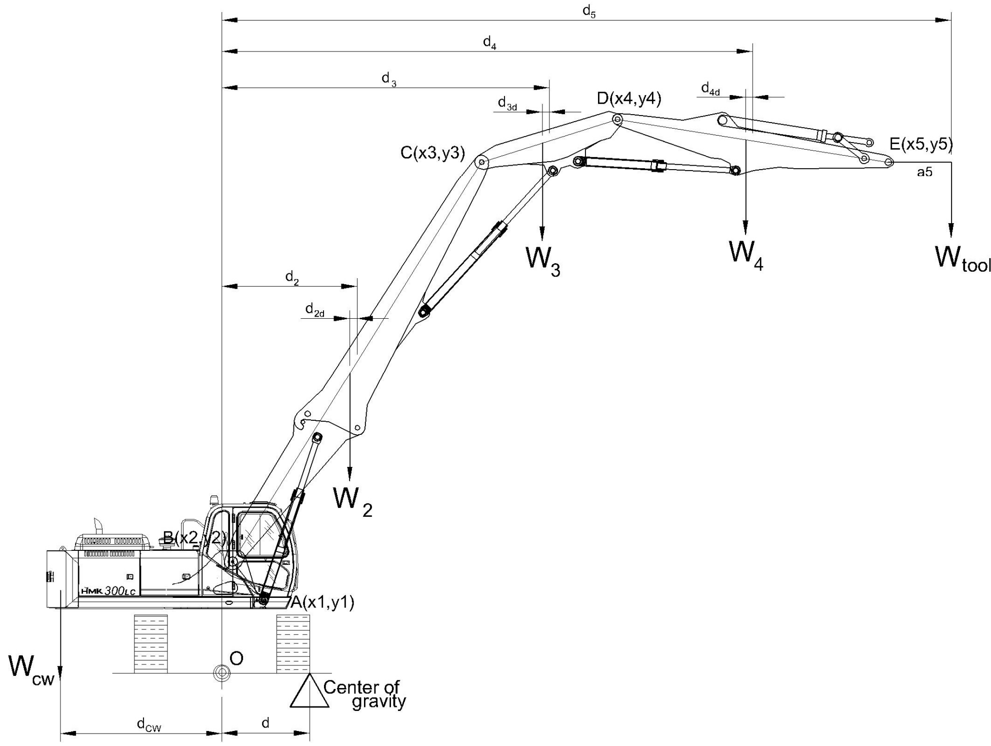



Figure 3.8: Locations of weights

The most critical location of the total center of gravity is the end of long crawler width. Because this position is the minimum distance between the rotation axis and the palette limit (d in Figure 3.7). The calculations for the minimum counter weight will be done at maximum front reach position because the weights are furthest from the centre of gravity at this position (Figure 3.8).

$$M_{cw} = M_{att} + M_{frame}$$

$$M_{cw} = W_{cw} \cdot d_{cw}$$

$$W_{cw} = \frac{M_{frame} + M_{att}}{d_{cw} + d}$$

Where;

$W_{cw}$= Counter weight

$d_{cw}$= Horizontal length between center of gravity of counterweight and total center of gravity.

$M_{att}$= Moment of attachments (boom, arms and tool) to the total center of gravity.

$M_{frame}$= Moment of the all other elements of standard excavator to the total center of gravity.

d= Distance between center of gravity and rotation axis

$$W_{cw} = \frac{-M_{frame} + W_2 \cdot (d_2 - d_{2d} - d) + W_3 \cdot (d_3 - d_{3d} - d) + W_4 \cdot (d_4 - d_{4d} - d) + \left(W_{tool} + F_{tool}{}^{y}\right) \cdot (d_5 - d)}{d_{cw} + d}$$

$$W_{cw(add)} = W_{cw} - W_{cw(standard)}$$

Where;

$$d_2 = \frac{x_3 - x_2}{2} + x_2$$

$$d_3 = \frac{x_4 - x_3}{2} + x_3$$

$$d_4 = \frac{x_5 - x_4}{2} + x_4$$

$$d_5 = x_5 + a_5$$

$W_{cw}$(add): Minimum additional counterweight required.

$W_{cw}$(standard): Counterweight of standard machine.

$d_{2d}$: horizontal distance between the linear mid point of the boom length and the real centre of gravity of the boom.

$d_{3d}$: horizontal distance between the linear mid point of the middle arm length and the real centre of gravity of the middle arm.

$d_{4d}$: horizontal distance between the linear mid point of the end arm length and the real centre of gravity of the end arm.

$F_{tool}{}^{y}$: Vertical component of force acting on the tool from the building or the weights of the scraps

At this step additional counterweight will be calculated approximately because the attachment weights and their centers of gravity are not known exactly. To get approximate boom weights it is considered that they are manufactured as box profile.

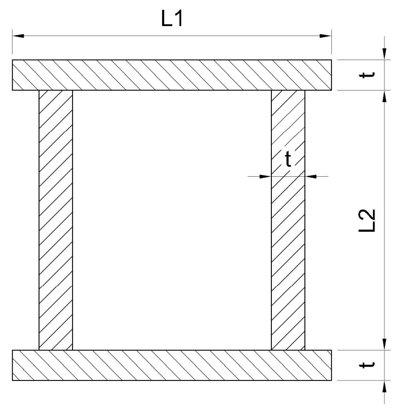



Figure 3.9: Box profile of the boom

Weight of the attachments:

$W=A_{ave}*a*\delta+W_{add}$

$A_{ave}=2*t*(L_1+L_2)$

Where;

$L_1$ : width of the boom

$L_2$ : height of the boom

t: thickness of the sheet

$A_{ave}$: average area

a: length of the boom

δ: density of steel ($0,785x10^{-5}kg/mm^3$)

$W_{add}$: additional weights for cylinders and other components

At the beginning $d_{2d}$, $d_{3d}$ and $d_{4d}$ values are taken as zero and $F_{tool}=W_{tool}$. After the structural design the real additional counterweight can be calculated using the real weights and the real center of gravities which will be taken by the designed model.

### 3.5.2. Transportation Position

The last step in mechanism design is to check the transportation position. If an acceptable transportation position could be obtained than the next step is structural design.

If only the booms are folded up, the appropriate position of transportation will be obtained. In order to make this the cylinders should be at minimum stroke positions. A special construction is used to fix the booms at that position during the transportation of the demolition excavator. An example is shown in Figure 3.9.

Figure 3.10: An example for demolition excavator transportation position

# CHAPTER 4

# STRUCTURAL DESIGN

This chapter deals with the design of the structure of the booms whose endurance is sufficient. In this design the cross-section dimensions of parts are determined. The aim of the structural design is modeling the demolition boom. The basic matter is the endurance of the booms during the work. A lot of dimensions are required for this process and are directly related to this matter. Using approximate weights the permissible external forces acting on the system, the forces acting on each part are determined and the critical cross-sections are checked.

The structural design algorithm is as shown in the Figure 4.1. First step is to determine the critical working position in which the effect of the weights and the external forces are more than the other positions. The weights will cause more moments on the excavator body at the maximum front reach position, so the critical position is selected as the maximum front reach position. The next step is to determine where the cross-sections will be taken on the booms for stress calculations in other words where the larges stresses will acquire. Locations of these sections and the dimensions of them are estimated regarding on former experiences. The forces and moments acting on each cross-section will be calculated which are required for the stress calculations. If the calculated stresses are higher than the permissible stress; these calculations must be repeated by changing the thicknesses of the materials or dimensions of the cross-section until getting acceptable stresses. To check the design stresses at the same cross-sections for different working positions will be also calculated. Using the final cross-sections the design will be modeled by a design program named Pro-Engineer. But the cross-section dimensions are not sufficient for drawing a model, the other dimensions are estimated. After drawing the model the mass distribution is redetermined and the forces are recalculated. If all calculated stresses are acceptable than the model will be analyzed by Finite Element Method.

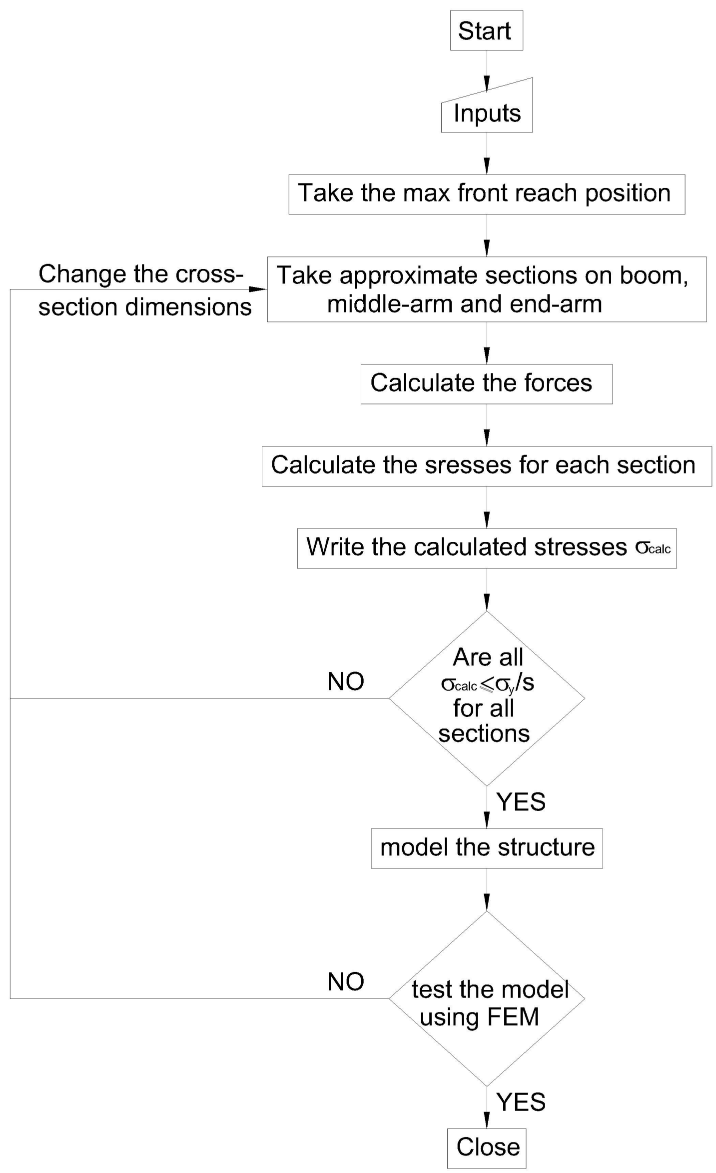



Figure 4.1: Algorithm of the structural design

### 4.1. Forces Acting on The Demolition Attachments

The Forces acting on the attachments are grouped into three types. The first type contains the cylinder forces which are the hydraulic forces created at the cylinders. Only these forces are powered by the engine. The second type contains the weights of the bodies. And the last type contains the external force which is the reaction force acted on the tool from the building during the demolition or the weight of the scraps cut from the building.

#### 4.1.1. The cylinder forces

The cylinder force is a force related to the hydraulic pressure and the rod inner surface diameter. The hydraulic pressure is generated at the cylinders which act as barriers to the hydraulic flow created by the main pump, and the hydraulic pump is actuated by the engine. So the cylinder forces are limited by the engine power, by the pump capacity and the cylinder dimensions. The general cylinder force formula is:

$F_s = P \cdot A$

Where;

$F_s$: Cylinder force.

P: Pressure of the hydraulic oil.

A: Rod inner surface area where the oil pressure act

Direction of cylinder force is normal to the cylinder rod inner surface. The directions of the cylinder forces which are equal to the angular positions of the cylinders are calculated below for the main cylinders of the attachments. These directions are defined with respect to the universal coordinate system that is x-axis parallel to the ground and y-axis perpendicular to the ground.

The magnitude of maximum cylinder force created by hydraulic pressure is:

$F_{smax} = P_{max} \cdot A$

Where;

$P_{max}$: Maximum hydraulic oil pressure at the cylinders. The anti-shock pressure will be taken for the calculations. Because overtaking the anti-shock pressure may cause leakage of hydraulic oil at the cylinders.

A: Rod inner surface area where the oil pressure act. The cylinder opens when the hydraulic pressure acts on the piston side and the cylinder closes when the hydraulic pressure acts on the rod side.

$A_2 = A_1 - A_r$

Where;

$A_1$: The piston side area

$A_2$: The rod side area

$A_r$: Rod bar cross section area

When the cylinder is opening the created cylinder force is larger than which created when closing. Because the piston side area is always larger than the rod side area.

a) Angular Position of The Boom Cylinder

Angular position of the boom cylinder changes with respect to the cylinder length. Cylinder length is the distance between the joint of piston of the cylinder and the joint of rod of the cylinder. Position of the cylinder rod with respect to the cylinder piston presents the boom angular position.

Formulation of the angular position of the boom cylinder (Figure 4.2): $\psi_2$

$$\psi_2 = \pi - \alpha_1 - \gamma_2$$

$$\gamma_2 = \cos^{-1} \cdot \left( \frac{a_1^2 + s_2^2 - b_2^2}{2 \cdot a_1 \cdot s_2} \right)$$

The descriptions of parameters are given in chapter 3.3

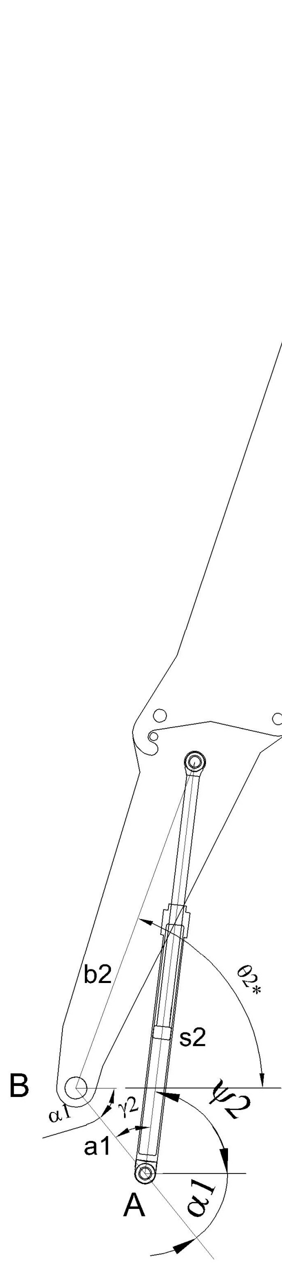



Figure 4.2: Angular position of boom cylinder: $\psi_2$

b) Angular Position of The Middle-Arm Cylinder

Formulation of the angular position of the middle-arm cylinder (Figure 4.3): $\psi_3$

$$\psi_3 = \theta_2^{\xi} + \beta_2 - \gamma_3$$

$$\gamma_3 = \cos^{-1}\left(\frac{c_2^2 + s_3^2 - b_3^2}{2 \cdot c_2 \cdot s_3}\right)$$

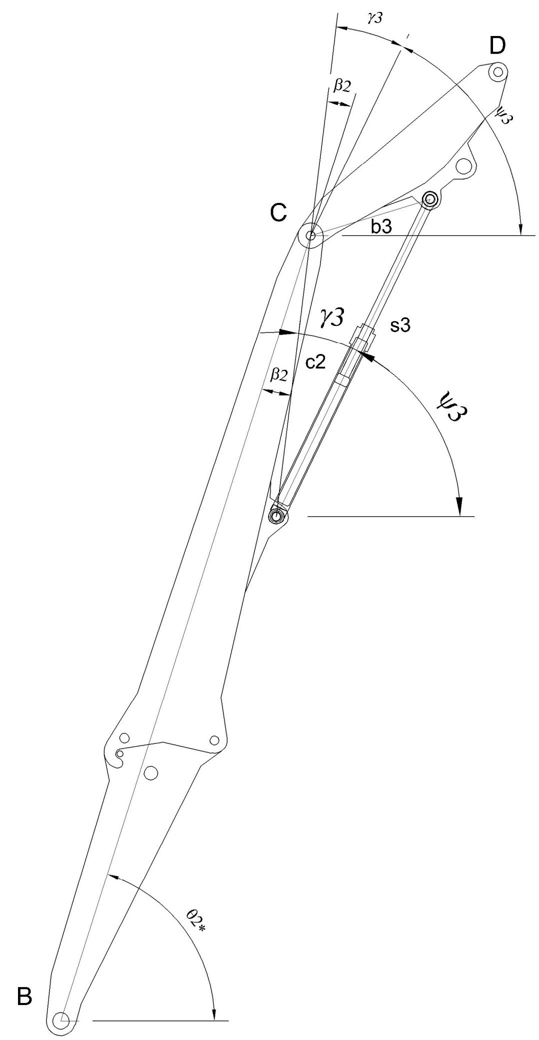



Figure 4.3: Angular position of middle-arm cylinder: $\psi_3$

c) Angular Position of The End-Arm Cylinder

Formulation of the angular position of the end-arm cylinder (Figure 4.4): $\psi_4$

$$\psi_4 = \theta_3^{\xi} + \beta_3 - \gamma_4$$

$$\gamma_4 = \cos^{-1}\cdot\left(\frac{c_3^2 + s_4^2 - b_4^2}{2 \cdot c_3 \cdot s_4}\right)$$

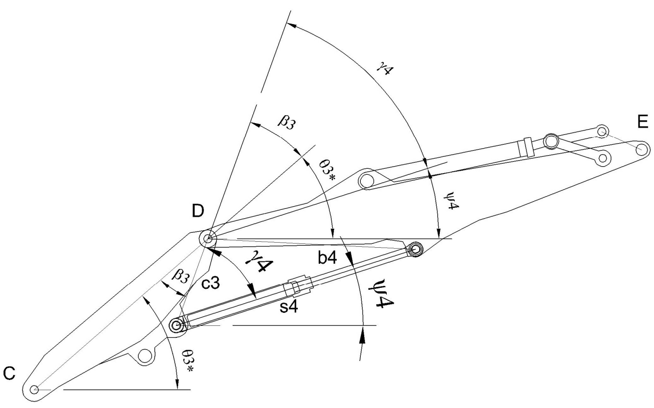



Figure 4.4: Angular position of end arm cylinder: $\psi_4$

### 4.1.2. The Weights of The Bodies

The main bodies are the attachments and the cylinders. Tool weight is limited by the excavator product class. And it is determined at the beginning the design based on marketability of the excavator. Weight of the boom, middle arm and end arm is taken approximately as in chapter 3.5.1. After drawing the CAD model of the booms, weights will be calculated more accurately by multiplying the volume, obtained from the CAD model, with the density of the material and the calculations will be repeated using these more accurate weights. The weights of the cylinders get from the cylinder manufacturer catalogues. The cylinder weights are added to the boom weights to shorten the calculations. Also there are some additional weights which are the pipes, hoses hydraulic oil etc. but these weights are negligible.

