3.1 Overview

This chapter explains the final design for all parts and features of the project, which include mechanical, electrical and electronic aspects that embodies desired practical benefits for a mechatronics project design as shows in figure (3.1).

The design of a poly aluminum chloride (PACL) Automatic Filling System which aims to discharge drums that contain chemical substances (PACL) which used in drinking water treatment into huge tanks that provide dosing pumps with enough amounts to complete the treatment processes.

The project Hardware components such as the Programmable Logic Controller (PLC)**,** sensors, pump, motors and limit switches will be used to build poly aluminum chloride Automatic Filling System. Power screw drive to move the pump up and down depends on an electric motor clockwise and anticlockwise rotation direction.

As we know, new designs usually depends on primary assumptions as shown in table (3.1) as a prototype reflecting the product expectations and involve all element that impact it, and trying to improve all assumptions in parallel with achieving the optimum final product as shown in table (3.13).

3.2 System description

This system replaces the manual discharging process for PACL drums into buffer tanks that normally used in water treatment plants as we mentioned in the problem statement.

The system depends on converting the rotational movement (electrical motor) into linear movement via a ball screw to lift the pumps up and down. Two limit switches are used to determine the upper position (idle point) and the lower position (running point) as well as using a level sensor to determine if there is a liquid inside the drum or not as illustrate in figure (3.1).

The system is driven near to target drum via wheels, whereas the pump should be at the idle point before starting the system.

The operator must ensure that the system is located in the right position before pressing the start push button as illustrated in figure (3.2).

When starting the electric motor it rotates in clockwise direction moving the ball screw via pulley and belt so that the ball nut goes down carrying the pump with it until they reach the running point, the running point limit switch sends a signal to the PLC unit to stop the motor, the level sensor will detect a liquid presence and sends a signal to the PLC, when the PLC receives the two signals from the limit switch (idle) and the level sensor it will send a signal to run the pump directly. When the liquid is fully discharged from the drum the level sensor will send a signal to stop the pump then the motor will rotate anticlockwise to lift the pumps up to the idle point, at that time the limit switch (idle) will stop the motor to complete the process and we can consider this as the last step of the discharging process.