#### 4.1.3. Reaction Force Acting on the Tool

The main problem of the stress calculations is the reaction force of the building during the demolition or the weights of the scraps which will sometimes lifted with the tool. What will be the magnitude of the force and its direction? Two approximations have been used in this thesis to cope with this problem: One approximation is calculation of direction of the reaction force and the other is taking the direction of the force downward. The calculated reaction forces acting at the tool and the stresses obtained at a few region will be compared for each approximation. The approximation which gives maximum stress will also be used for finite element analysis.

The reaction force ($F_{tool}$) calculations have been done for maximum cylinder forces. This means that the cylinder may leak hydraulic oil when a force greater than $F_{tool}$ is applied on the tool.

##### 4.1.3.1. 1st Approximation: Calculate a Reaction Force Direction

1st Approximation is based on to calculate the reaction force direction which effect to reduce the maximum reaction force more than the other directions. The main idea is that this force direction is perpendicular to an imaginary line from the joint which is taken as moment center to the force acting point. Because this imaginary line is the largest distance for the moment calculations of the reaction force.

###### 4.1.3.1.1. Force Calculation Using Maximum Boom Cylinder Force

The external forces acting on the boom are the cylinder force ($Fs_2$), tool weight ($W_{tool}$), body weight of end arm ($W_4$), body weight of middle arm ($W_3$), body weight of boom ($W_2$), and reaction force acting on the tool ($F_{tool2}$) during the demolition process (Figure 4.5). $F_{s2}$ is taken maximum.

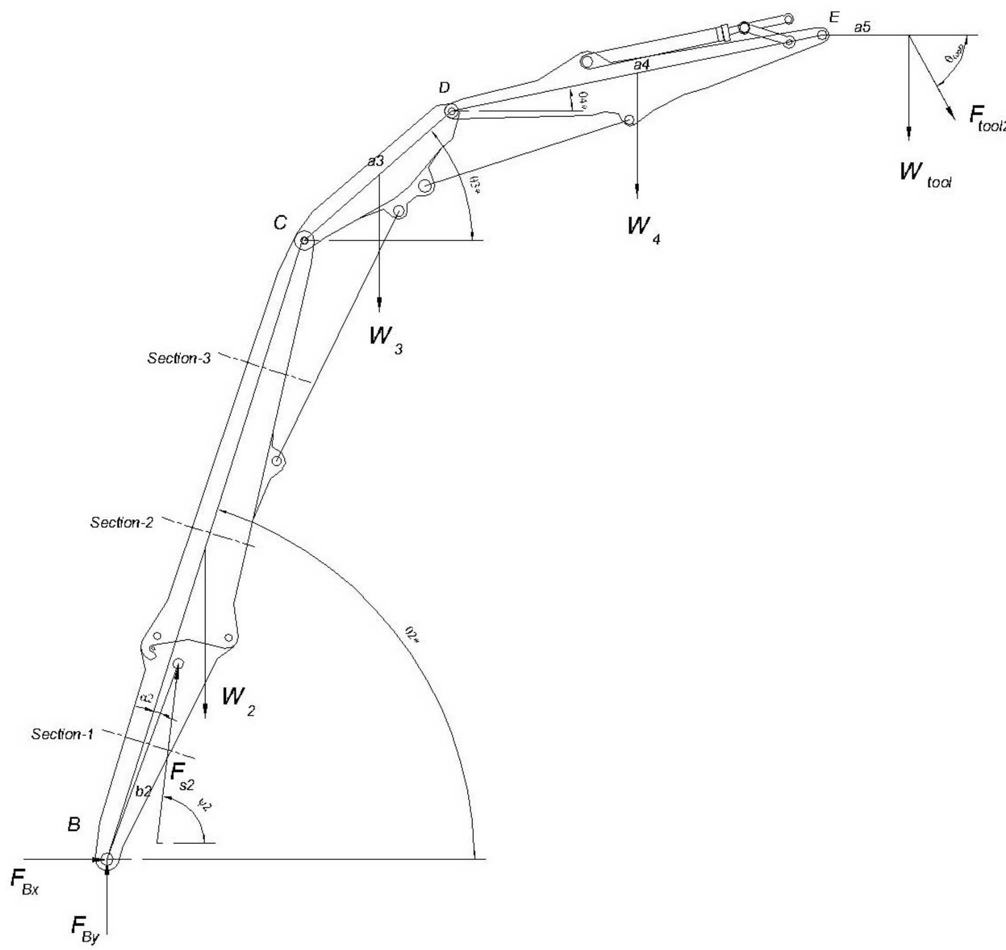



Figure 4.5: Free body diagram of boom

The moment of $F_{tool2}$ with respect to point B will be maximum when $\theta_{tool2}$ is taken as below (Figure4.6).

The direction of $F_{tool2}$:

$$\theta_{tool2} = \theta_2^{\zeta} - \cos^{-1}\left[\frac{e_2^{\,2} + a_2^{\,2} - \left[\left(x_5 + a_5 - x_3\right)^2 + \left(y_5 - y_3\right)^2\right]}{2 \cdot e_2 \cdot a_2}\right] - \frac{\pi}{2}$$

$$e_2 = \sqrt{\left(x_5 + a_5 - x_2\right)^2 + \left(y_5 - y_2\right)^2}$$

Where;

$e_2$: moment bar length of $F_{tool2}$

$\theta_{tool2}$: direction of $F_{tool2}$

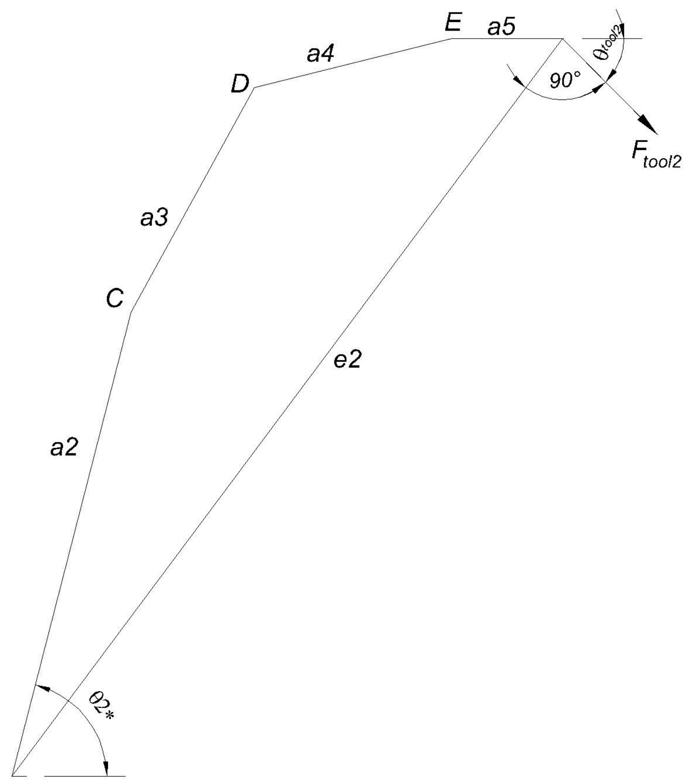



Figure 4.6: Direction of $F_{tool2}$

Force Formulations:

$$\sum M_B = 0$$

$$F_{tool2} = -\frac{-F_{s2x} \cdot b_2 \cdot \sin\left(\theta_2^{\xi} - \alpha_2\right) + F_{s2y} \cdot b_2 \cdot \cos\left(\theta_2^{\zeta} - \alpha_2\right) - W_2 \cdot \left(\frac{x_3 - x_2}{2}\right)}{e_2} + \frac{-W_3 \cdot \left(\frac{x_4 + x_3}{2} - x_2\right) - W_4 \cdot \left(\frac{x_5 + x_4}{2} - x_2\right) - W_{tool} \cdot \left(x_5 + a_5 - x_2\right)}{e_2}$$

Where;

$F_{tool2}$: Reaction force acted on the tool when cylinder force of boom is at the maximum value.

$$\sum F_x = 0$$

$$F_{Bx} = -F_{s2x} - F_{tool2} \cdot \cos\theta_{tool2}$$

$$\sum F_y = 0$$

$$F_{By} = -F_{s2y} + W_2 + W_3 + W_4 + W_{tool} - F_{tool2} \cdot \sin\theta_{tool2}$$

$$F_B = \sqrt{\left(F_{Bx}^2 + F_{By}^2\right)}$$

$$\theta_B = \tan^{-1}\left(\frac{F_{By}}{F_{Bx}}\right)$$

#### 4.1.3.1.2. Force Calculation Using Maximum Middle-Arm Cylinder Force

The external forces acting on the middle arm are the cylinder force ($F_{s3}$), tool weight ($W_{tool}$), body weight of end arm ($W_4$), body weight of middle arm ($W_3$), and reaction force acting on the tool ($F_{tool}$) during the demolition process (Figure 4.7). $F_{s3}$ is taken maximum.

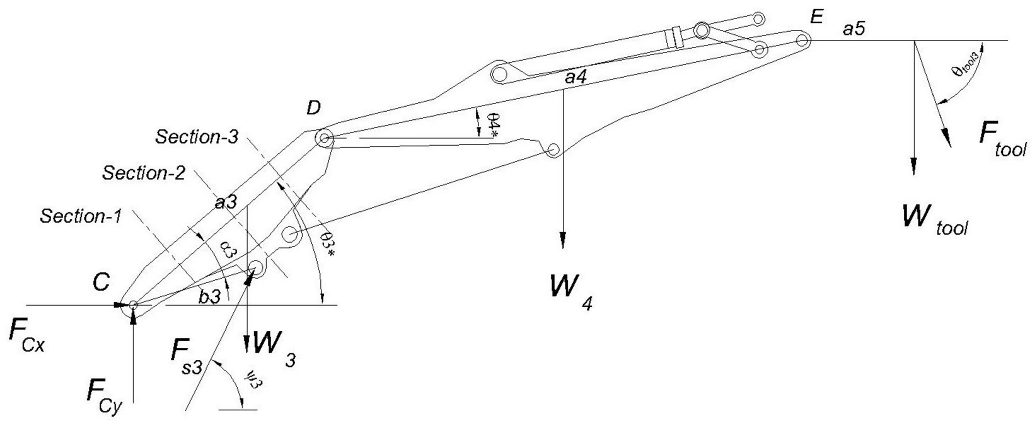



Figure 4.7: Free body diagram of middle arm

The moment of $F_{tool3}$ with respect to point C will be maximum when $\theta_{tool3}$ is taken as below (Figure 4.8):

The direction of $F_{tool3}$:

$$\theta_{tool3} = \theta_3^{\zeta} - \cos^{-1}\left[\frac{e_3^2 + a_3^2 - \left[\left(x_5 + a_5 - x_4\right)^2 + \left(y_5 - y_4\right)^2\right]}{2 \cdot e_3 \cdot a_3}\right] - \frac{\pi}{2}$$

$$e_3 = \sqrt{\left(x_5 + a_5 - x_3\right)^2 + \left(y_5 - y_3\right)^2}$$

Where;

$e_3$: moment bar length of $F_{tool3}$

$\theta_{tool3}$: direction of $F_{tool3}$

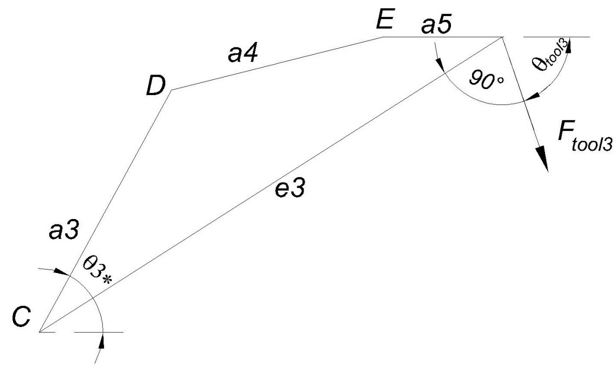



Figure 4.8: Direction of $F_{tool3}$

Force Formulations:

$$\sum M_C = 0$$

$$F_{tool3} = -\frac{-F_{s3x} \cdot b_3 \cdot \sin\left(\theta_3{}^{\xi} - \alpha_3\right) + F_{s3y} \cdot b_3 \cdot \cos\left(\theta_3{}^{\zeta} - \alpha_3\right)}{e_3} + \frac{-W_3 \cdot \left(\frac{x_4 - x_3}{2}\right) - W_4 \cdot \left(\frac{x_5 + x_4}{2} - x_3\right) - W_{tool} \cdot \left(x_5 + a_5 - x_3\right)}{e_3}$$

$$\sum F_x = 0$$

$$F_{Cx} = -F_{s3x} - F_{tool3} \cdot \cos\theta_{tool3}$$

$$\sum F_y = 0$$

$$F_{Cy} = -F_{s3y} + W_3 + W_4 + W_{tool} - F_{tool3} \cdot \sin\theta_{tool3}$$

$$F_C = \sqrt{\left(F_{Cx}{}^2 + F_{Cy}{}^2\right)}$$

$$\theta_C = \tan^{-1}\left(\frac{F_{Cy}}{F_{Cx}}\right)$$

#### 4.1.3.1.3. Force Calculation Using Maximum End-Arm Cylinder Force

The external forces acting on the end arm are the cylinder force (Fs4), tool weight ($W_{tool}$), body weight ($W_4$), and reaction force acting on the tool ($F_{tool}$) during the demolition process (Figure 4.9). $F_{s4}$ is taken maximum.

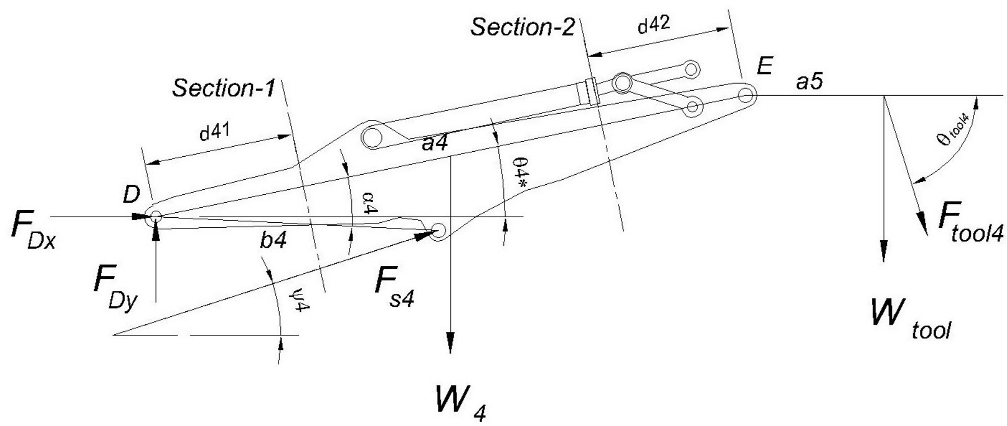



Figure 4.9 Free body diagram of end arm

The moment of $F_{tool}$ with respect to point D will be maximum when θ is taken as below (Figure 4.10):

The direction of $F_{tool4}$:

$$\theta_{tool4} = \theta_4^{\zeta} - \cos^{-1}\left(\frac{e_4^{\,2} + a_4^{\,2} - a_5^{\,2}}{2 \cdot e_4 \cdot a_4}\right) - \frac{\pi}{2}$$

$$e_4 = \sqrt{\left(x_5 + a_5 - x_4\right)^2 + \left(y_5 - y_4\right)^2}$$

Where;

$e_4$: moment bar length of $F_{tool4}$

$\theta_{tool4}$: direction of $F_{tool4}$

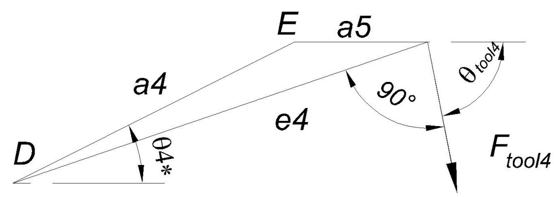



Figure 4.10: Direction of $F_{tool4}$

Force Formulations:

$$\sum M_D = 0$$

$$F_{tool4} = -\frac{F_{s4x}\cdot b_4\cdot \sin\left(\theta_4^{\xi} - \alpha_4\right) + F_{s4y}\cdot b_4\cdot \cos\left(\theta_4^{\zeta} - \alpha_4\right) - W_4\cdot\left(\frac{x_5 - x_4}{2}\right) - W_{tool}\cdot\left(x_5 + a_5 - x_4\right)}{e_4}$$

$$\sum F_x = 0$$

$$F_{Dx} = -F_{s4x} - F_{tool4}\cdot \cos\theta_{tool4}$$

$$\sum F_y = 0$$

$$F_{Dy} = -F_{s4y} + W_4 + W_{tool} - F_{tool4}\cdot \sin\theta_{tool4}$$

$$F_D = \sqrt{\left(F_{Dx}^{\ 2} + F_{Dy}^{\ 2}\right)}$$

$$\theta_D = \tan^{-1}\left(\frac{F_{Dy}}{F_{Dx}}\right)$$

### 4.1.3.2. 2nd Approximation: Take the Reaction Force Direction Downward

2nd Approximation is based on calculating a reaction force downward direction. Such a downward force is taken because that the demolition tool also carries the scraps that it cut from the building. Thus another critical force is also taken into consideration.