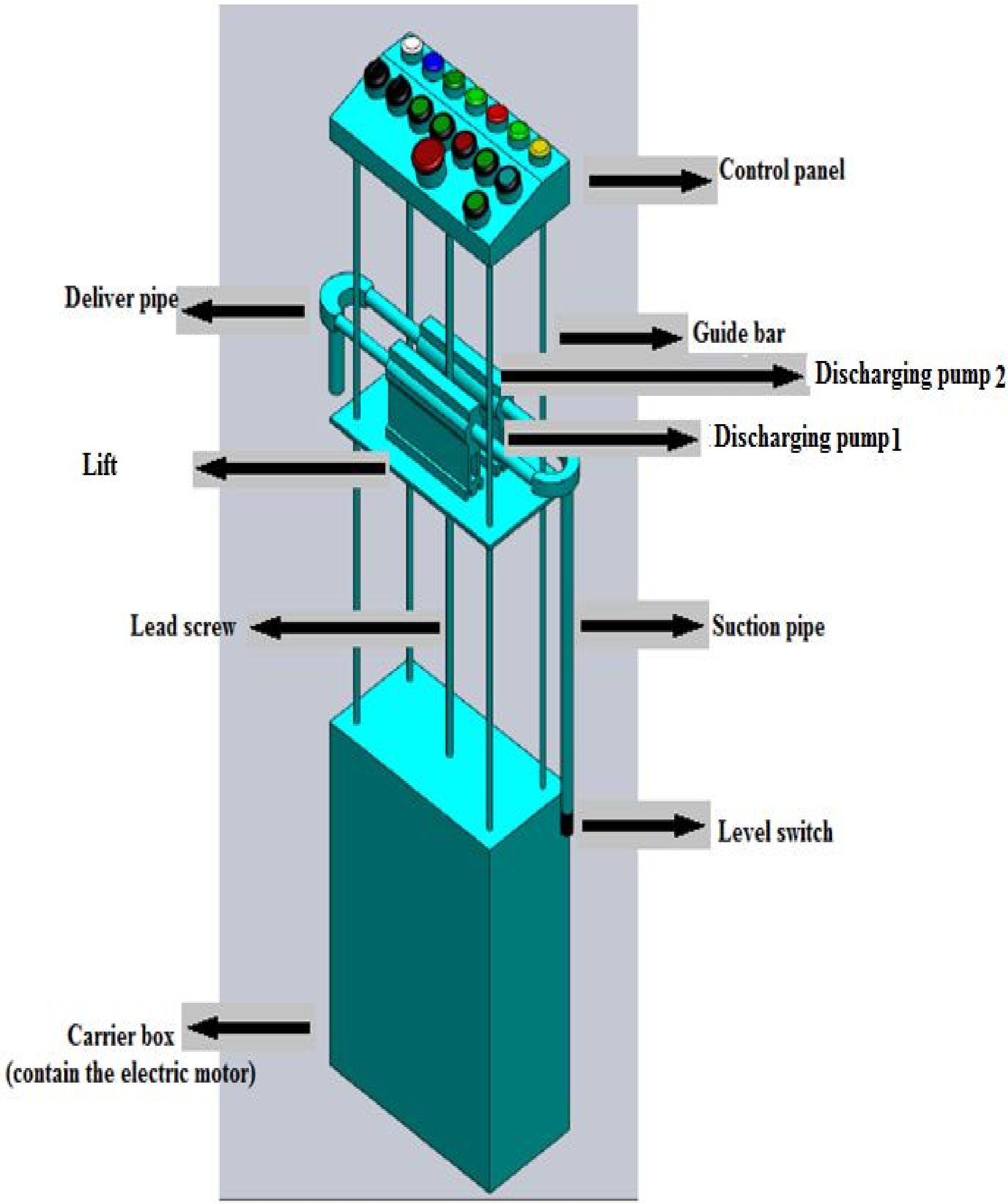

Figure (3.1): Overview of PACL Automatic Filling System

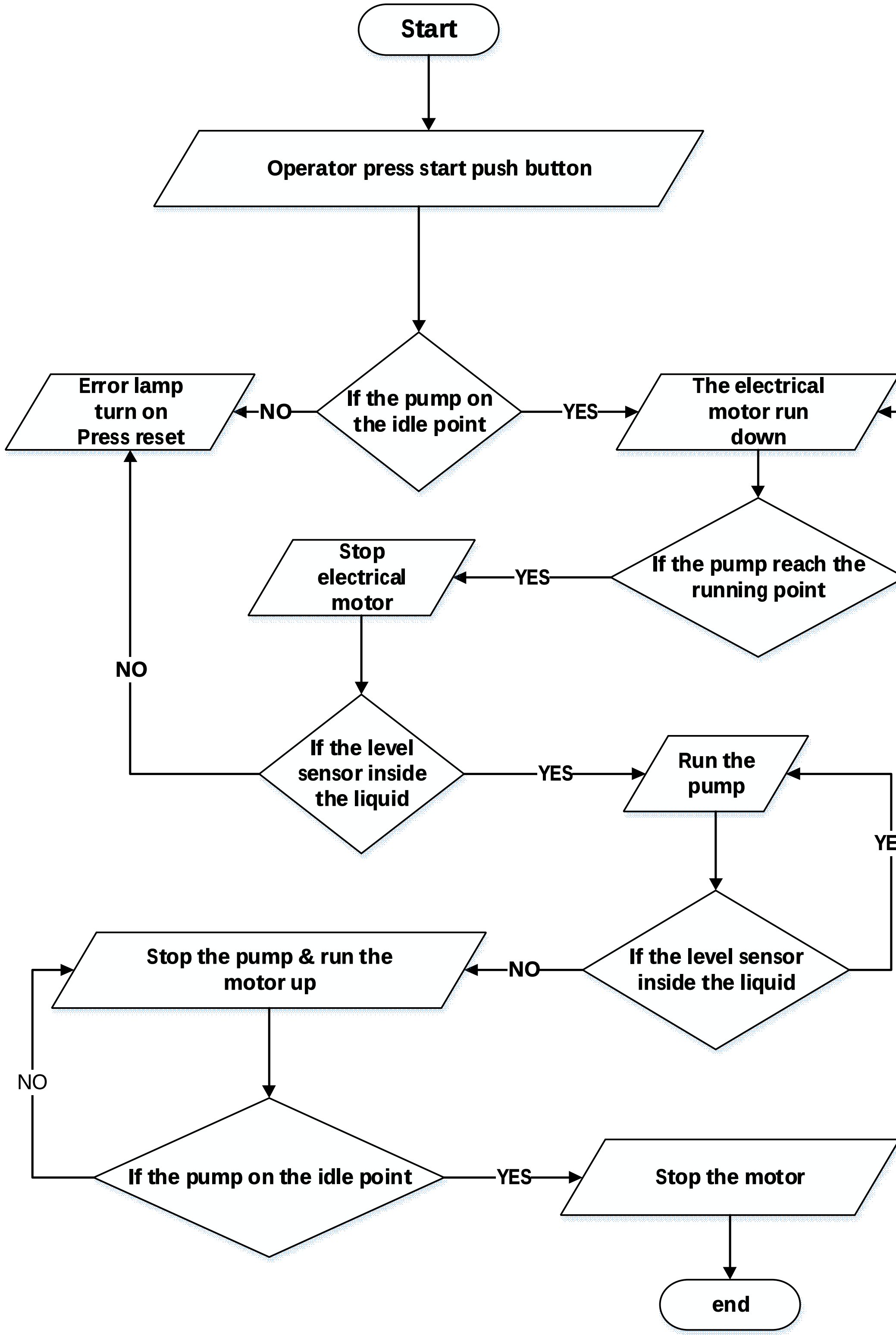

Figure (3.2): System flow chart

3.3 calculations

The necessary calculation which involve the main parts on the system such as the pump, ball screw and electrical motor have been done as shown below:

3.3.1 Pump calculations

To select the right flow rate we have to know the drum volume and the discharging time as shown in Equation (3.1).

The target is to discharge the drum in about 2 minute as well as the drum has 200 liter of fluid (PACL) so

$$Q = \frac{\text{drum fluid volume}}{time} \qquad (3.1)$$

$$Q = \frac{200\ L}{2\ \ MIN}$$

$$Q = \mathbf{1.67 * 10^{-3}}\ m^3\ /s$$

To calculate the static head we have to calculate the suction head and the discharge head as shown in equation (3.2)

Static head = discharge head – suction head (3.2)

Equation (3.2) illustrates the relationship between static, discharge and suction heads

From the design the suction and discharge head was selected as follow:

Discharge head = **0.1** m

Suction head = **- 0.9** m (blew the pump center)

$$\text{Static head} = 0.1 - (-0.9)$$

Static head = **1** m

To calculate the head loss the types of the losses have been know as follow:

1- Friction loss (pipe line)

2- Fitting loss (elbow, tee, check valve…etc.)

Friction loss:

Equation (3.3) reflects the friction loss calculation method [8]:

$$h_{friction} = f * \frac{l * v^2}{2 * d * g} \qquad (3.3)$$

Where:

$h_{friction}$: the loss head due to friction (meter)

f: friction coefficient

l: length of pipe (meter)

v: fluid velocity

d: cross section diameter

g: gravity acceleration

To find the f (friction coefficient) that depends on Reynold number ($\boldsymbol{R_e}$).