#### 4.1.3.2.1. Force Calculation Using Maximum Boom Cylinder Force

The external forces acting on the boom are tool weight ($W_{tool}$), body weights ($W_2$, $W_3$ and $W_4$), and reaction force acting on the tool ($F_{tool}$) during the demolition process.

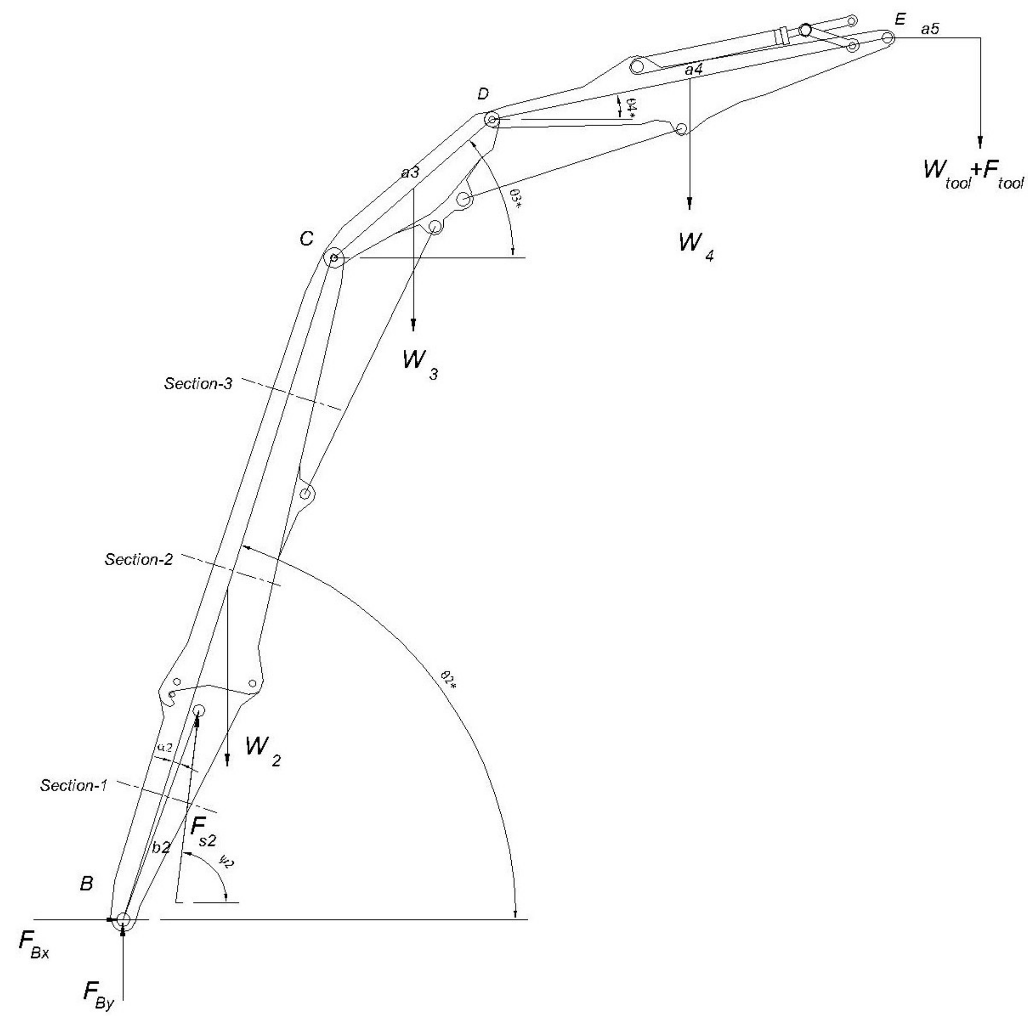



Figure 4.11: Free body diagram of boom

$$\sum M_B = 0$$

$$F_{tool2} = -\frac{-F_{s2x} \cdot b_2 \cdot \sin\left(\theta_2^{\xi} - \alpha_2\right) + F_{s2y} \cdot b_2 \cdot \cos\left(\theta_2^{\zeta} - \alpha_2\right) - W_2 \cdot \left(\frac{x_3 - x_2}{2}\right)}{x_5 + a_5 - x_2} + \frac{-W_3 \cdot \left(\frac{x_4 + x_3}{2} - x_2\right) - W_4 \cdot \left(\frac{x_5 + x_4}{2} - x_2\right) - W_{tool} \cdot \left(x_5 + a_5 - x_2\right)}{x_5 + a_5 - x_2}$$

$$\sum F_x = 0$$

$$F_{Bx} = -F_{s2x}$$

$$\sum F_y = 0$$

$$F_{By} = -F_{s2y} + W_2 + W_3 + W_4 + W_{tool} - F_{tool2}$$

$$F_B = \sqrt{\left(F_{Bx}^{\ 2} + F_{By}^{\ 2}\right)}$$

$$\theta_B = \tan^{-1}\left(\frac{F_{By}}{F_{Bx}}\right)$$

#### 4.1.3.2.2. Force Calculation Using Maximum Middle-Arm Cylinder Force

The external forces acting on the middle arm are tool weight ($W_{tool}$), body weights ($W_3$ and $W_4$), and reaction force acting on the tool ($F_{tool}$) during the demolition process.

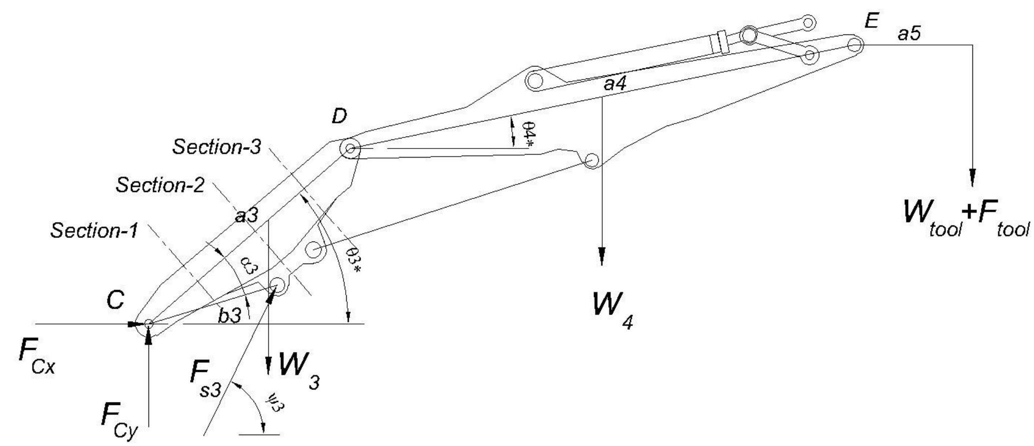



Figure 4.12: Free body diagram of middle arm

$$\sum M_C = 0$$

$$F_{tool3} = -\frac{-F_{s3x} \cdot b_3 \cdot \sin\left(\theta_3{}^{\xi} - \alpha_3\right) + F_{s3y} \cdot b_3 \cdot \cos\left(\theta_3{}^{\zeta} - \alpha_3\right)}{x_5 + a_5 - x_3} + \frac{-W_3 \cdot \left(\frac{x_4 - x_3}{2}\right) - W_4 \cdot \left(\frac{x_5 + x_4}{2} - x_3\right) - W_{tool} \cdot \left(x_5 + a_5 - x_3\right)}{x_5 + a_5 - x_3}$$

$$\sum F_x = 0$$

$$F_{Cx} = -F_{s3x}$$

$$\sum F_y = 0$$

$$F_{Cy} = -F_{s3y} + W_3 + W_4 + W_{tool}$$

$$F_C = \sqrt{\left(F_{Cx}{}^2 + F_{Cy}{}^2\right)}$$

$$\theta_C = \tan^{-1}\left(\frac{F_{Cy}}{F_{Cx}}\right)$$

#### 4.1.3.2.3. Force Calculation Using Maximum End-Arm Cylinder Force

The external forces acting on the end arm are tool weight ($W_{tool}$), body weight ($W_4$), and reaction force acting on the tool ($F_{tool}$) during the demolition process.

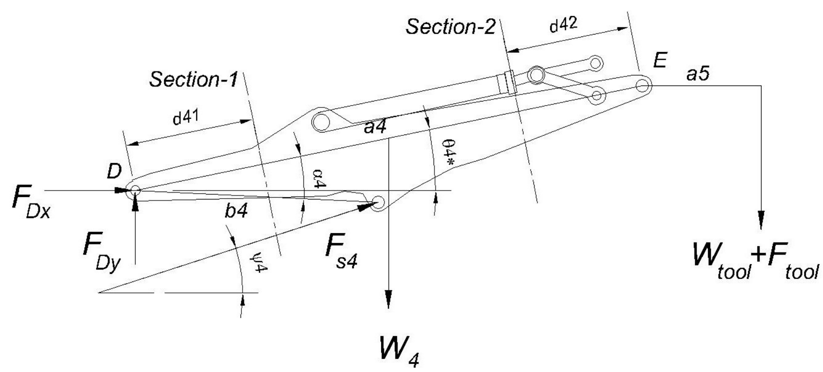



Figure 4.13: Free body diagram of end arm

$$\sum M_D = 0$$

$$F_{tool4} = -\frac{F_{s4x} \cdot b_4 \cdot \sin\left(\theta_4^{\xi} - \alpha_4\right) + F_{s4y} \cdot b_4 \cdot \cos\left(\theta_4^{\zeta} - \alpha_4\right) - W_4 \cdot \left(\frac{x_5 - x_4}{2}\right) - W_{tool} \cdot \left(x_5 + a_5 - x_4\right)}{x_5 + a_5 - x_4}$$

$$\sum F_x = 0$$

$$F_{Dx} = -F_{s4x}$$

$$\sum F_y = 0$$

$$F_{Dy} = -F_{s4y} + W_4 + W_{tool} + F_{tool4}$$

$$F_D = \sqrt{\left(F_{Dx}^2 + F_{Dy}^2\right)}$$

$$\theta_D = \tan^{-1}\left(\frac{F_{Dy}}{F_{Dx}}\right)$$

### 4.2. Stress Analysis

The boom, middle arm and end arm have box type cross sections. Stress formulation of the box type cross section is given in Appendix B. The stress analysis of $1^{st}$ and $2^{nd}$ Approximations have been done by using the same calculations and they are given subsequently. The calculations have been done for sections taken at several locations for each boom.

#### 4.2.1. Stress Analysis of Boom

There are three sections taken on boom for stress analysis. One of them is between point B and joint of boom cylinder, second is between boom cylinder and joint of middle arm cylinder and the other is between joint of middle arm cylinder and point C.

##### 4.2.1.1. Stress Analysis Of Section-1

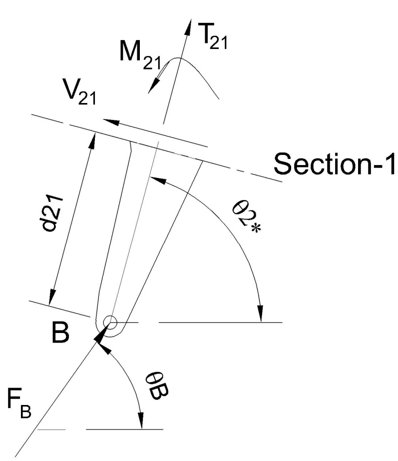



Figure 4.14: Section-1 on boom

Section-1 is taken between point B and joint of cylinder of boom.

$$T_{21} = -F_B \cdot \cos\left(\theta_B - \theta_2^{\zeta}\right)$$

$$V_{21} = -F_B \cdot \sin\left(\theta_B - \theta_2^{\zeta}\right)$$

$$M_{21} = F_B \cdot \sin\left(\theta_B - \theta_2^{\zeta}\right) \cdot d_{21}$$

$$\sigma_{upper21} = -\frac{M_{21}}{Z_{upper21}} + \frac{T_{21}}{A_{s21}} < \frac{\sigma}{S}$$

$$\sigma_{lower21} = \frac{M_{21}}{Z_{lower21}} + \frac{T_{21}}{A_{s21}} < \frac{\sigma}{S}$$

#### 4.2.1.2. Stress Analysis Of Section-2

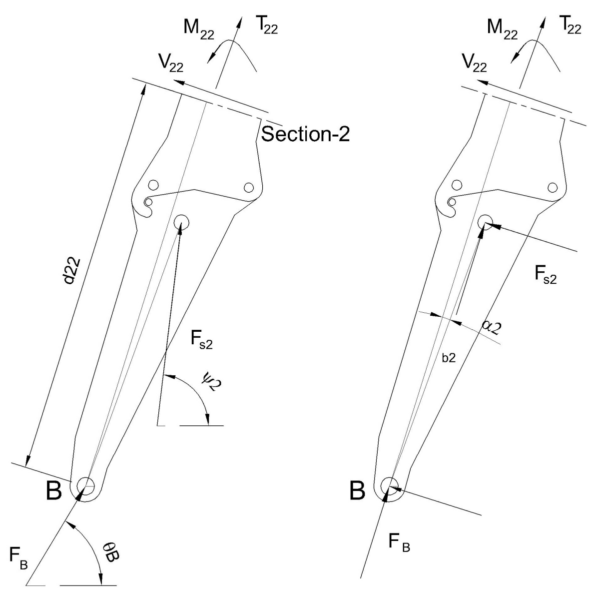



Figure 4.15: Section-2 on boom

Section-2 is taken between joint of cylinder of boom and joint of cylinder of middle arm.

$$T_{22} = -F_B \cdot \cos\left(\theta_B - \theta_2^{\zeta}\right) - F_{s2} \cdot \cos\left(\psi_2 - \theta_2^{\zeta}\right)$$

$$V_{22} = -F_B \cdot \sin\left(\theta_B - \theta_2^{\zeta}\right) - F_{s2} \cdot \sin\left(\psi_2 - \theta_2^{\zeta}\right)$$

$$M_{22} = F_B \cdot \sin\left(\theta_B - \theta_2^{\zeta}\right) \cdot d_{22} + F_{s2} \cdot \sin\left(\psi_2 - \theta_2^{\zeta}\right) \cdot \left(d_{22} - b_2 \cdot \cos\alpha_2\right) - F_{s2} \cdot \cos\left(\psi_2 - \theta_2^{\zeta}\right) \cdot \left(b_2 \cdot \sin\alpha_2\right)$$

$$\sigma_{upper22} = -\frac{M_{22}}{Z_{upper22}} + \frac{T_{22}}{A_{s22}} < \frac{\sigma}{S}$$

$$\sigma_{lower22} = \frac{M_{22}}{Z_{lower22}} + \frac{T_{22}}{A_{s22}} < \frac{\sigma}{S}$$

### 4.2.1.3. Stress Analysis Of Section-3

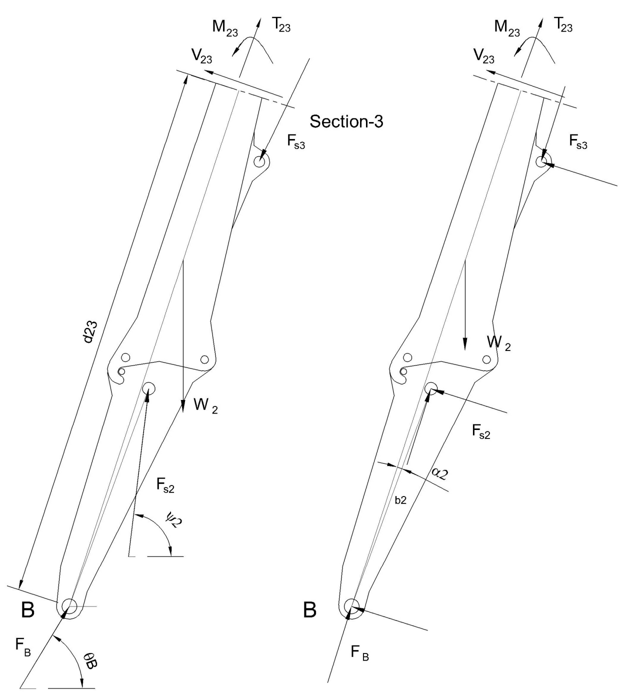



Figure 4.16: Section-3 on boom

Section-3 is taken between joint of cylinder of middle arm and point C.

$$F_{s32} = \frac{W_3 \cdot \left( \frac{x_4 - x_3}{2} \right) + W_4 \cdot \left( \frac{x_5 + x_4}{2} - x_3 \right) + W_{tool} \cdot (x_5 + a_5 - x_3) + F_{tool2}{}^{x} \cdot (y_5 - y_3) - F_{tool2}{}^{y} \cdot (x_5 + a_5 - x_3)}{b_3 \cdot \cos\left( \theta_3{}^{\zeta} - \alpha_3 \right) \cdot \sin\psi_2 - b_3 \cdot \sin\left( \theta_3{}^{\zeta} - \alpha_3 \right) \cdot \cos\psi_2}$$

$$T_{23} = -F_B \cdot \cos\left( \theta_B - \theta_2{}^{\zeta} \right) - F_{s2} \cdot \cos\left( \psi_2 - \theta_2{}^{\zeta} \right) + F_{s32} \cdot \cos\left( \psi_3 - \theta_2{}^{\zeta} \right) + W_2 \cdot \cos\left( \frac{\pi}{2} - \theta_2{}^{\zeta} \right)$$

$$V_{23} = -F_B \cdot \sin\left( \theta_B - \theta_2{}^{\zeta} \right) - F_{s2} \cdot \sin\left( \psi_2 - \theta_2{}^{\zeta} \right) + F_{s32} \cdot \sin\left( \psi_3 - \theta_2{}^{\zeta} \right) + W_2 \cdot \sin\left( \frac{\pi}{2} - \theta_2{}^{\zeta} \right)$$

$$\begin{aligned} M_{23} = {} & F_B \cdot \sin\left( \theta_B - \theta_2{}^{\zeta} \right) \cdot d_{23} + F_{s2} \cdot \sin\left( \psi_2 - \theta_2{}^{\zeta} \right) \cdot \left( d_{23} - b_2 \cdot \cos\alpha_2 \right) - F_{s2} \cdot \cos\left( \psi_2 - \theta_2{}^{\zeta} \right) \cdot \left( b_2 \cdot \sin\alpha_2 \right) \\ & + F_{s32} \cdot \sin\left( \psi_3 - \theta_2{}^{\zeta} \right) \cdot \left( d_{23} + c_2 \cdot \cos\beta_2 - a_2 \right) + F_{s32} \cdot \cos\left( \psi_3 - \theta_2{}^{\zeta} \right) \cdot \left( c_2 \cdot \sin\beta_2 \right) - W_2 \cdot \sin\left( \frac{\pi}{2} - \theta_2{}^{\zeta} \right) \cdot \left( d_{23} - \frac{a_2}{2} \right) \end{aligned}$$

$$\sigma_{upper23} = -\frac{M_{23}}{Z_{upper23}} + \frac{T_{23}}{A_{s23}} < \frac{\sigma}{S}$$

$$\sigma_{lower23} = \frac{M_{23}}{Z_{lower23}} + \frac{T_{23}}{A_{s23}} < \frac{\sigma}{S}$$

### 4.2.2. Stress Analysis of Middle arm

There are three sections taken on middle arm for stress analysis. One of them is between point C and joint of middle arm cylinder, second is between middle arm cylinder and joint of end arm cylinder and the other is between joint of end arm cylinder and point D.

#### 4.2.2.1. Stress Analysis of Section-1

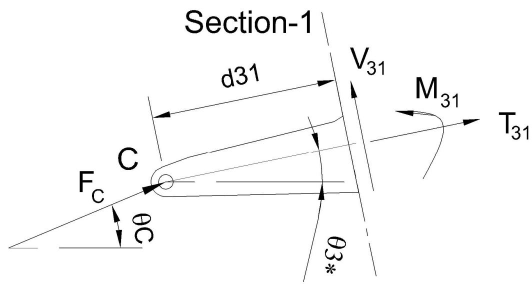



Figure 4.17: Section-1 on middle arm

Section-1 is taken between point C and joint of cylinder of middle arm.

$$T_{31} = -F_C \cdot \cos\left(\theta_C - \theta_3^{\zeta}\right)$$

$$V_{31} = -F_C \cdot \sin\left(\theta_C - \theta_3^{\zeta}\right)$$

$$M_{31} = F_C \cdot \sin\left(\theta_C - \theta_3^{\zeta}\right) \cdot d_{31}$$

$$\sigma_{upper31} = -\frac{M_{31}}{Z_{upper31}} + \frac{T_{31}}{A_{s31}} < \frac{\sigma}{S}$$

$$\sigma_{lower31} = \frac{M_{31}}{Z_{lower31}} + \frac{T_{31}}{A_{s31}} < \frac{\sigma}{S}$$

### 4.2.2.2. Stress Analysis Of Section-2

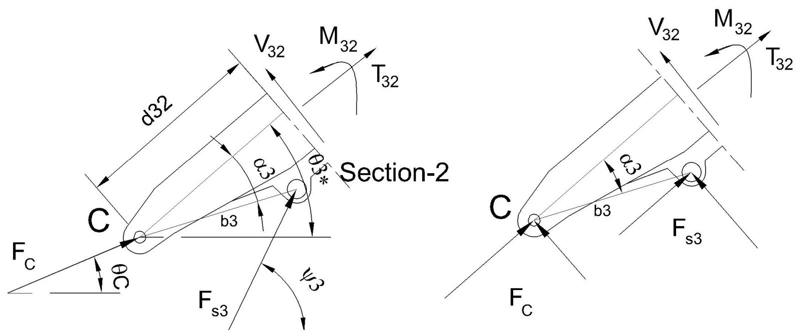



Figure 4.18: Section-2 on middle arm

Section-2 is taken between joint of cylinder of end arm and joint of cylinder of middle arm.

$$T_{32} = -F_C \cdot \cos\left(\theta_C - \theta_3^{\zeta}\right) - F_{s3} \cdot \cos\left(\psi_3 - \theta_3^{\zeta}\right)$$

$$V_{32} = -F_C \cdot \sin\left(\theta_C - \theta_3^{\zeta}\right) - F_{s3} \cdot \sin\left(\psi_3 - \theta_3^{\zeta}\right)$$

$$M_{32} = F_C \cdot \sin\left(\theta_C - \theta_3^{\zeta}\right) \cdot d_{32} + F_{s3} \cdot \sin\left(\psi_3 - \theta_3^{\zeta}\right) \cdot \left(d_{32} - b_3 \cdot \cos\alpha_3\right) - F_{s3} \cdot \cos\left(\psi_3 - \theta_3^{\zeta}\right) \cdot \left(b_3 \cdot \sin\alpha_3\right)$$

$$\sigma_{upper32} = -\frac{M_{32}}{Z_{upper32}} + \frac{T_{32}}{A_{s32}} < \frac{\sigma}{S}$$

$$\sigma_{lower32} = \frac{M_{32}}{Z_{lower32}} + \frac{T_{32}}{A_{s32}} < \frac{\sigma}{S}$$

#### 4.2.2.3. Stress Analysis Of Section-3

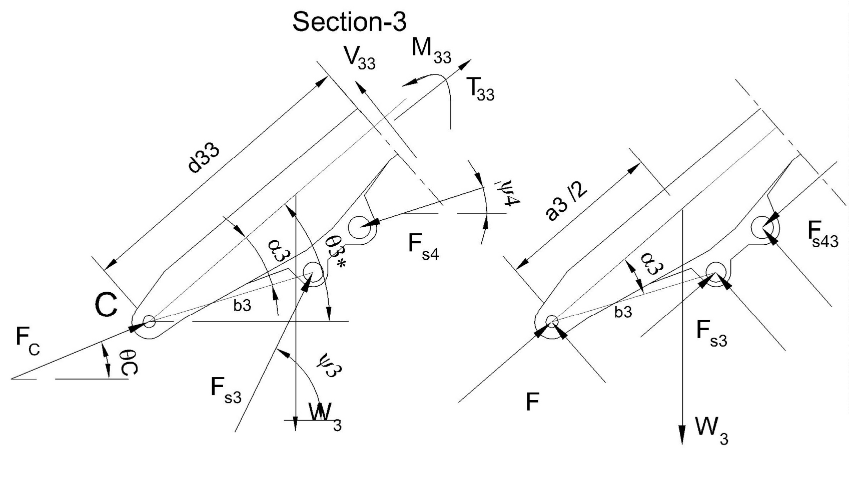



Figure 4.19: Section-3 on middle arm

Section-3 is taken between joint of cylinder of end arm and point D.

$$F_{s43} = \frac{W_4 \cdot \left(\frac{x_5 - X_4}{2}\right) + W_{tool} \cdot (x_5 + a_5 - x_4) + F_{tool3}{}^{x} \cdot (y_5 - y_4) - F_{tool3}{}^{y} \cdot (x_5 + a_5 - x_4)}{b_4 \cdot \cos\left(\theta_4{}^{\zeta} - \alpha_4\right) \cdot \sin\psi_3 + b_4 \cdot \sin\left(\theta_4{}^{\zeta} - \alpha_4\right) \cdot \cos\psi_3}$$

$$T_{33} = -F_C \cdot \cos\left(\theta_C - \theta_3{}^{\zeta}\right) - F_{s3} \cdot \cos\left(\psi_3 - \theta_3{}^{\zeta}\right) + F_{s43} \cdot \cos\left(\psi_4 - \theta_3{}^{\zeta}\right) + W_3 \cdot \cos\left(\frac{\pi}{2} - \theta_3{}^{\zeta}\right)$$

$$V_{33} = -F_C \cdot \sin\left(\theta_C - \theta_3{}^{\zeta}\right) - F_{s3} \cdot \sin\left(\psi_3 - \theta_3{}^{\zeta}\right) + F_{s43} \cdot \sin\left(\psi_4 - \theta_3{}^{\zeta}\right) + W_3 \cdot \sin\left(\frac{\pi}{2} - \theta_3{}^{\zeta}\right)$$

$$M_{33} = F_C \cdot \sin\left(\theta_C - \theta_3{}^{\zeta}\right) \cdot d_{33} + F_{s3} \cdot \sin\left(\psi_3 - \theta_3{}^{\zeta}\right) \cdot \left(d_{33} - b_3 \cdot \cos\alpha_3\right) - F_{s3} \cdot \cos\left(\psi_3 - \theta_3{}^{\zeta}\right) \cdot \left(b_3 \cdot \sin\alpha_3\right)$$
$$-F_{s43} \cdot \sin\left(\psi_4 - \theta_3{}^{\zeta}\right) \cdot \left(d_{33} + c_3 \cdot \cos\beta_3 - a_3\right) + F_{s43} \cdot \cos\left(\psi_4 - \theta_3{}^{\zeta}\right) \cdot \left(c_3 \cdot \sin\beta_3\right) - W_3 \cdot \sin\left(\frac{\pi}{2} - \theta_3{}^{\zeta}\right) \cdot \left(d_{33} - \frac{a_3}{2}\right)$$

$$\sigma_{upper33} = -\frac{M_{33}}{Z_{upper33}} + \frac{T_{33}}{A_{s33}} < \frac{\sigma}{S}$$

$$\sigma_{lower33} = \frac{M_{33}}{Z_{lower33}} + \frac{T_{33}}{A_{s33}} < \frac{\sigma}{S}$$

### 4.2.3. Stress Analysis of End Arm

There are two sections taken on end arm for stress analysis. One of them is between point D and joint of end arm cylinder and the other is between joint of end arm cylinder and point E.

#### 4.2.3.1. Stress Analysis On Section-1

Section-1 is taken between point D and joint of end arm cylinder.

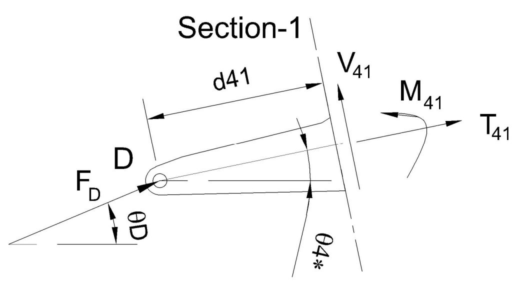



Figure 4.20: Section-1 on end arm

$$T_{41} = -F_D \cdot \cos\left(\theta_D - \theta_4^{\zeta}\right)$$

$$V_{41} = -F_D \cdot \sin\left(\theta_D - \theta_4^{\zeta}\right)$$

$$M_{41} = F_D \cdot \sin\left(\theta_D - \theta_4^{\zeta}\right) \cdot d_{41}$$

$$\sigma_{upper41} = -\frac{M_{41}}{Z_{upper41}} + \frac{T_{41}}{A_{s41}} < \frac{\sigma}{S}$$

$$\sigma_{lower41} = \frac{M_{41}}{Z_{lower41}} + \frac{T_{41}}{A_{s41}} < \frac{\sigma}{S}$$

### 4.2.3.2. Stress Analysis On Section-2

Section-2 is taken between point E and joint of cylinder.

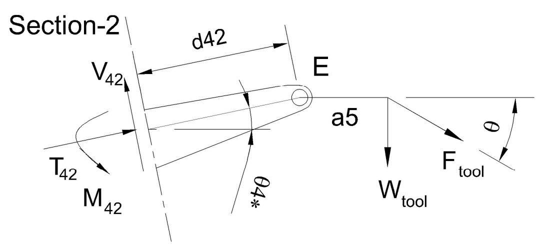



Figure 4.21: Section-2 on end arm

$$T_{42} = -F_{tool4}\cdot\cos\left(\theta_4^{\zeta} - \theta_{tool4}\right) + W_{tool}\cdot\cos\left(\frac{\pi}{2} - \theta_4^{\zeta}\right)$$

$$V_{42} = F_{tool4}\cdot\sin\left(\theta_4^{\zeta} - \theta_{tool4}\right) - W_{tool}\cdot\sin\left(\frac{\pi}{2} - \theta_4^{\zeta}\right)$$

$$M_{42} = \left(-F_{tool4}\sin\left(\theta_4^{\zeta} - \theta_{tool4}\right) + W_{tool}\cdot\sin\left(\frac{\pi}{2} - \theta_4^{\zeta}\right)\right)\cdot\left(d_{42} + a_5\cdot\cos\theta_4^{\zeta}\right)$$

$$\sigma_{upper42} = \frac{-M_{42}}{Z_{upper42}} + \frac{T_{42}}{A_{s42}} < \frac{\sigma}{S}$$

$$\sigma_{lower42} = \frac{M_{44}}{Z_{lower42}} + \frac{T_{42}}{A_{s42}} < \frac{\sigma}{S}$$

## 4.3. Design the Model of the Demolition Boom

The results of two approximations are compared and results of 2nd approximation gave higher stress values for this study. The stability of the excavator is recalculated at this stage using the force and weights (Chapter 3.5.1.) get from 2nd approximation. The reaction force acting on the tool is reduced up to the excavator tipping limit where the maximum counter weight was determined. This new reaction force get from the stability calculation and the boom weights get from the demolition boom model will be used for Finite Element Analysis.

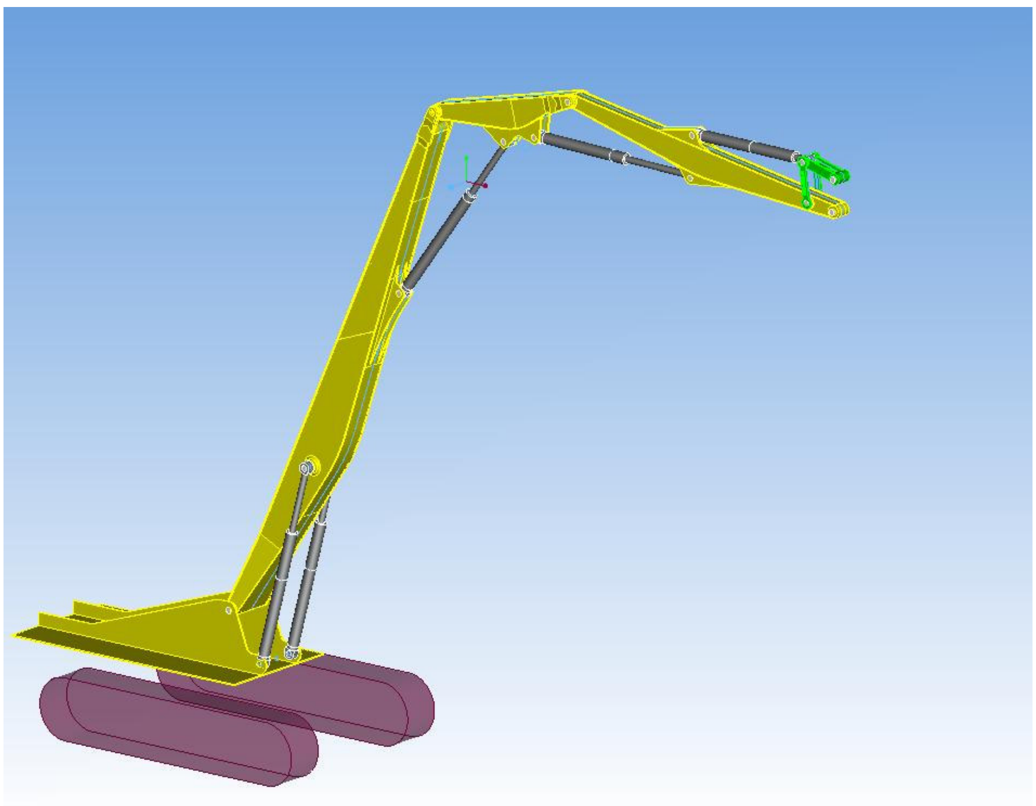


Figure 4.22: The model designed in Pro-Engineer

A structure is modeled in a parametric design program which is named Pro-Engineer by using the parameter values. The parameters mentioned here are the dimensions needed to get the mechanism of the demolition excavator. The other dimensions needed to model the booms are lengths, widths and thickness of sheet metals. Some of them are taken from the structural design results and the others are selected by evaluating standard excavator and material handler excavator which are manufactured by the thesis study supporter company. This is an approach to decide on the initial dimensions.

The parameters and the initial dimensions of the model have been changed and the stress calculations have been done until get all calculated $\sigma_{lower}$ and $\sigma_{upper}$ values are lower than $\sigma_y/S$. The final model will be used for Finite Element Analysis.

# CHAPTER 5

# FINITE ELEMENT ANALYSIS

## 5.1. Introduction to Finite Element Analysis

The finite element method (FEM) is a solution method for partial differential equations by computer. Finite Element Analysis is a technique by which the stresses, deflections and reaction forces etc in an object can be estimated using finite element method. This technique is based on dividing the object into small elements which are connected by nodes. The force and stress etc are calculated for each element using the neighbor elements and boundary conditions. The results of each element are put together to estimate stresses and deflections of the entire object.

Deformations can only be obtained at the node points therefore the choice of the number of nodes is important. Increasing the number of nodes means increasing the number of elements and usually increasing the accuracy of the results. Nodes define admissible degree of freedom. Each node has six degrees of freedom: X, Y and Z translation, and X, Y and Z rotation. Therefore a general 3D model has the total number of freedom of six times the number of nodes.

The demolition booms will be analyzed by static analysis. Static analysis is used to determine the displacements, stresses, strains and forces in structures or components caused by loads that do not induce significant inertia and damping effects. Steady loading and response conditions are assumed; that is, the loads and structure's response are assumed to vary slowly with respect to time. The kinds of loading that can be applied in a static analysis include: [12], [3]

- Externally applied forces and pressures
- Steady-state inertial forces (such as gravity or rotational velocity)
- Imposed (nonzero) displacements
- Temperatures (for thermal strain)

The steps of Finite Element Analysis:

1. Create or import the geometry,
2. Cleanup of geometry,
3. Specify material and element properties,
4. Mesh the geometry into nodes and elements,
5. Apply the loads and boundary conditions,
6. Solution,
7. Check the results,
8. If the results are not satisfactory; change the model with respect to the FEA results and analyze the new model.