If $\boldsymbol{R_e}$ < 2000 that means the flowrate is laminar and f is given by [8]: (3.4):

$$f = \frac{64}{R_e} \tag{3.4}$$

where:

f: friction coefficient

R_e : Reynold number

If $\boldsymbol{R_e}$ > 4000 that means the flowrate is turbulent and f equation as equation (3.5):

$$f = \frac{0.316}{{R_e}^{0.25}} \tag{3.5}$$

The Reynold number is depend on the velocity, the viscosity and internal pipe diameter as shown in equation (3.6):

$$R_e = \frac{V*D}{v} \tag{3.6}$$

Where:

R_e : Reynold number

V: fluid velocity (m^2/sec)

D: cross section diameter (m)

v: fluid viscosity

$$R_e = \frac{3.3*0.0254}{4*10^{-6}}$$

$$\boldsymbol{R_e} = \mathbf{20955}$$

Then:

$$f = \frac{0.316}{(20955)^{0.25}}$$

$$f = \mathbf{0.0263}$$

$$h_{friction} = 0.0263 * \frac{1.20 \; * \; 3.3^2}{2 * 0.0254 * 9.81}$$

$$\boldsymbol{h_{friction}} = \mathbf{0.688 \approx 0.7\ m}$$

$$h_{fitting} = \frac{K * V^2}{2\,g} \qquad (3.7)$$

Where:

$h_{fitting}$: the head loss due to fitting

K: fitting coefficient

V: fluid velocity (m/sec)

g: gravity acceleration (m^2/sec)

We used (2) pieces of elbows (K = 0.20)

Applying equation (3.7) to find fitting loss:

$$h_{fitting1} = \frac{0.20 * \; 3.3^2}{2 * 9.81}$$

$$\boldsymbol{h_{fitting1}} = \mathbf{0.11}\,m \quad \text{(For one piece)}$$

We used (2) pieces of check valves (K = 2)

$$h_{fitting2} = \frac{2 * \; 3.3^2}{2 * 9.81}$$

$$h_{fitting2} = 1.10\text{ m} \quad \text{(For one piece)}$$

$$H_L = 2\,(h_{fitting1}) + 2(\,h_{fitting2})$$

$$H_L = 2 * .11 + 2 * 1.1$$

$$\boldsymbol{H_L} = \mathbf{2.42\ m}$$

Applying equation (3.8) to find the total head:

$$H_{total} \; = \; H_{static} \; + H_{Loss} \qquad (3.8)$$

Where:

H_{total} : the summation of all effected heads (m)

H_{static} : the difference between delivery head and suction head (m)

H_{Loss} : the dissipated heads (m)

$$H_{loss} = 0.7 + 2.42$$

$$H_{loss} = \mathbf{3.12 \approx 3.50\ m}$$

Due to add (10% as a safety factor)

$$\boldsymbol{H_{total}} = \mathbf{1 + 3.50 = 4.50\ m}$$

Applying equation (3.9) to find the hydraulic power

$$P_{hydarulic} = \rho * g * H * Q \qquad (3.9)$$

Where:

$P_{hydarulic}$: the hydraulic power (watt)

ρ: fluid density(m^3/kg)

g: gravity acceleration(m^2/sec)

H: total head(m)

Q: fluid flow rate (m^3/sec)

$$P_{hydarulic} = \frac{1360*9.81*4.50*1.67*10^{-3}}{100}$$

$$\boldsymbol{P_{hydarulic}} = \mathbf{1.003 \approx 1.10\ kW}$$

Let the efficiency (η) = 0.75 (due to design is built on the worst choices)

Applying equation (3.10) to find electrical pump motor:

$$\text{The pump power (P)} = P_{hydarulic} / \eta \qquad (3.10)$$

Where:

P: the pump power

$P_{hydarulic}$: hydraulic power

η: the pump efficiency

$$P = \frac{1.10}{0.75}$$

$$P = \mathbf{1.4667 \approx 1.50\ kW}$$

The 1.50 **kW** centrifugal pump will be selected

3.3.2 Ball screw calculations

According to design specification in table (3.1) the required dimensions for the ball screw which is shown in figure (3.3) will be calculated. The required calculations needed for:

- ✓ Screw shaft diameter
- ✓ Lead
- ✓ Nut module no
- ✓ Accuracy
- ✓ Axial clearance
- ✓ Screw shaft support method

Table (3.1): Design requirements

NO	REQUIRED	VALUE
01	Table Mass	6.00 kg
02	Work Mass	54.00 kg
03	Stroke Length	1000 mm
04	Maximum Speed	0.10 m/s
05	Acceleration Time	0.15 s
06	Deceleration Time	0.15 s
07	Number Of Reciprocations Per Minute	2.50 min^{-1}
08	Back Lash	15 mm
09	Positioning Accuracy	±0.3mm/1000mm
10	Positioning Accuracy Repeatability	± 0.10 mm
11	Positioning Accuracy	---- mm
12	Minimum Feed Amount	0.02 mm/pulse
13	Service Life Time	3 YEARS
14	Rated Rotational Speed	1200 rpm
15	Frictional Coefficient Of The Guide Surface	µ= 0.003
16	Guide Surface Resistance	F= 15 N

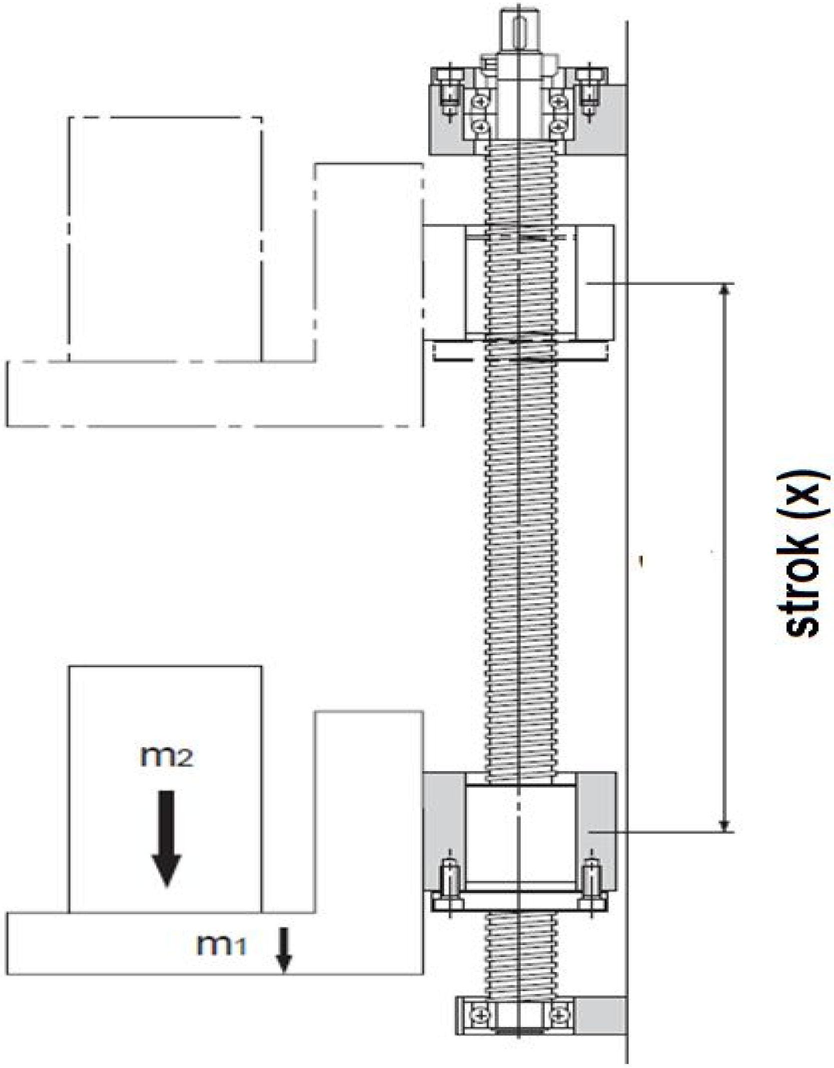

Figure (3.3): Ball screw function

To select the right screw shaft the primary selection should be selected as follow:

1. Selection of accuracy grade

C5 accuracy was selected due to screw shaft transportation usage as well as to grantee high accuracy as shown in table (2.1).

2. Screw shaft length

- ✓ Nut length = **100** mm
- ✓ Stroke length = **1000** mm
- ✓ Shaft end length = **100** mm

So the total length = 1200 mm

3. Selecting lead

The lead screw calculations are depended on equation (3.11):

$$\text{Lead (L)} \geq \frac{V_{max}}{N_{max}} \qquad (3.11)$$

Where:

V_{max} : Maximum linear speed (m/sec)

N_{max} : Maximum rotational speed (rpm)

$$\text{Lead (L)} \geq \frac{200*60}{1200} = 10 \text{ mm}$$

4. Selecting screw shaft diameter

The screw shaft diameter has a close relationship with the lead as shown in table (3.2) as a same amount can't be neglected the relationship between the screw shaft diameter and the accuracy as illustrates in table (3.3). Addition to that the screw shaft diameter has a relationship with accuracy grade and length of the screw thread as shown in table (3.4) so the balancing between this relationships should be done.

Table (3.2): Relationship between shaft diameter and leads

Unit: mm

Lead / Shaft diameter	3	4	5	6	8	10	12	16	20	25	32	40	50	64	80
10	●			●											
12					●		●								
14		●	●												
15									●						
16						●		●			●				
18					●										
20			●			●			●			●			
25			●			●				●			●		
28				●											
32						●					●			●	
36						●									
40						●						●			●
45							●								
50						●		●					●		

According to the calculated value of lead in equation (3.11) and from table (3.2) we can select the diameters (**16, 20, 25, 32, 36, 40** and **50)**

Table (3.3): Relationship between shaft diameter and accuracy

Accuracy grade / Screw shaft diameter	C0	C1	C2	C3	C5	Ct7	rolled ball screw (Ct10)
4	90	110	120	140	140	140	—
6	150	180	200	250	250	250	—
8	240	280	340	340	340	340	—
10	350	400	500	500	500	550	800
12	450	500	650	700	750	800	800
14	600	650	750	800	1000	1000	1000
15	600	700	800	900	1250	1250	1500
16	600	750	900	1000	1500	1500	1500
18	—	—	—	—	—	—	1500
20	850	1000	1200	1400	1900	1900	2000
25	1100	1400	1600	1900	2500	2500	2500
28	1100	1400	1600	1900	2500	2500	2500
32	1500	1750	2250	2500	3200	3200	3000(4000)
36	1500	1750	2250	2500	3200	3500	3000
40	2000	2400	3000	3400	3800	4300	4000 (5000)
45	2000	2400	3000	3400	4000	4500	4000
50	2000	3200	4000	4500	5000	5750	4000
63	2000	4000	5000	6000	6800	7700	
80		4000	6300	8200	9200	10000	
100		4000	6300	10000	12500	14000	
125				10000	14000	14000	

Now we compare the previous diameters with the capable screw shaft length according to accuracy grade of C5 as shown in table (3.3) taking into account the stroke length ≥ 1000 mm.

When comparing between C5 and the selected diameters, all capable screw shaft length is accepted (1500 mm, 1900 mm, 2500 mm, 2500 mm, 3200 mm, 3800 mm and 5000 mm).

Table (3.4): Range between shaft diameter and accuracy range

Screw shaft diameter	Effective length of the screw thread (maximum)				
	Axial play *T*		Axial play *S*		
	C0~C3	**C5**	**C3**	**C5**	**Ct7**
4~6	80	100	80	100	–
8~10	250	200	250	300	–
12~16	500	400	500	600	700
20~25	800	700	1000	1000	1000
32~40	1000	800	2000	1500	1500
50~63	1200	1000	2500	2000	2000
80~125	–	–	4000	3000	3000

From the selected stroke length (1000 mm) with accuracy grade of C5 and axial play S, the range of screw shaft diameter is 20-25 mm as shown in table (3.4).

20 mm shaft screw diameter is selected due to cost advantage.