### 5.2. Assumptions and Boundary Conditions

It is assumed that material behavior is linear elastic, the strains and the displacements are small. Thus, linear elastic analysis will be carried out. The kinds of loading that will be applied in the static demolition boom system analysis are externally applied forces and steady-state inertial forces (gravity). The loads are applied statically.

The boundary conditions are conditions given in Chapter 4 (Structural Design): The critical analysis position is maximum front reach position. The model is fixed at points A and B in all directions except rotation about Z-axis. The steady state inertial forces acting on the booms are weights of each boom and weight of the tool. The unique external force is reaction force acting on the tool ($F_{tool}$). (Figure 5.1)

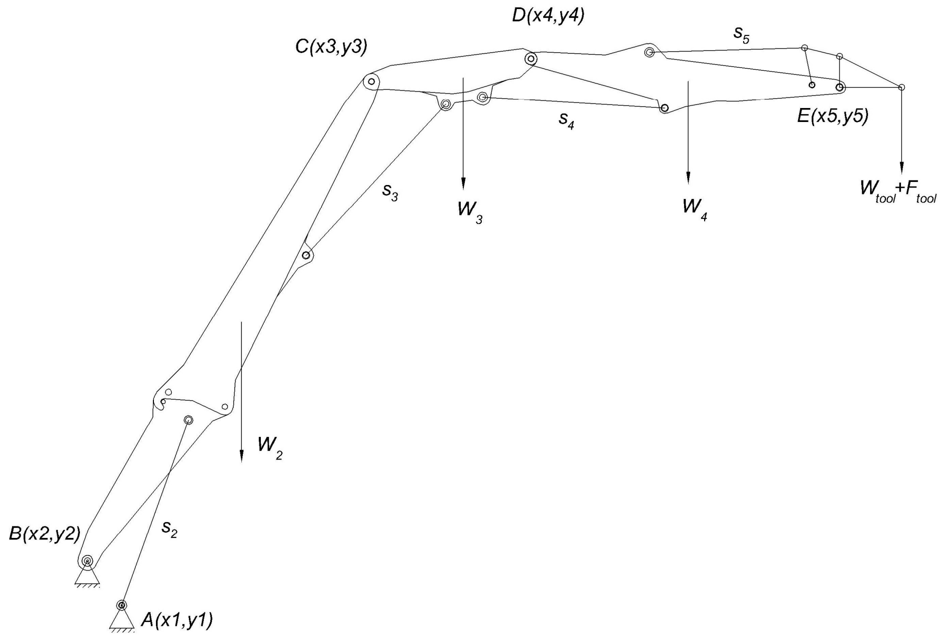



Figure 5.1: Finite element analysis position

## 5.3. Finite Element Analysis

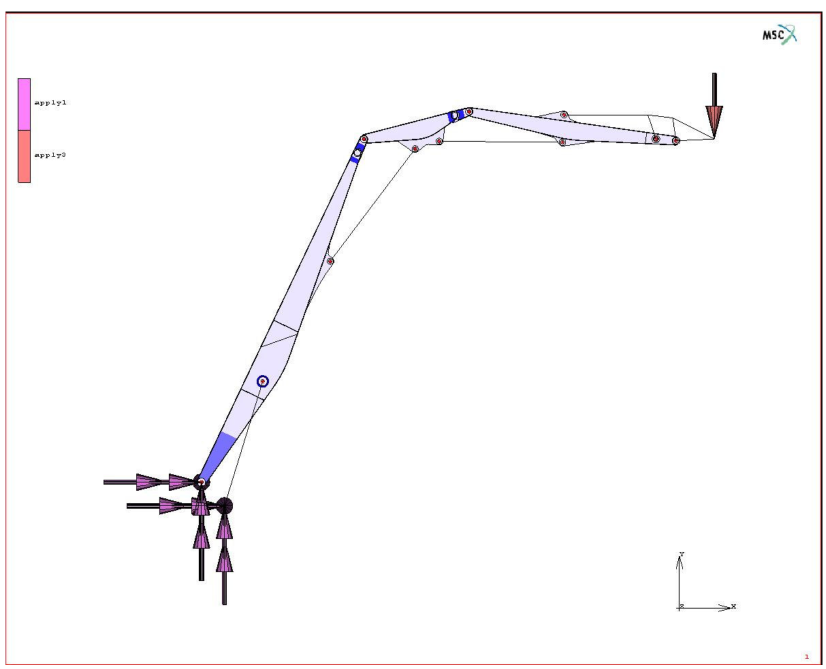



Figure 5.2: Finite element analysis model

Algorithm of the analysis:

1. A copy of the solid model designed in Pro-Engineer has been exported in IGES format, because the FEA program MSC.Marc.Mentat can not import a Pro-Engineer data directly. The IGES data is imported in MSC.Marc. The imported models are boom, middle-arm and end-arm.
2. The booms are positioned to maintain the maximum reach position. To do this the proper cylinder lengths ($s_2$, $s_3$ and $s_4$) are used which are taken from the results of the chapter 3 (Mechanism design).
3. The model is meshed by using linear tetra (four nodes) solid elements. The average element size is taken 40mmx40mmx40mm.
4. FEM analysis links are created. This links are elements to equalize the selected degrees of freedom of connected nodes. Various types of these links are used to simulate the pin joints between the booms.
5. Cylinders are simulated with line elements. These line elements are defined as cylindrical rigid beams and the hydraulic leakage in cylinders are neglected. The real cross-sections are defined in these elements.
6. The material is St52. The properties of St52 are taken as: Modulus of Elasticity E=210GPa, Poisons Ratio $\nu$=0,3 and the density $\rho=7.85\text{x}10^{-9}$ton/mm$^2$. The density is required to the weight definition of the components.
7. The boundary conditions which are the forces and the constraints are applied on the model.
8. The analysis is solved.
9. The Von-Misses results are taken in colored views for the ease of the discussions on the results. This views show the stress distribution of the elements.
10. The results are discussed and the model is revised. The revised model is reanalyzed with finite element method.

## 5.4. The Finite Element Analysis Results

The results of the first application (Results-1) are given in Figure 5.3 to Figure 5.8.


Figure 5.3: General view-1 of the Results-1


Figure 5.4: General view-2 of the Results-1


Figure 5.5: View of the boom of Results-1


Figure 5.6: View of the middle-arm of Results-1

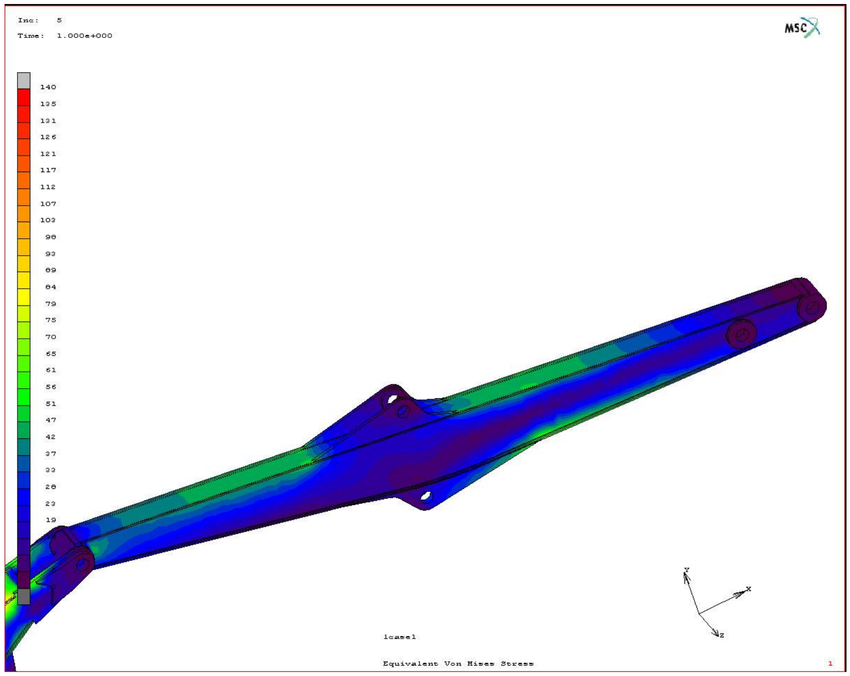



Figure 5.7: View of the end-arm of Results-1

It is find out that the stresses are low in Results-1. In order to revise the model some dimensions are changed and the material thicknesses are decreased at some regions. The following figures are the results of final design.