Table (3.5): Ball nut type section according to lead and diameter of screw shaft

Ball nut No.	Shaft dia. d	Lead l	Ball dia. D_w	Ball circle dia. d_m	Root dia. d_r	Effective turns of balls: Turns x Circuits	Basic load rating (N) Dynamic C_a	Basic load rating (N) Static C_{0a}	Basic load rating {kgf} Dynamic C_a	Basic load rating {kgf} Static C_{0a}	Axial play Max.
RNSTL 1404A3.5S	14	4	2.778	14.5	11.5	3.5 x 1	5370	10800	545	1100	0.10
RNSTL 1405A2.5S	14	5	3.175	14.5	11.0	2.5 x 1	5260	9720	535	990	0.10
RNSTL 1808A3.5S	18	8	4.762	18.5	13.6	3.5 x 1	13200	25800	1350	2630	0.15
RNSTL 2005A2.5S	20	5	3.175	20.5	17.0	2.5 x 1	6360	14200	650	1450	0.10
RNSTL 2010A2.5S	20	10	4.762	21.25	16.2	2.5 x 1	10900	21800	1110	2220	0.15
RNSTL 2505A2.5S	25	5	3.175	25.5	22.0	2.5 x 1	7070	18200	720	1850	0.10
RNSTL 2510A5S	25	10	6.35	26	19.0	2.5 x 2	31800	70300	3240	7170	0.20
RNSTL 2806A2.5S	28	6	3.175	28.5	25.0	2.5 x 1	7430	20300	760	2070	0.10
RNSTL 2806A5S						2.5 x 2	13500	40600	1380	4140	
RNSTL 3210A2.5S	32	10	6.35	33.75	27.0	2.5 x 1	19700	46100	2010	4700	0.20
RNSTL 3210A5S						2.5 x 2	35700	92200	3640	9410	
RNSTL 3610A2.5S	36	10	6.35	37	30.0	2.5 x 1	21000	51000	2140	5200	0.20
RNSTL 3610A5S						2.5 x 2	38100	102000	3890	10400	
RNSTL 4512A5S	45	12	7.144	46.5	39.0	2.5 x 2	49600	147000	5060	15000	0.23

According to shaft screw diameter of 20 mm and shaft screw lead of 10 mm; the ball nut no. RNSTL 2010A2.5S is selected as shown in table (3.5).

The ball nuts and screw shafts specifications (diameter, lead, ball diameter dw, ball circle diameter dm, root diameter, turns x circuits, basic load rating and dimensions) as clearly shown in tables (3.5) and (3.6).

Table (3.6): Screw shaft section according to lead and diameter of screw shaft

Ball nut dimensions											Arbor		Screw shaft	
Length	Width	Center height	Bolt hole					Oil hole			Outside dia.	Bore	Standard length	Screw shaft No.
L_n	W	H	A	B	C	J	K	E	F	U	d_0	d_1	L_s	
38	34	13	22	26	8	M4	7	7	3	20	11.5	9.5	500 1000	RS1404A**
38	34	13	22	26	8	M4	7	7	3	21	11.0	9.0	500 1000	RS1405A**
56	48	17	35	35	10.5	M6	10	8	3	26	13.6	11.6	500 1000 1500	RS1808A**
38	48	17	22	35	8	M6	9	6	2	27	17.0	14.6	500 1000 2000	RS2005A**
58	48	18	35	35	11.5	M6	10	10	2	28	16.2	13.8	500 1000 2000	RS2010A**
35	60	20	22	40	6.5	M8	10	6	0	27	22.0	19.6	1000 2000 2500	RS2505A**
94	60	23	60	40	17	M8	12	10	0	32	19.0	16.6	1000 2000 2500	RS2510A**
42 67	60 60	22 22	18 40	40 40	12 13.5	M8	12	8	0	32	25.0	22.6	1000 2000 2500	RS2806A**
64 94	70 70	26 26	45 60	50 50	9.5 17	M8	12	10	0	38	27.0	24.6	1000 2000 3000	RS3210A**
64 96	86 86	29 29	45 60	60 60	9.5 18	M10	16	11	0	41	30.0	27.6	1000 2000 3000	RS3610A**
115	100	36	75	75	20	M12	20	13	0	46	39.0	35.8	2000 3000 4000	RS4512A**

The screw shaft no. RS 2010A2.5S is selected from table (3.6) according to ball nut selected previously.

From the above equations and tables the primary selections is found as follow:

LEAD	SHAFT DIAMETER	STROKE	ACCURACY GRADE
10 mm	20 mm	1000 mm	C5

The ball nut number **RNSTL 2010A2-5S** and screw shaft number **RS 2010A**

- ❖ basic safety checks

i. Buckling load

When a slender shaft is subject to a compression force, as illustrated in the figure (3.4), a certain amount of lateral bending could occur due to the fact that most axial loads do not, in reality, bear down exactly on the center line of a shaft, and the induced stress combines with the axial compression stress. If the load is smaller than a certain level, the bending recovers. However, when the load is larger than such a level, the bending increases, and axial breakage eventually occurs. Such a phenomenon is called “buckling”.

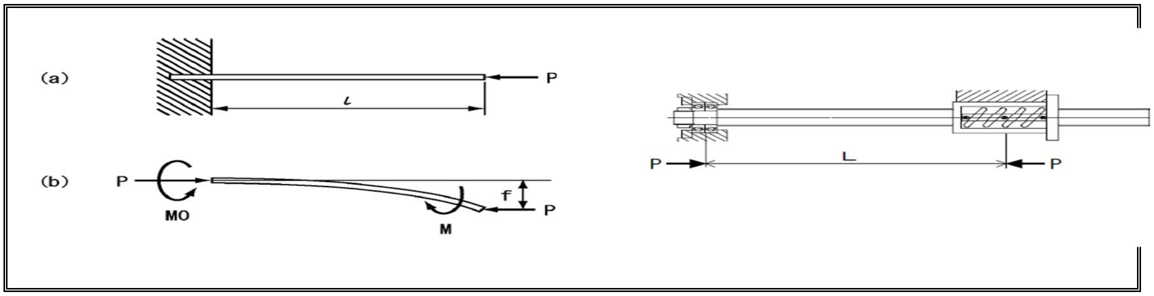

Figure (3.4): Buckling load

$$d_r \geq \left[\frac{P * L^2}{m * 10^4}\right]^{\frac{1}{4}} \qquad (3.12): [2].$$

Where:

dr: root diameter (mm)

P: axial allowable load (N)

L: screw shaft stroke length (mm)

m: supporting type coefficient

The buckling load is a main element that determine the supporting type that should be selected according to equation (3.12).

ii. Allowable tensile or compressive load (Po)

This refers to the limit of tensile or compression axial load that is applied to the screw shaft. If the axial load goes beyond a certain limit, the shaft will not withstand any more, and starts to show a permanent deformation (a deformation that does not disappear even after the load has been removed).

And if it goes further, the shaft will break. (Although this is a rough approach, there is no problem when the axial load is lower than somewhere around the allowable static load rating.)

When an excessive axial load is applied, the contact surface between steel balls and ball grooves create dent which won't recover even after the load has been removed, leaving permanent deformations on each surface. Measures must be taken to limit such deformations within a certain limit. Permanent deformations lead to noise and vibration, and to deteriorated functionality and shorter service life.

Based on the basic static load rating, confirm that no permanent deformations will be created on the contact surfaces between steel balls and ball grooves.

An allowable load rating **(Po)** against permanent deformation is as follows; (An allowable load is approximately one half of the allowable static load rating.)

iii. Basic Static Load Rating (C0a)

Defined as an axial load that causes the sum of a deformation that is formed on a steel ball and its contacting ball groove surfaces, while in a static state, to exceed 0.01% of the diameter of that steel ball. This figure is listed in table (3.5). A permanent deformation of 0.01% means, however, a dent of only 0.001 mm in the case of a 10 mm steel ball. Therefore, it is almost undetectable.

$$P_0 = C0a / fs \qquad (3.13): [2].$$

Where:

P_0 : Allowable load (N)

$C0a$: Basic static load (N)

f_s : Static allowable load coefficient (a safety factor)

For regular operation: 1 to 2

If there are vibrations or impact: 1.5 to 3

iv. Fatigue life

Even if ball screws are made using the most appropriate design and they are used properly, the surfaces of ball grooves will still start flaking (the surface metal falls apart in the form of scales) after a certain period of time since the steel balls roll on the ball groves with load applied to them. (This is a material fatigue phenomenon of contact surfaces caused by repetitive compressive stress between steel balls and the ball groove.) The total rotation number (or time period, travel distance) up until the first flaking occurs is called a “fatigue life”.

v. Basic dynamic load rating (Ca)

Defined as the axial load under which, when a group of identical ball screws are individually rotated under identical conditions, 90% will successfully achieve one million rotations without any flaking. This figure is listed in table (3.5).

vi. Mean effective load (Fm)

Of the various applications for ball screws, there is a case in which the axial load or feed speed varies with time. In such a case, obtain a mean effective load to compute life expectancy. It is important to prevent any trouble from occurring by getting detailed operating conditions and as much information as possible from customers.

Although there are three different ways to express “service life of ball screws” as shown below, “life” means “fatigue life” in most cases.

- check on allowable axial load

- calculation of axial load :

  Acceleration and deceleration at start- up and slow-down:

$$\alpha_1 = \frac{V_{max}}{t_1} \qquad (3.14)$$

Where:

α_1: Acceleration (m/sec^2)

V_{max} : Maximum velocity (m/sec)

t_1 : time period (sec)

$$\alpha_1 = \frac{0.2}{0.15} = \underline{1.333} \text{ m/s}^2$$

- During upward acceleration

The axial load that impact on the ball screw and shaft screw during upward acceleration can be calculated by equation (3.15): [2]

$$F_1 = (m_1 + m_2) * g + f + (m_1 + m_2) * \alpha_1 \qquad (3.15)$$

Where:

F_1: Axial load at upward acceleration period (N)

m_1: Work mass (kg)

m_2: Table mass (kg)

g: Gravity acceleration (m/sec^2)

f: Friction force (N)

α_1 : Acceleration (m/sec^2)

$$= \mathbf{(54 + 6) * 9.81 + 20 + (54 + 6) * 1.333 = 689 \text{ N}}$$

- During upward uniform motion

The axial load that impact on the ball screw and shaft screw during upward uniform motion can be calculated by equation (3.16): [2]

$$F_2 = (m_1+m_2) * g + f \qquad (3.16)$$

Where:

F_2: Axial load at upward uniform motion period (N)

m_1: Work mass (kg)

m_2: Table mass (kg)

g: Gravity acceleration (m/sec^2

f: Friction force (N)

= (54+6) * 9.81 + 20 = 609 N

- During upward deceleration

The axial load that impact on the ball screw and shaft screw during upward deceleration can be calculated by equation (3.17): [2]

$$F_3 = (m_1+m_2) * g + f - (m_1+m_2) * \alpha_1 \qquad (3.17)$$

Where:

F_3: Axial load at upward declaration period (N)

m_1: Work mass (kg)

m_2: Table mass (kg)

g: Gravity acceleration (m/sec^2)

f: Friction force (N)

α_1 : Acceleration (m/sec^2)

= **(54 + 6) * 9.81+ 20 - (54+ 6)* 1.333 = 529 N**

- During down ward acceleration

The axial load that impact on the ball screw and shaft screw during downward acceleration can be calculated by equation (3.18): [2]

$$F_4 = (m_1+m_2) * g - f - (m_1+m_2) * \alpha_1 \qquad (3.18)$$

Where:

F_4 : Axial load at downward acceleration period (N)

m_1: Work mass (kg)

m_2: Table mass (kg)

g : Gravity acceleration (m/sec^2)

f : Friction force (N)

α_1 : Acceleration (m/sec^2)

= **(54 + 6) * 9.81 - 20 - (54+ 6)* 1.333 = 489 N**

- During down ward uniform motion

The axial load that impact on the ball screw and shaft screw during downward uniform motion can be calculated by equation (3.19): [2]

$$F_5 = (m_1+m_2) * g - f \qquad (3.19)$$

Where:

F_5 : Axial load at downward uniform motion period (N)

m_1: Work mass (kg)

m_2: Table mass (kg)

g : Gravity acceleration (m/sec^2)

f : Friction force (N)

= **(54 + 6) * 9.81 – 20 = 568.6 N**

- During down ward deceleration

The axial load that impact on the ball screw and shaft screw during downward deceleration can be calculated by equation (3.20): [2]

$$F_6 = (m_1+m_2) * g - f + (m_1+m_2)* \alpha_1 \qquad (3.20)$$

Where:

F_6 : Axial load at downward deceleration period (N)

m_1: Work mass (kg)

m_2: Table mass (kg)

g : Gravity acceleration (m/sec^2)

f : Friction force (N)

α_1 : Acceleration (m/sec^2)

$$= \mathbf{(54 + 6) * 9.81 - 20 + (54+ 6)* 1.333 = 608.6\ N}$$

Usually at designing calculations the highest value is selected. The axial load during upward acceleration is a maximum axial load so

$$\boldsymbol{F_{max}} \mathbf{= 689\ N \approx 690\ N}$$

- ✓ buckling load :

the bulking load depend on the root diameter that shown in equation (3.12) as well as the root diameter calculations that should be done according to supporting coefficient (m) as shown in table (3.7) according to type of supporting method.