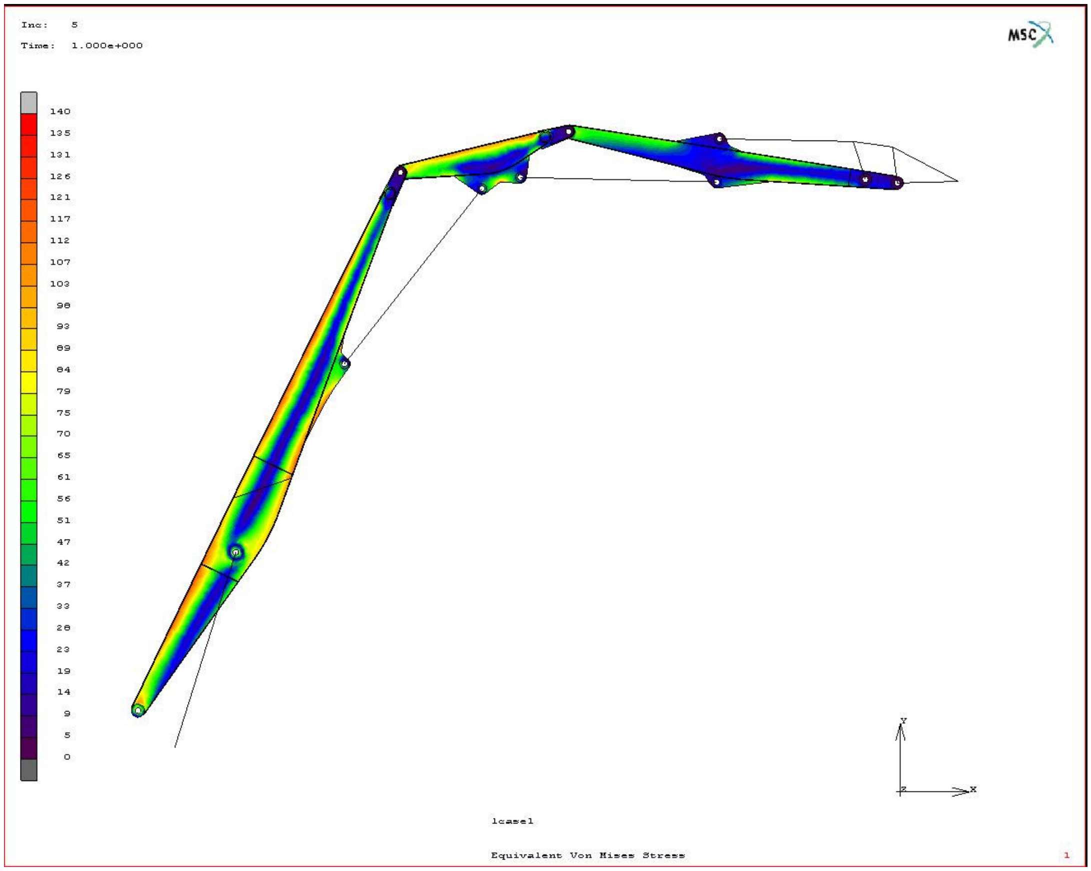



Figure 5.8: General view-1 of the final design results


Figure 5.9: General view-2 of the final design results

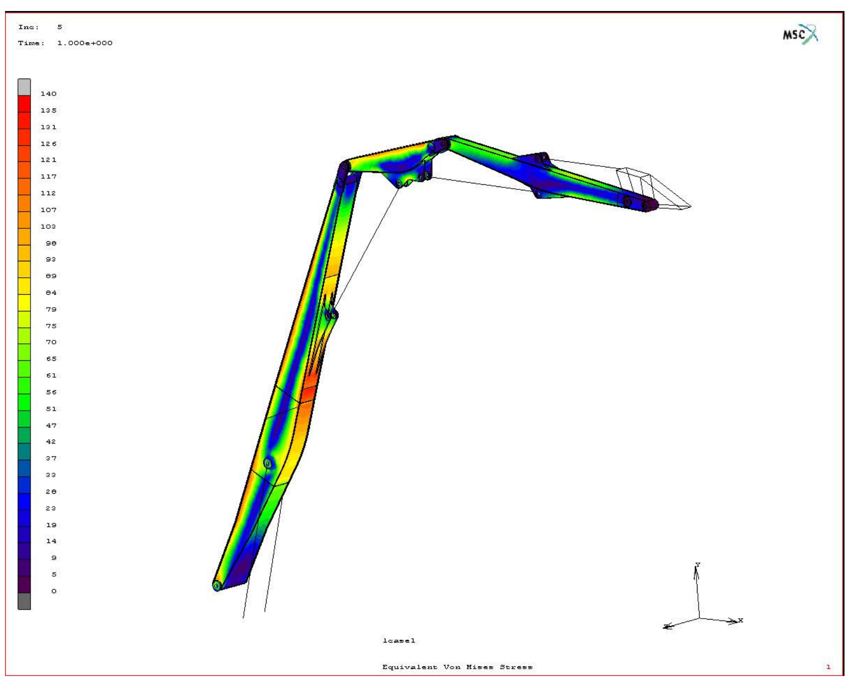



Figure 5.10: General view-3 of the final design results


Figure 5.11: View-1 of the boom of the final design results

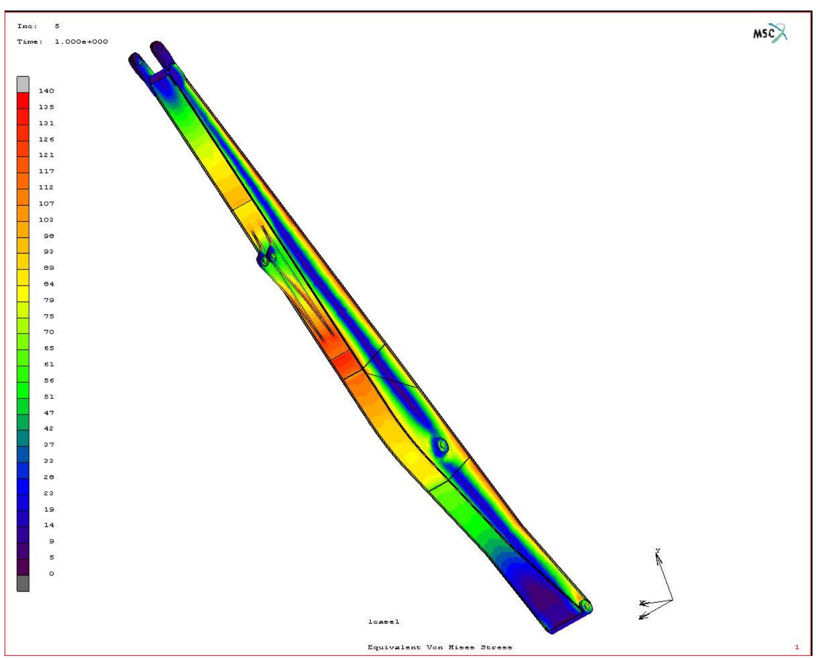



Figure 5.12: View-2 of the boom of the final design results

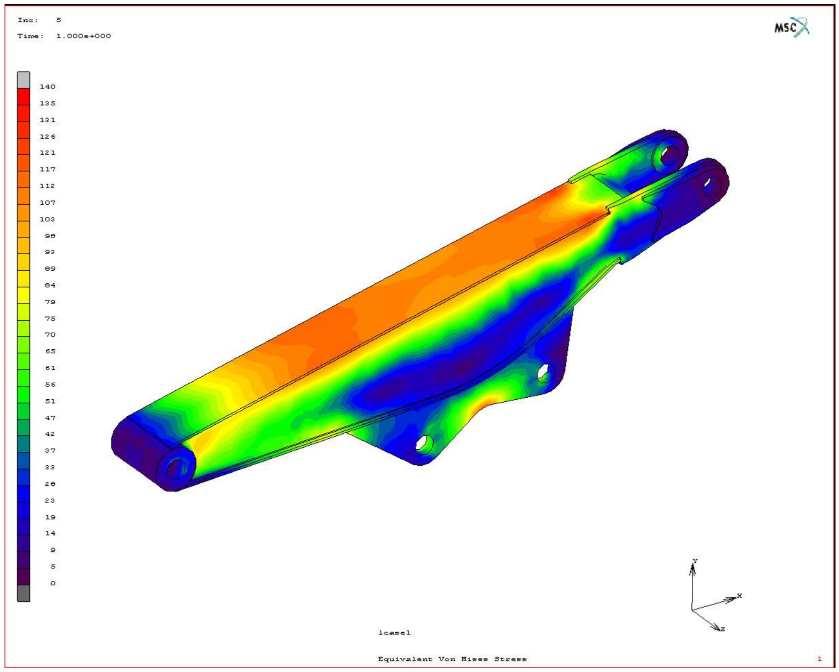



Figure 5.13: View-1 of the middle arm of the final design results

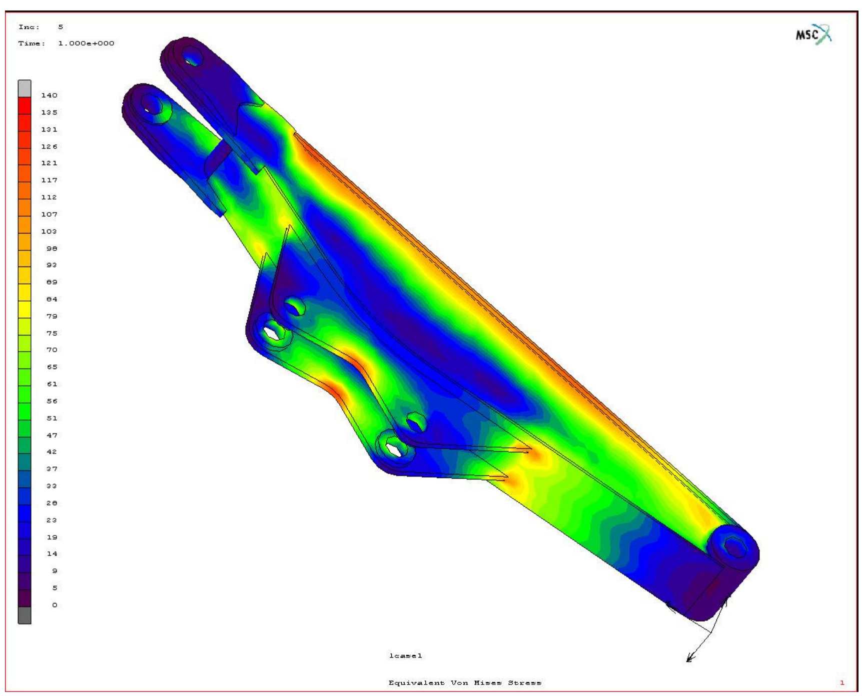



Figure 5.14: View-2 of the middle arm of the final design results

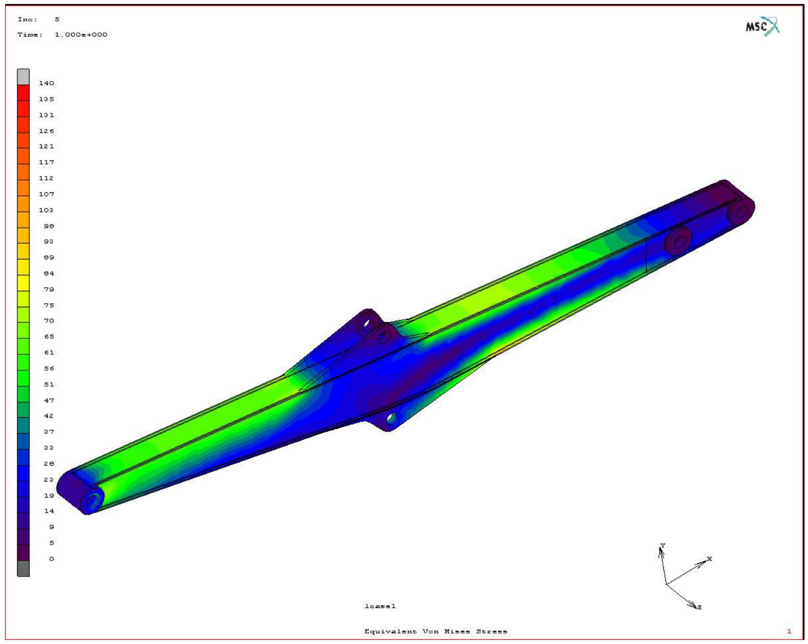



Figure 5.15: View-1 of the end arm of the final design results

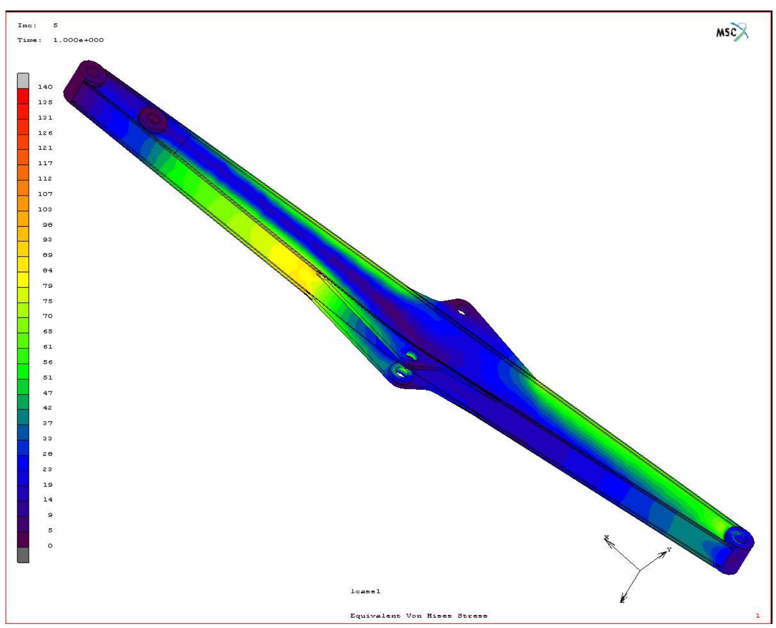



Figure 5.16: View-2 of the end arm of the final design results

# CHAPTER 6

# CASE STUDY

The aim of this thesis study is to design a demolition boom for HMK300LC which is a standard hydraulic excavator manufactured by HIDROMEK Ltd. Şti. HMK300LC is an excavator with 30,500 ton operation weight used for earth moving applications. Working range of the standard HMK300LC is given in Figure 6.1, its maximum working height is 10.4m. The standard excavator can be used for the demolition of a 3 floored building which is nearly 10m height. The same excavator will be redesigned to demolish higher buildings. Properties of the demolition excavators are given in Chapter 2.

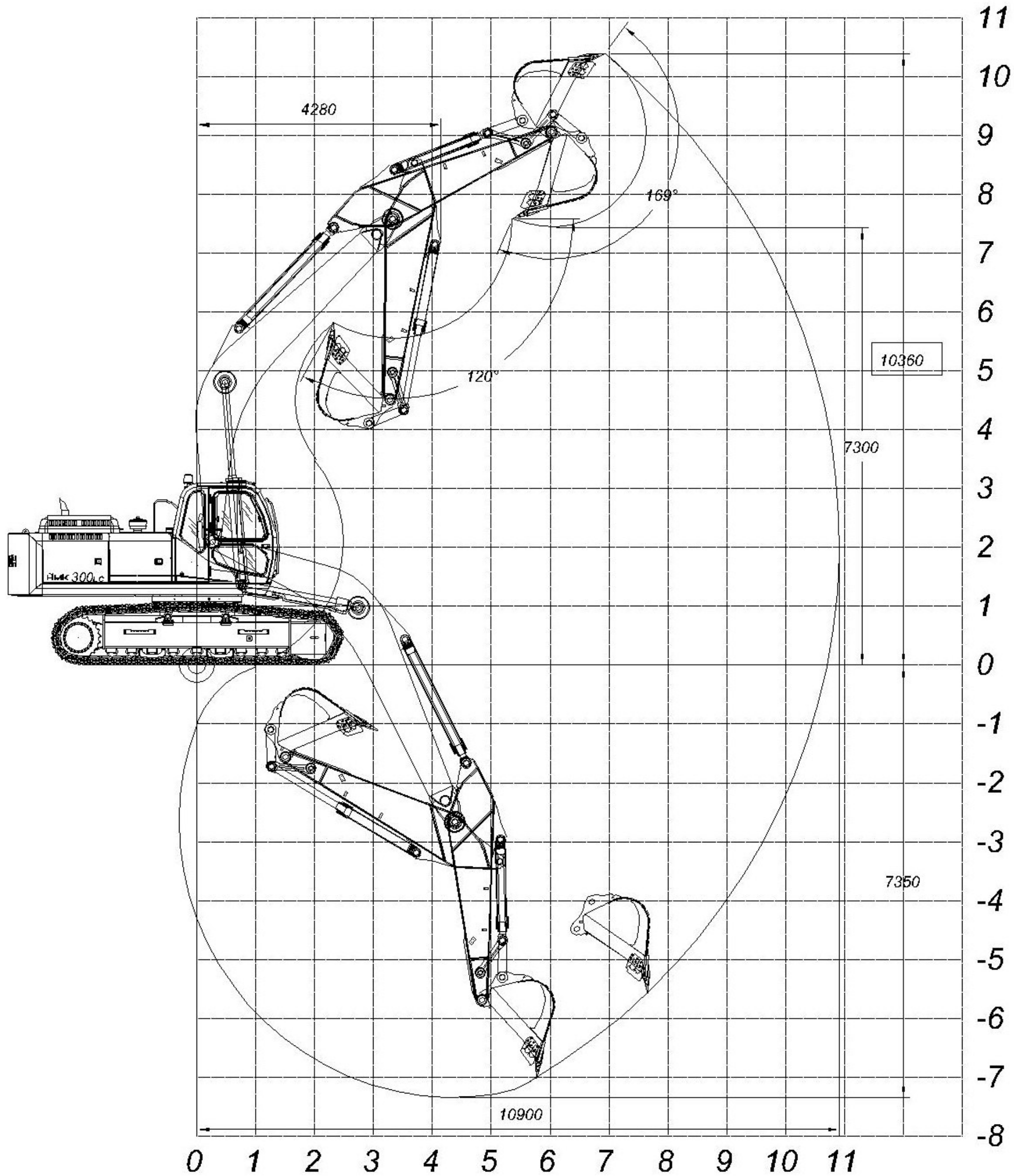



Figure 6.1: Working Range of HMK300LC

### 6.1. Mechanism Design

The design criteria's and the limitations of the demolition boom are given in Chapter 3. One of the design criteria's is the marketability. A marketable product must have same or better properties than the products sold in the market. By comparing the data's of different demolition excavator manufacturers and demolition boom manufacturers in the market, it is assumed to design a demolition excavator with maximum pin height between 17m and 18m and maximum front reach between 10m and 12m for a permission of maximum 2,2 ton weighted tool (Table 6.1). The relevant information's of demolition excavators of different manufacturers are given in Appendix C.

Table 6.1: Comparison of demolition excavators of different manufacturers

| | **Company** | **Model** | **Max Pin Height (A)** | **Max Front Reach (B)** | **Tool Weight (ton)** | **Operation Weight (ton)** |
|---|---|---|---|---|---|---|
| Excavator manufacturers | DAEWOO[13] | 300LC-V | 18 | 12 | 2-2,6 | 34,650ton + Wtool |
| | HITACHI [14] | 240LCK | 16 | 11,25 | 2,1 | 26,900ton + Wtool |
| | CASE [15] | CX290 | 17 | 10 | 2,2 | no information |
| | Komatsu [16] | PC 300LC-7 | 20 | 11 | 2,3 | 39,800ton |
| Demolition boom manufacturers | SNS [17] | 27-32ton | 18 | 11 | 2,2 | no information |
| | Boforce [18] | 27ton | 18 | 10 | 2,5 | no information |
| | Boforce [18] | 31ton | 17 | 10 | 2,2 | no information |
| | Kocurek [19] | 29-30ton | 17 | 10 | 2,2 | no information |
| **An Assumption For HMK300LC** | | | **17-18** | **10-12** | **2,2** | |

The mechanism of the demolition boom is designed by changing the design parameters and checking the design limitations which are explained in Chapter 3. In consequence a mechanism with a maximum pin height of 18m and maximum pin height of 11.6m is designed (Figure 6.2).

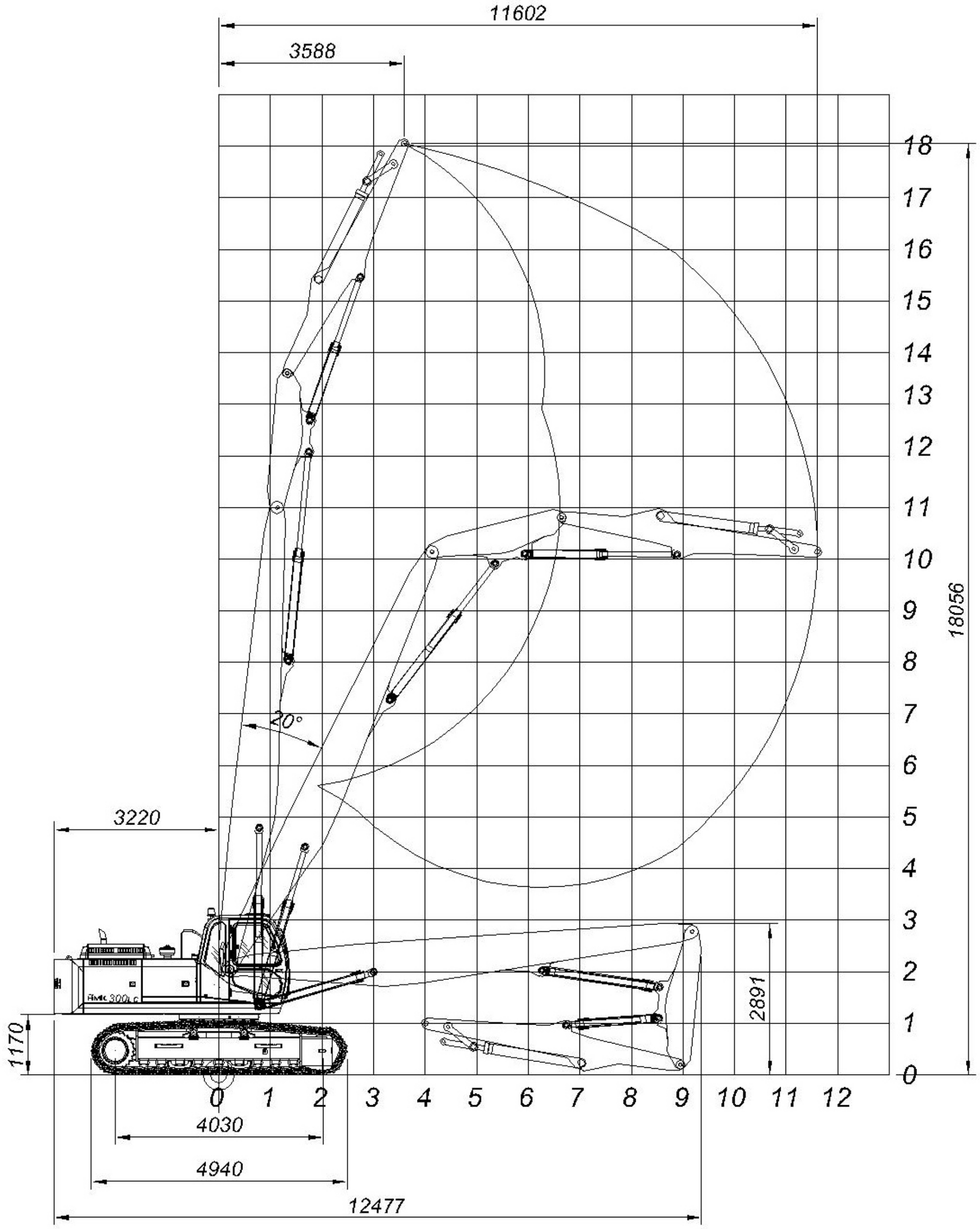



Figure 6.2: Working range of HMK300LC-demolition

## 6.2. Structural Design

Firstly, the forces acting on joints are calculated. The forces created in the cylinders are related with cylinder inner rod areas which are act as barriers to the hydraulic flow. Therefore decision on the cylinder dimensions is important. At the beginning a cylinder cross-section is estimated by taking care of the selection criteria's of cylinders of material handler and standard HMK300LC. All cylinders have the same cross-section dimensions. But it is also important what reaction force the cylinders can resist to. The direction of the reaction force ($F_{tool}$) acting on the tool is taken downward and the maximum $F_{tool}$ values which will be resisted by each cylinder are calculated. The minimum value of these $F_{tool}$ values was at the boom cylinder ($F_{tool2}$) which is limited with anti-shock pressure. This means all cylinders can carry out the $F_{tool2}$. The middle-arm, end-arm and tool cylinder cross-sections are reduced to result all $F_{tool}$ values equal to $F_{tool2}$ or little more than $F_{tool2}$. The process of reducing the cylinder dimensions will reduce the cost of the cylinders. It is seen that $F_{tool}$ is not very large or very less than the weight of the tool ($F_{tool}=W_{tool}\pm10000N$) so the cylinder selections can be assumed as accurate.

The forces acting on each joint of the mechanism is calculated for an external force of $F_{tool}$ and internal forces of $W_2$, $W_3$, $W_4$ and $W_{tool}$. The stresses of the booms will be calculated for these forces at maximum front reach position of the booms. Because the affect of these forces (moments) acting on the excavator body will be maximum at the front reach position. The next step is to define the dimensions of box type cross-sections taken on the booms at different locations. These dimensions are the material thicknesses and the width of this material used at each position. These cross-sections are estimated taking into consideration of past applications of HIDROMEK products. Then the stresses of these cross-sections are calculated as given in Chapter 4. The cross-sections are changed where calculated stresses are greater than $\sigma_y/S$.

A solid model is designed in Pro-Engineer as mentioned in Chapter 4.3. This model is defined parametrically so modifications and changes can be applied easily on the model. The material thicknesses and the boom shapes are defined parametric too, because in most cases the changes are on the material thicknesses and the boom shapes.

The model is analyzed by using a Finite element analysis program MSC.Marc (Chapter 5.3). The model is redesigned and reanalyzed a few times to get a better demolition boom design.

### 6.3. The Other Components of the Demolition Excavator

This thesis includes only the demolition boom design. However some other components are also needed for a complete demolition excavator. One of them is the additional counterweight whose weight is calculated while the shape and the location of the standard counter weight are redesigned (Figure 6.3).

Figure 6.3: An example for the additional counterweight of a demolition excavator

Another component which will be redesigned is the cabin. Because of the high reach property of the demolition boom the ergonomics of a standard cabin will be insufficient. The cabin will be redesigned to give a better field of view for the operator (Figure 6.4). A tilting cabin can be used. Protection maintenance on the cabin is also a need because of the falling scraps which are cut and crushed from the building during the demolition purpose.

Figure 6.4: An example for the cabin of a demolition excavator

The transportation of the excavator is an other issue. A transportation support will be required for the demolition boom. This support will be mounted on the demolition excavator at the transportation position to fix the booms at this position during the transportation. (Figure 6.5)

Figure 6.5: An example for the transportation support

# CHAPTER 7

# DISCUSSIONS AND CONCLUSIONS

In this study design of a demolition boom for HMK300LC is performed. HMK300LC is an excavator produced by HİDROMEK. Parametric design approach is used so that the work done can be easily adapted for different working ranges. This design approach gives rapid and right results. The results obtained are comparable with the existing demolition booms of the competitive manufacturers.

As mentioned in the previous chapters, different booms are designed to use the excavators more effectively. For example mining excavators have a shorter boom and a shorter arm with respect to the standard excavator. The digging action is applied upwards owing to the upwards assembled bucket. Shorter boom and arm cause a smaller working area which is not a problem for a mining excavator. But this gives a big advantage in increasing the digging force. Another example is the material handler which has a longer boom and a longer arm.

As a future work the first boom can be designed for multi purpose use. In such a design the boom is made up of two pieces, where the two pieces are joined to each other rigidly. One piece is common for all use while there are three or more different second pieces. The operator changes this second piece according to the work to be performed. A quick coupling system can be included for rapid change of the second piece.

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## APPENDIX A

## FORMULATION OF OUT REACH DRAWING

As mentioned in chapter 3.4. "Out reach is the curve which is formed by connecting the furthermost points of the pin; E($x_5$,$y_5$)."

The formulas needed to calculate the point E($x_5$,$y_5$) are given below. To comprehend the formulas see the Figure 3.3 and Figure 3.4.

### a) Formulas of the Boom Position

Cartesian coordinates of point A are calculated using polar coordinates of A with respect to point B ($x_2$,$y_2$) which is given as an input.

$$x_1 = x_2 + a_1 \cdot \cos\alpha_1$$

$$y_1 = y_2 - a_1 \cdot \sin\alpha_1$$

$\theta_2$ is a dimension which specifies the boom position for a given boom cylinder length ($s_2$).

$$\theta_2 = \cos^{-1}\left(\frac{a_1^2 + b_2^2 - s_2^2}{2 \cdot a_1 \cdot b_2}\right) + \alpha_2$$

$$\theta_2^{\xi} = \theta_2 - \alpha_1$$

### b) Formulas of the Middle Arm Position

Coordinates of point C($x_3$,$y_3$) are defined using coordinates of point B and angular position of the boom.

$C(x_3,y_3)$

$$x_3 = x_2 + a_2 \cdot \cos\theta_2^{\xi}$$
$$y_3 = y_2 + a_2 \cdot \sin\theta_2^{\xi}$$

$\theta_3$ is a dimension which specifies the middle arm position for a given middle arm cylinder length ($s_3$).

$$\theta_3 = \cos^{-1}\left(\frac{c_2^2 + b_3^2 - s_3^2}{2 \cdot c_2 \cdot b_3}\right) + \alpha_3 + \beta_3$$
$$\theta_3^{\xi} = \theta_3 - \left(\pi - \theta_2^{\xi}\right)$$

### c) Formulas of the End Arm Position

Coordinates of point $D(x_4,y_4)$ are defined using coordinates of point C and angular position of the middle arm.

$$x_4 = x_3 + a_3 \cdot \cos\theta_3^{\xi}$$
$$y_4 = y_3 + a_3 \cdot \sin\theta_3^{\xi}$$

$\theta_4$ is a dimension which specifies the end arm position for a given end arm cylinder length (s4).

$$\theta_4 = \cos^{-1}\left(\frac{c_3^2 + b_4^2 - s_3^2}{2 \cdot c_3 \cdot b_4}\right) + \alpha_4 + \beta_3$$
$$\theta_4^{\xi} = \theta_4 - \left(\pi - \theta_3^{\xi}\right)$$

Coordinates of point $E(x_5,y_5)$ are defined using coordinates of point D and angular position of the end arm.

$$x_5 = x_4 + a_4 \cdot \cos\theta_4^{\xi}$$
$$y_5 = y_4 + a_4 \cdot \sin\theta_4^{\xi}$$

**APPENDIX B**

**STESS CALCULATION**

The box type cross section of boom, middle arm and end is given in the Figure B.1.

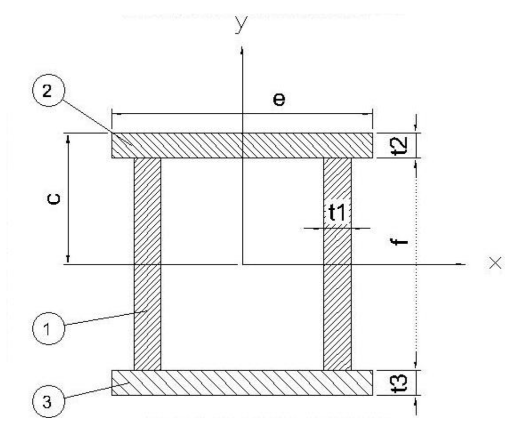



Figure B.1: Cross section of the demolition boom

The formulas below are formed by taking care of references [20] and [21]. Areas of cross sections of sheets:

$A_1 = f \cdot t_1$

$A_2 = e \cdot t_2$

$A_3 = e \cdot t_3$

Calculation of the moment of inertia:

$$c = \frac{2 \cdot A_1 \cdot \left(\frac{f}{2} + t_2\right) + A_2 \cdot \frac{t_2}{2} + A_3 \cdot \left(t_2 + f + \frac{t_3}{2}\right)}{2 \cdot A_1 + A_2 + A_3}$$

$$I_{1x} = t_1 \cdot \frac{f^3}{12}$$

$$I_{2x} = e \cdot \frac{t_2^3}{12}$$

$$I_{3x} = e \cdot \frac{t_3^3}{12}$$

$$I_x = 2 \cdot I_{1x} + 2 \cdot A_1 \cdot \left(\frac{f}{2} + t_2 - c\right)^2 + I_{2x} + A_2 \cdot \left(c - \frac{t_2}{2}\right)^2 + I_{3x} + A_3 \cdot \left(t_2 + f + \frac{t_3}{2} - c\right)^2$$

$$Z_{upper} = \frac{I}{c}$$

$$Z_{lower} = \frac{I}{f + t_2 + t_3 - c}$$

$$A_s = 2 \cdot A_1 + A_2 \cdot A_3$$

Stress existing on the cross section area:

$$\sigma_{upper} = \sigma_{bending} + \sigma_{tensile}$$

$$\sigma_{upper} = \frac{-M}{Z_{upper}} + \frac{F}{A_s} \le \frac{\sigma_y}{S}$$

$$\sigma_{lower} = \frac{M}{Z_{lower}} + \frac{F}{A_s} \le \frac{\sigma_y}{S}$$

Where;

t: sheet thickness

d,e: sheet width

A: cross section area of the sheet

I: moment of inertia

$\sigma_{bending}$: bending stress

$\sigma_{tensile}$: tensile stress

$\sigma_{upper}$, $\sigma_{lower}$: calculated stresses

$\sigma_y$ : Yield strength of the material

S: safety factor

**APPENDIX C**

**DEMOLITION EXCAVATORS OF DIFFERENT MANUFACTURERS**


| CX SERIES | Maximum height<br>Altezza massima<br>Hauteur maxi<br>Max. Arbeitshöhe<br>Altura máxima | Maximum reach<br>Sbraccio massimo<br>Portée maxi<br>Max. Reichweite<br>Alcance máx | Max dig depth<br>Profondità max di scavo<br>Profondeur portée maxi<br>Max. Grabtiefe<br>Profundidad máxima | Attachment weight<br>Peso operativo<br>Poids de equipement<br>Gewicht Anbauwerkzeug<br>Peso operativo |
|---|---|---|---|---|
| CX290 Demolition | 17 m | 10 m | - | 2200 kg |
| CX330 Demolition | 21 m | 11 m | - | 2000 kg |
| CX460 Demolition | 26 m | 14 m | - | 2500 kg |
| CX800 Demolition | 40 m | 20 m | - | 2500 kg |
| CX130 Long reach | 13 m | 13 m | 10 m | 210 kg |
| CX210 Long reach | 13 m | 16 m | 12 m | 340 kg |
| CX240 Long reach | 14 m | 18 m | 15 m | 340 kg |
| CX330 Long reach | 15 m | 21 m | 17 m | 370 kg |
| WX210 Rehandling | 13 m | 12 m | 4 m | 1500 kg |
| WX240 Rehandling | 13 m | 12 m | 4 m | 1600 kg |

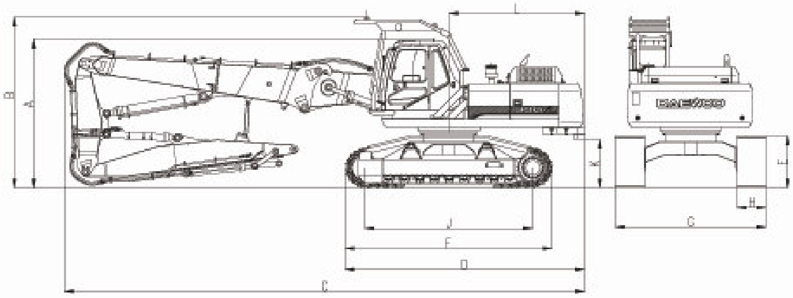
22

Figure C.1: CASE CX demolition series [13]

# Operational data

## Standard configuration 3 pieces: Boom 5,58m

### Dimensions & Specifications




| Dimensions | | | Track Type | | |
|---|---|---|---|---|---|
| | | | Log Loader | Standard LC | Narrow LC |
| Overall Transport Height (without Canopy) | A | mm | 3.553 | 3.276 | 3.276 |
| Overall Transport Height (with Canopy) | B | mm | 4.086 | 3.809 | 3.809 |
| Overall Length | C | mm | 12.402 | 12.402 | 12.402 |
| Overall Length (without Front) | D | mm | 5.712 | 5.712 | 5.712 |
| Track Height | E | mm | 1.244 | 1.044 | 1.044 |
| Track overall Length | F | mm | 4.950 | 4.950 | 4.950 |
| Track overall Width | G | mm | 3.600 | 3.200 | 3.000 |
| Track Shoe Width | H | mm | 700 | 600 | 600 |
| Tumbler Distance | J | mm | 4.010 | 4.010 | 4.010 |
| Upper Structure Ground Clearance | K | mm | 1.170 | 893 | 893 |
| Tail Swing Radius | L | mm | 3.240 | 3.240 | 3.240 |
| **Specifications** | | | | | |
| Operating Weight (without Tool) | | kg | 36.650 | 34.650 | 34.530 |
| Max. Reach at Ground Level (Pin Arm) | | mm | 12.085 | 12.085 | 12.085 |
| Max. Height (Pin Arm) | | mm | 17.877 | 17.600 | 17.600 |
| Working Capacity | | kg | 2.600 | 2.200 | 2.000 |
| Safety Working Angle | | ° | 30 | 30 | 30 |

4 SOLAR 300 LCV DEMOLITION

Figure C.2: DAEWOO Solar 300LCV Demolition- Dimension and specifications [14]

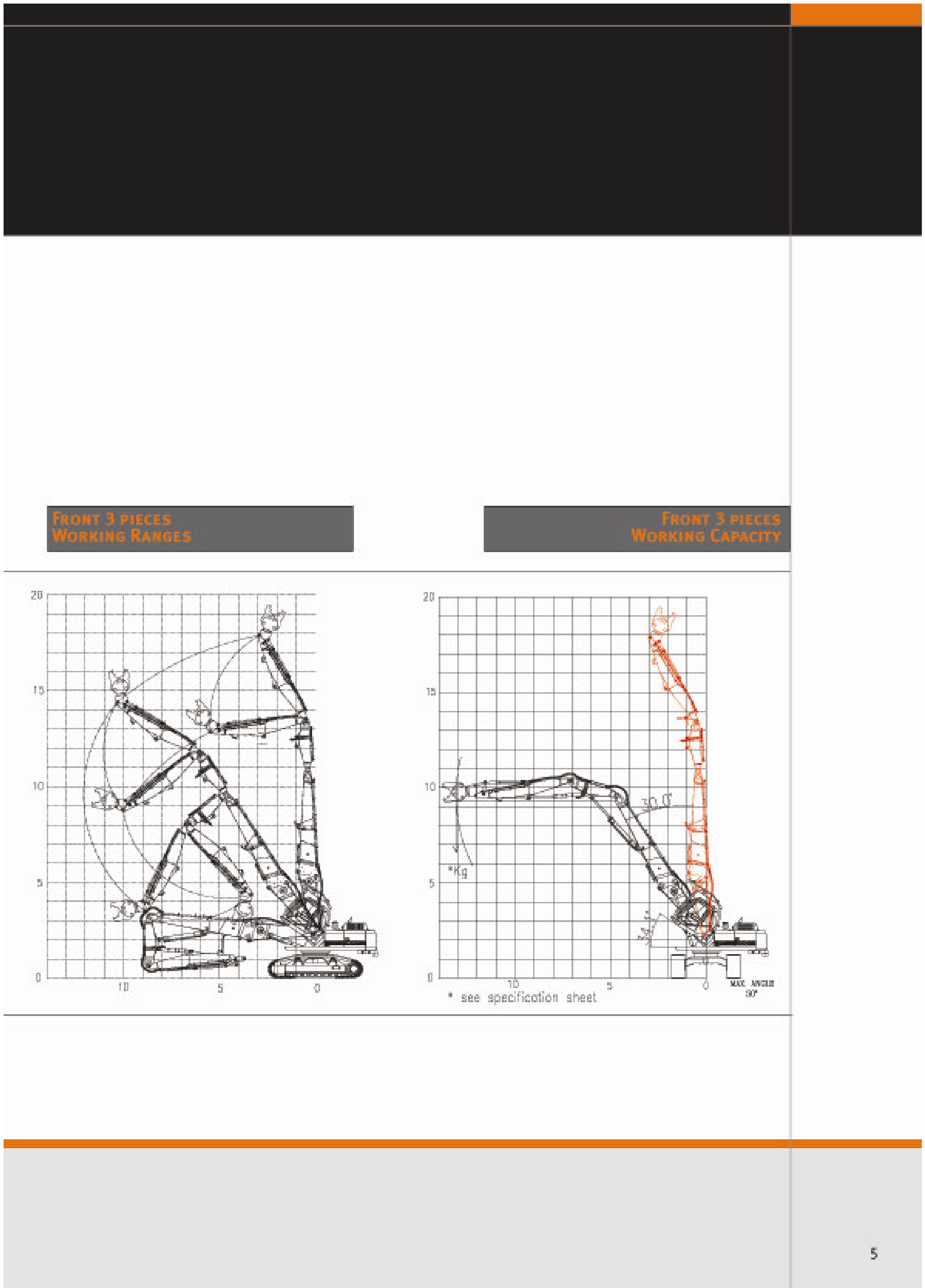



Figure C.3: DAEWOO Solar 300LCV Demolition- Dimension and specifications [14]

PC210LC
PC300LC
PC400LC

Working Gear

# DEMOLITION HIGH REACH

## ■A W ide Lineup

### ●DEMO LITION HIGH REACH

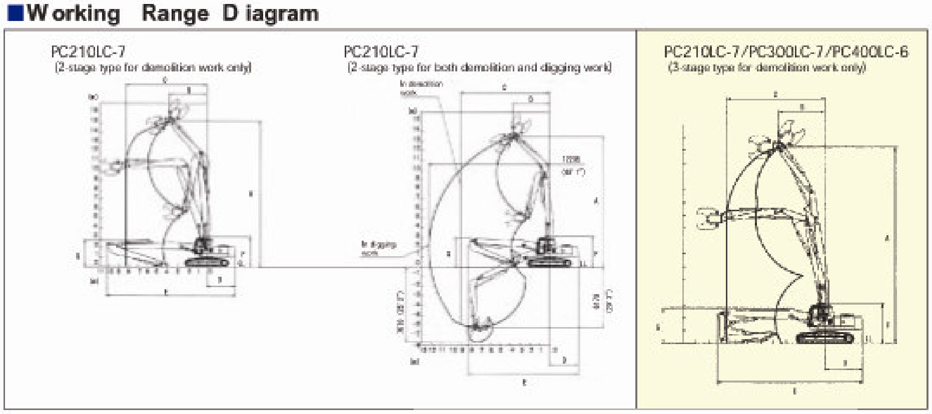
| High reach Type | | 2-Stage | | 3-Stage | | |
|---|---|---|---|---|---|---|
| Boom Type | | 1-Piece | | 1-Piece | 3-Piece | 4-Piece |
| Application | | For demolition work only | For both demolition and digging work | For demolition work only | For demolition work only | for demolition work only |
| Model | PC210LC-7 | O | O | O | | |
| | PC300LC-7 | | | O | O | O |
| | PC400LC-6 | | | O | O | |

Note 1: A circle mark means "Applicable".
Note 2: As to a demolition high reach for PC230 model, consult your Komatsu distributor.