Table (3.7): Supporting coefficient (m)

Supporting method	**m**	**N**
Fixed-fixed support	19.9	4
Fixed-simple support	10.0	2
Fixed support-free	1.2	0.25
Simple- simple support	5.0	1

The supporting type that is applied is (fixed - fixed support) so the supporting coefficient (m) = 19.9

Depend on equation (3.12) the root diameter is calculated as:

$$\boldsymbol{d_r} \geq \left[\frac{690 * 1000^2}{19.9 * 10^4}\right]^{\frac{1}{4}}$$

$$\boldsymbol{d_r} \geq \mathbf{7.67\ mm}$$

At the table the $\boldsymbol{d_r}$ **= 16.20 mm**

Accepted result

vii. Maximum rotational speed

This is the highest speed that screw shaft can rotate with.

$$\boldsymbol{N_{max}} = \frac{\boldsymbol{V_{max} * 60}}{\boldsymbol{lead}} \qquad \textbf{(3.21)}$$

Where:

N_{max} : Maximum rotational speed

V_{max} : Maximum linear speed

$$\boldsymbol{N_{max}} = \frac{\mathbf{0.20 * 60}}{\mathbf{10}} * \mathbf{10^3} = \mathbf{1200\ rpm}$$

$$\boldsymbol{N_{max}} = \mathbf{1200}\ \boldsymbol{rpm}$$

To calculate the permissible Rotational Speed Determined by the Dangerous Speed of the Screw Shaft, the equation (3.22) should be applied.

$$N = f * \frac{d_r}{L^2} * 10^7 \qquad (3.22): [2].$$

Where:

N : permissible Rotational Speed (rpm)

$\boldsymbol{f}$: factor depend on supporting condition from table (3.8)

d_r : root diameter (mm)

L : screw shaft stroke length (mm)

Table (3.8): $\boldsymbol{f}$ value

Supporting condition	$\boldsymbol{f}$	$\boldsymbol{\lambda}$
Fixed-simple support	15.1	3.927
Fixed- fixed support	21.9	4.730
Fixed- support-free	3.4	1.875
Simple-simple support	9.7	$\boldsymbol{\pi}$

$$\mathbf{N = 21.9 * \frac{16.2}{1000} * 10^7 = 3547.8}$$

$$\mathbf{N \approx 3500\ rpm}$$

Accepted result

Permissible rotational speed is also limited by d m • n value (dm: ball pitch circle diameter mm; n: rotational speed per minute rpm). dm • n value indicates peripheral speed (revolution speed of balls).

Table (3.9): factor affecting on root diameter and Permissible rotational

<table>
<tr><td rowspan="2">For positioning (C_5 grade or higher)

For transporting type (Ct7 grade)</td><td>Standard specification</td><td>$d_m .n \le 70000$</td></tr>
<tr><td>High speed specification</td><td>$d_m .n \le 100000$</td></tr>
<tr><td colspan="2">For transportation type (Ct10 grade)</td><td>$d_m .n \le 50000$</td></tr>
</table>

In this case **dm • n =< 70000**

From table (3.5) **dm = 21.25 mm**

$$\mathbf{N = \frac{70000}{21.25} = 3294\ rpm}$$

$$\mathbf{N \approx 3300\ rpm}$$

Accepted result

To calculate a permissible load by C0a the equation (3.23) should be applied.

$$F_{a_{max}} = \frac{C0a}{f_s} \qquad (3.23)$$

Where:

$F_{a_{max}}$: maximum load

C0a : static load rating

$\boldsymbol{f_s}$: static permissible load factor from Table (3.10)

Table (3.10): Static permissible load factor ($\boldsymbol{f_s}$)

$\mathbf{f_s}$: static permissible load factor	
At time of normal operation	1~ 2
With vibration impact	1.5~ 3

The system was operated at normal time according to table (3.10) static permissible load factor ($\boldsymbol{f_s}$) = 2

From table (3.5) **C0a = 21800 N**

$$\boldsymbol{F_{a_{max}}} = \frac{\mathbf{21800}}{\mathbf{2}} = \mathbf{10900\ N}$$

$$\boldsymbol{F_{a_{max}}} = \mathbf{10900\ N}$$

Accepted result

- studying the service life

The calculation of service life and estimation of work period time that can support the evaluation of economic and technical feasibility which depend on equation (3.24)

$$L\ (rev) = \left(\frac{C_a}{f_s * F_m}\right)^3 * 10^6 \qquad (3.24)$$

Where:

L : Nominal life (rpm)

Ca : Dynamic load rating (N)

$\boldsymbol{f_s}$: Static permissible load factor from Table (3.10)

F_m : Average axial load (N)

Calculating the travel distance

Maximum speed V_{max} = 0.20 m/s

Acceleration time t_1= 0.15s

Deceleration time t_3 = 0.15 s

The system travel distance is divided to 6 partitions as follow:

1: Upward acceleration

2: Upward uniform motion

3: Upward deceleration

4: Downward acceleration

5: Downward uniform motion

6: downward deceleration

- ✓ Travel distance during acceleration

To calculate travel distance during acceleration equation (3.25) should be applied.

$$l_{1,4} = \frac{V_{max} * t_1}{2} \qquad \textbf{(3.25)}$$

Where:

$l_{1,4}$: Acceleration distance (mm)

V_{max} : Maximum linear speed (m/sec)

t_1 : Acceleration time (sec)

$$l_{1,4} = \frac{0.20 * .15}{2} = \mathbf{7.50\ mm}$$

✓ Travel distance during uniform motion

To calculate travel distance during uniform motion equation (3.26) should be applied.

$$\boldsymbol{l_{2,5}} = \boldsymbol{l_s} - \frac{\boldsymbol{V_{max} * t_1 + V_{max} * t_3}}{\boldsymbol{2}} \tag{3.26}$$

Where:

$\boldsymbol{l_{2,5}}$: Uniform motion distance (mm)

V_{max} : Maximum linear speed (m/sec)

t_1 : Acceleration time (sec)

l_s : Stroke length (mm)

$$\boldsymbol{l_{2,5}} = \boldsymbol{1000} - \frac{\boldsymbol{0.20 * 0.15 + 0.20 * 0.15}}{\boldsymbol{2}} = \mathbf{970\ mm}$$

✓ Travel distance during deceleration

To calculate travel distance during deceleration equation (3.27) should be applied.

$$l_{3,6} = \frac{V_{max} * t_3}{2} \tag{3.27}$$

Where:

$l_{3,6}$: Uniform motion distance (mm)

V_{max} : Maximum linear speed (m/sec)

t_1 : deceleration time (sec)

$$\boldsymbol{l_{3,6}} = \frac{\mathbf{0.20 * .15}}{2} = 15 \text{ mm}$$

Based on the conditions above, the relationship between the applied axial load and the travel distance is shown in the table (3.11).