3-stage type with 3-piece boom


2-stage type with 1-piece boom

## ■W orking Range D iagram




## ■Spe cification s

| Item \ Model | | | PC210LC-7 | | PC210LC-7 | PC300LC-7 | | | PC400LC-6 | |
|---|---|---|---|---|---|---|---|---|---|---|
| Structure | High Reach Type | | 2-stage | | 3-stage | | | | | |
| | Boom Type | | 1-piece | 1-piece | 1-piece | 1-piece | 3-piece | 4-piece | 1-piece | 3-piece |
| Item \ Application | | | For Demolition Work Only | For Both Demolition & Digging Work | For Demolition Work Only | For Demolition Work Only | For Demolition Work Only | For Demolition Work Only | For Demolition Work Only | For Demolition Work Only |
| Operating Weight *1 | | kg/lb | 26040/57410 | 25200/55560 | 26480/58380 | 39800/87740 | 41150/90720 | 41730/92000 | 50500/111330 | 52200/115080 |
| Max. Allowable Crusher Weight | | kg/lb | 2300/5070 | 2300/5070 | 2100/4630 | 2300/5070 | 2300/5070 | 2300/5070 | 2300/5070 | 2300/5070 |
| A Max. Working Height *2 | | mm/ft. in | 15200/49' 10" | 13500/44' 3" | 15200/49' 10" | 20000/65' 7" | 20500/67' 3" | 22500/73' 10" | 24000/78' 9" | 25000/82' |
| B Arm Top Pin Radius at Max. Working Height | | mm/ft. in | 3810/12' 6" | 3625/11' 11" | 3600/11' 10" | 4060/13' 4" | 4120/13' 6" | 4430/14' 6" | 4350/14' 3" | 4335/14' 3" |
| C Max. Allowable Working Radius | | mm/ft. in | 8300/27' 3" | 9000/29' 6" | 7600/24' 11" | 11000/36' 1" | 11000/36' 1" | 11000/36' 1" | 12300/40' 4" | 12300/40' 4" |
| D Tail Swing Radius | | mm/ft. in | 2910/9' 7" | 2910/9' 7" | 2910/9' 7" | 3780/12' 5" | 3780/12' 5" | 3780/12' 5" | 3800/12' 6" | 3800/12' 6" |
| E Overall Length *3 | | mm/ft. in | 13210/43' 4" | 11100/36' 5" | 11260/36' 11" | 14220/46' 8" | 14790/48' 6" | 16770/55' | 16610/54' 6" | 17610/57' 9" |
| F Overall Height *3 | | mm/ft. in | 3160/10' 4" | 3160/10' 4" | 3160/10' 4" | 3270/10' 9" | 3270/10' 9" | 3270/10' 9" | 3420/11' 3" | 3420/11' 3" |
| G Height of Folded Work Equipment | | mm/ft. in | 2850/9' 4" | 3000/9' 10" | 2750/9' | 3110/10' 2" | 2900/9' 6" | 2920/9' 7" | 3100/10' 2" | 3170/10' 5" |

• The photos may partially differ from the standard specifications of demolition high reach.
• For precautions when operating the machine, refer to the operation and maintenance manual.

*1 Excluding allowable crusher weight
*2 Arm top pin
*3 When work equipment is folded and lowered to the ground

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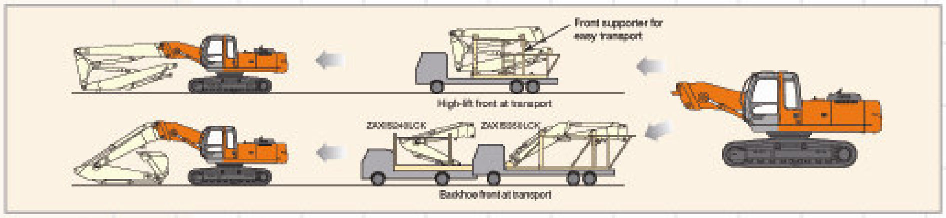
www.komatsu.com/kti/

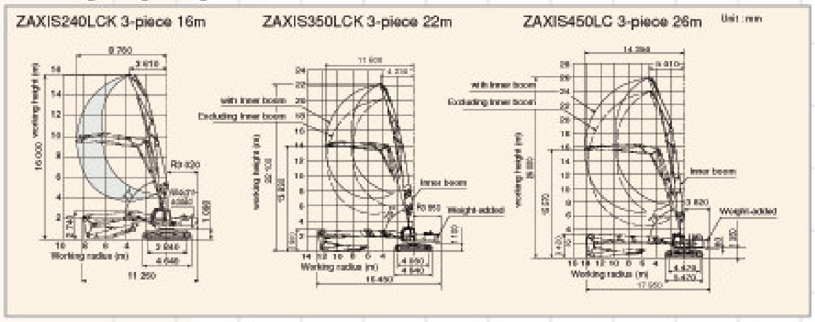
Materials and specifications are subject to change without notice.


Figure C.4: KOMATSU Demolition High reach PC300LC-7 [15]

## Crusher-Backhoe Type (3-piece front)




### Working Range Diagram




### Specifications

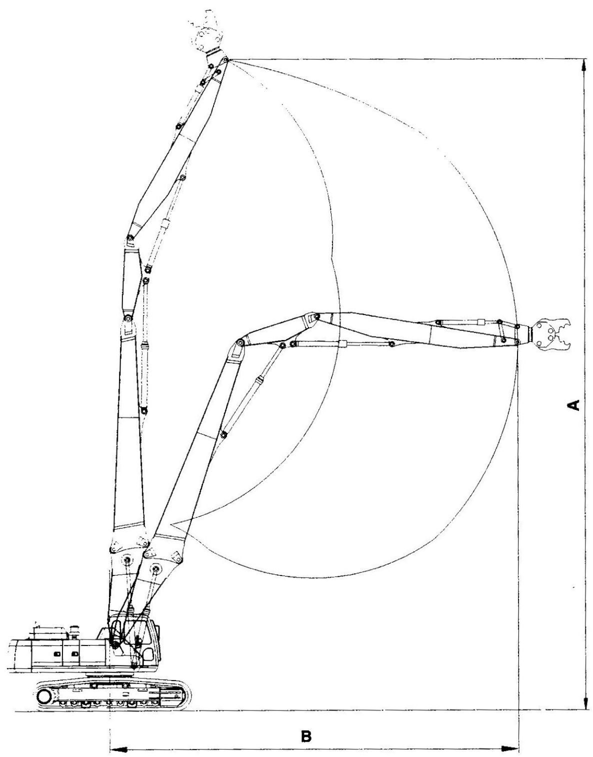
| Applicable Models | | ZAXIS240LCK | ZAXIS350LCK | ZAXIS450LC |
|---|---|---|---|---|
| Type | | HL230 | HL350 | HL450 |
| Crusher | | | | |
| Max. working height | mm | 16 000 | 22 000 | 26 000 |
| Max. working radius | mm | 8 760 | 11 600 | 14 350 |
| Operating weight* | kg | 28 000 | 39 000 | 52 600 |
| Crusher weight | kg | 2 100 | 2 100 | 2 100 |
| Backhoe | | | | |
| Max. working height | mm | 10 240 | 12 180 | 11 880 |
| Max. working radius | mm | 10 450 | 11 820 | 12 370 |
| Operating weight | kg | 26 100 | 34 640 | 49 400 |
| Bucket capacity (PCSA heaped) | $m^3$ | 0.8 | 1.15 | 1.9 |

<Note> * Excluding crusher weight.

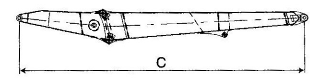
Figure C.5: HITACHI Long Reach Demolition Front [16]

# High Reach Demolition

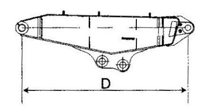
## Working Range



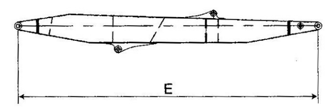

## Boom



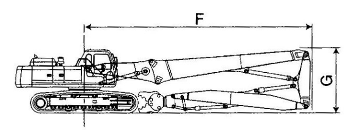

## Middle Arm




## End Arm




## Transportation Position




The Standard Package includes :
- Boom, Middle Arm and End Arm
- Middle Arm Cylinder
- End Arm Cylinder
- Bucket Cylinder with linkage
- All Plumbing on the boom and arm
- Additional Counterweight
- Boom angle sensing and Warning device.

*** Specification**

| Model | | SDM 30-59 | SDM 36-69 | SDM 45-82 |
|---|---|---|---|---|
| Max. Pin Height | (A) | 18.0 M | 21.0 M | 25.0 M |
| *Max. Working Reach | (B) | 11.0 M | 13.1 M | 15.2 M |
| Boom Length | (C) | 8.6 M | 10.6 M | 13.3 M |
| Middle Arm Length | (D) | 2.6 M | 2.6 M | 2.6 M |
| End Arm Length | (E) | 5.0 M | 6.5 M | 7.5 M |
| Transportation Length | (F) | 9.05 M | 11.02 M | 13.75 M |
| Transportation Height | (G) | 3.35 M | 3.35 M | 3.35 M |
| *Allowable Tool Weight | | 2.2 Ton | 2.2 Ton | 2.2 Ton |
| *Additional Counterweight | | 4.0 Ton | 4.5 Ton | 5.5 Ton |
| Excavator Class | | 27~32 Ton | 32~40 Ton | 40~50 Ton |

- Asterisk (*) means the data can be changed according to base machine's condition.
- Above specifications can be changed according to customer's requirement.

Figure C.6: SNS Demolition Applications [17]

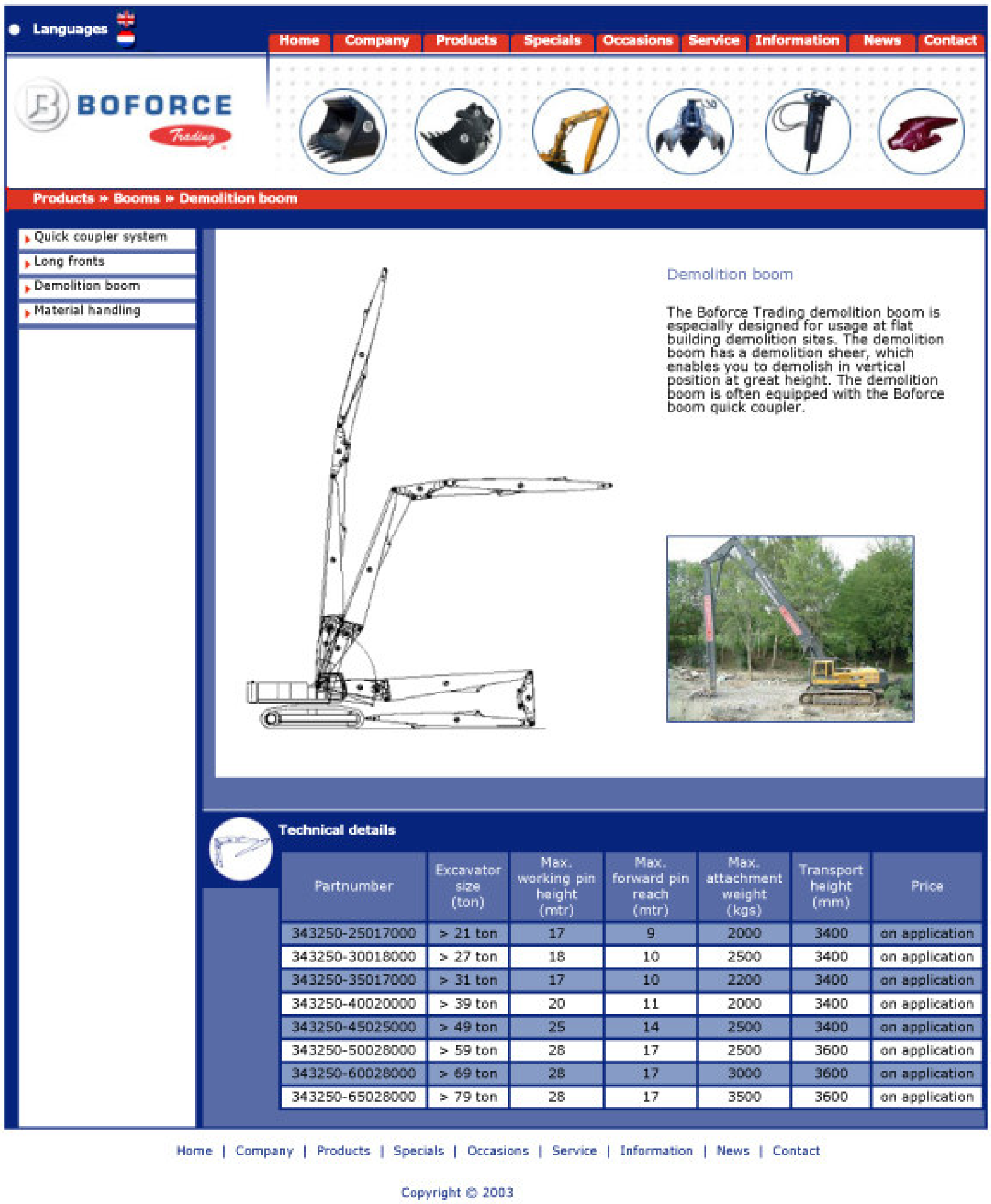



Figure C.7: BOFORCE Demolition Boom [18]

Kocurek : Product Range : Ultra High Reach : Technical Data

| TELESCOPIC ULTRA HIGH REACH DEMOLITION - TYPICAL WORKING RANGE | | | | |
|---|---|---|---|---|
| BASE MACHINE WEIGHT (TONNES) | Ref | 65 | 80 | 120 |
| MAX. WORKING PIN HEIGHT (M) | A | 35.0 | 40.0 | 50.0 |
| MAX. FORWARD REACH (M) | B | 17.0 | 20.0 | 25.0 |
| WORKING PIN HEIGHT (BOOM RETRACTED) (M) | C | 19.0 | 21.0 | 24.0 |
| TELESCOPIC SECTION (M) | D | 4.8 - 9.3 | 5.4 - 14.4 | 5.6 - 21.6 |
| MAX. ATTACHMENT WEIGHT (KG) | - | 2500 | 2500 | 2500 |

**These values are for guidance only. Kocurek pride themselves on the ability to develop bespoke solutions to meet individual customers requirements.**

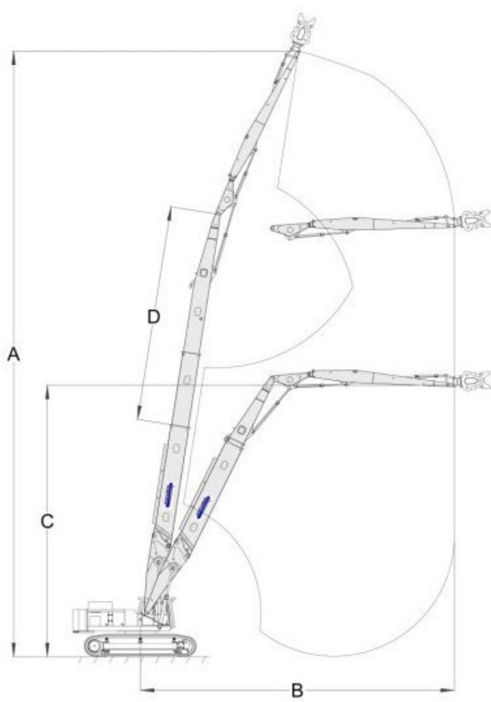



Figure C.8: KOCUREK Ultra High Reach [19]

# APPENDIX D

## LIST OF SYMBOLS

O (0,0) : The Origin (the intersection point of ground and rotation axis of upper frame)

A ($x_1$,$y_1$): The Cartesian coordinate of boom cylinder joint on the upper frame

B ($x_2$,$y_2$): The Cartesian coordinate of the joint between boom and upper frame

C ($x_3$,$y_3$): The Cartesian coordinate of the joint between boom and middle arm

D ($x_4$,$y_4$): The Cartesian coordinate of the joint between end arm and middle arm

E ($x_5$,$y_5$): The Cartesian coordinate of the joint between end arm and tool

Body-1: Upper frame

Body-2: Boom

Body-3: Middle arm

Body-4: End Arm

$a_1$ : Length between points A and B

$a_2$ : Length of the boom

$a_3$ : Length of the middle arm

$a_4$ : Length of the end arm

$a_5$ : Length between point E and centre of gravity of the tool

$b_2$ : Length from point B to boom cylinder joint on the boom

$b_3$ : Length from point C to middle arm cylinder joint on the middle arm

$b_4$ : Length from point D to end arm cylinder joint on the end arm

$c_2$ : Length from point C to middle arm cylinder joint on the boom

$c_3$ : Length from point D to end arm cylinder joint on the middle arm

$c_4$ : Length from point E to tool cylinder joint on the end arm

$\alpha_1$ : Angular position of the boom cylinder joint on the upper frame

$\alpha_2$ : Angular position of the boom cylinder joint on the boom

$\alpha_3$ : Angular position of the middle arm cylinder joint on the middle arm

$\alpha_4$ : Angular position of the end arm cylinder joint on the end arm

$\beta_2$ : Angular position of the middle arm cylinder joint on the boom

$\beta_3$ : Angular position of the end arm cylinder joint on the middle arm

$\beta_4$ : Angular position of the tool cylinder joint on the end arm

$s_2$ : Length of the boom cylinder

$s_3$ : Length of the middle arm cylinder

$s_4$ : Length of the end arm cylinder

$\Delta s_2$ : Stroke of the boom cylinder

$\Delta s_3$ : Stroke of the middle arm cylinder

$\Delta s_4$ : Stroke of the end arm cylinder

$\psi_1$ :Angular position of the boom with respect to the universal coordinate system

$\psi_2$ :Angular position of the middle arm with respect to the universal coordinate system

$\psi_3$ :Angular position of the end arm with respect to the universal coordinate system

$\gamma$ : Allowed maximum angular position of the boom

$W_{cw}$ : Counter weight

$d_{cw}$ : Horizontal length between center of gravity of counterweight and total center of gravity

$M_{att}$ : Moment of attachments (boom, arms and tool) to the total center of gravity

$M_{frame}$ : Moment of the all other elements of standard excavator to the total center of gravity

d : Distance between center of gravity and rotation axis

$W_2$ : Boom weight

$W_3$ : Middle arm weight

$W_4$ : End arm weight

$W_{tool}$ : Tool weight

$F_{s2}$ : Force created in boom cylinder

$F_{s3}$ : Force created in middle arm cylinder

$F_{s4}$ : Force created in end arm cylinder

$F_{tool}$ : Reaction force acting on the tool

$\theta_{tool}$ : Direction of $F_{tool}$

I : moment of inertia

$\sigma_{bending}$ : bending stress

$\sigma_{tensile}$ : tensile stress

$\sigma_{upper}$, $\sigma_{lower}$: calculated stresses

$\sigma_y$ : Yield strength of the material

S : safety factor