Table (3.11): Applied axial load and travel distances

motion	**Applied axial load $\boldsymbol{f_{a_N}}$ (N)**	**Travel distance $\boldsymbol{l_N}$ (mm)**
NO(1) during upward acceleration	689	15
NO(2) during upward uniform motion	609	970
NO(3) during upward deceleration	529	15
NO(4) during down ward acceleration	489	15
NO(5) during down ward uniform motion	568.6	970
NO(6) during down ward deceleration	648.58	15

Calculating average Axial Load

The average axial load (which is a main element to calculate the nominal life) according to equation (3.28) is given by: [2]

$$F_m = \sqrt[3]{\frac{1}{2 * L_s}} (F_1{}^3 * L_1 + F_2{}^3 * L_2 + F_3{}^3 * L_3 + F_4{}^3 * L_4 + F_5{}^3 * L_5 + F_6{}^3 * L_6) \quad (3.28)$$

$$F_m = \sqrt[3]{\frac{1}{2 * 1000}} (689^3 * 15 + 609^3 * 970 + 529^3 * 15 + 568.6^3 * 15 + 568.6^3 * 970 + 648.6^3 * 15)$$

$$F_m = 589.8 \text{ N}$$

$$\boldsymbol{F_m = 589.8 \approx 590 \text{ N}}$$

To calculate the nominal life the equation (3.6) should be applied as follow:

From the table (3.5) $\boldsymbol{C_a}$ **= 10900**

$$\mathbf{L\ (rev) = (\frac{10900}{2 * 590})^3 * 10^6 = 788.2 * 10^6\ rev}$$

$$\mathbf{L\ (rev) = 790 * 10^6\ rev}$$

Average revolutions per minute is calculated to use in service life time equation (3.31) by applying equation (3.30): [2]

$$N_m = \frac{2 * n * L_s}{lead} \quad (3.30)$$

Where:

N_m: Average revolutions per minute

Ls : Stroke length (mm)

n : turns of ball

From the table (3.5) **n** = **2.5** x 1 turns x circuits

$$\boldsymbol{N_m} = \frac{\mathbf{2 * 2.50 * 1000}}{\mathbf{10}} = \mathbf{500\ rpm}$$

$$\boldsymbol{N_m} = \mathbf{500\ rpm}$$

To calculate the service life time on the basis of the nominal life equation (3.31) should be applied and the Average revolutions per minute (N_m) which given by equation (3.30) is used as follow: [2]

$$L_h = \frac{L}{60 * N_m} \qquad (3.31)$$

Where:

L_h : service life time (hr)

L : Nominal life (rpm)

N_m : Average revolutions per minute

$$\boldsymbol{L_h} = \frac{\mathbf{790 * 10^6}}{\mathbf{60 * 500}} = \mathbf{26333\ hr.}$$

$$\boldsymbol{L_h} \approx \mathbf{26000\ hr}$$

To calculate the Service Life in Travel Distance on the Basis of the Nominal Life the equation (3.32) should be applied as follow: [2]

$$L_s = L * lead * 10^{-6} \qquad (3.32)$$

Where:

Ls : Service life (km)

L : Nominal life (rpm)

$$L_s = 790 * 10^6 * 10 * 10^{-6} = 7900 \text{ Km}$$

$$\boldsymbol{L_s = 7900 \text{ Km}}$$

The selection of supporting units is essential step to guarantee well system performance according to table (3.12).

Table (3.12): support unit and shaft diameter

Table I-6•7 Support units for light load and applicable "shaft diameter/lead combinations"

	Support unit / reference number			"Shaft diameter/lead combinations" of standard ball screws that are applicable to support unit
	Square		Round	
	Fixed support side (driving motor side)	Simple support side (opposite to driving motor)	Fixed support side	
Light load / small equipment	WBK06-01A	–	WBK06-11	$\phi4\times1$, $\phi6\times1$
	WBK08-01A	WBK08S-01	WBK08-11	$\phi8\times1$, $\phi8\times1.5$, $\phi8\times2$, $\phi10\times2$, $\phi10\times2.5$
	WBK10-01A	WBK10S-01	WBK10-11	$\phi10\times4$, $\phi12\times2$, $\phi12\times2.5$, $\phi12\times5$, $\phi12\times10$
	WBK12-01A	WBK12S-01	WBK12-11	$\phi14\times5$, $\phi14\times8$, $\phi15\times10$, $\phi15\times20$, $\phi16\times2$ $\phi16\times2.5$, $\phi16\times5$, $\phi16\times16$, $\phi16\times32$
	WBK15-01A	WBK15S-01	WBK15-11	$\phi20\times4$, $\phi20\times5$, $\phi20\times10$, $\phi20\times20$, $\phi20\times40$
	WBK20-01	WBK20S-01	WBK20-11	$\phi20\times4$, $\phi20\times5$, $\phi20\times6$, $\phi20\times10$, $\phi20\times20$ $\phi25\times25$, $\phi25\times50$, $\phi28\times5$, $\phi28\times6$
	WBK25-01	WBK25S-01	WBK25-11	$\phi32\times5$, $\phi32\times6$, $\phi32\times8$, $\phi32\times10$ $\phi32\times25$, $\phi32\times32$,

From table (3.12) and according to screw shaft specifications that is selected (diameter 20 mm & lead 10 mm) the support units are selected as follow:

Driving motor side No **WBK15-01A**

Non driving motor side No **WBK15S-01**

The more details about the support units that are selected above is involved in figure (3.5).

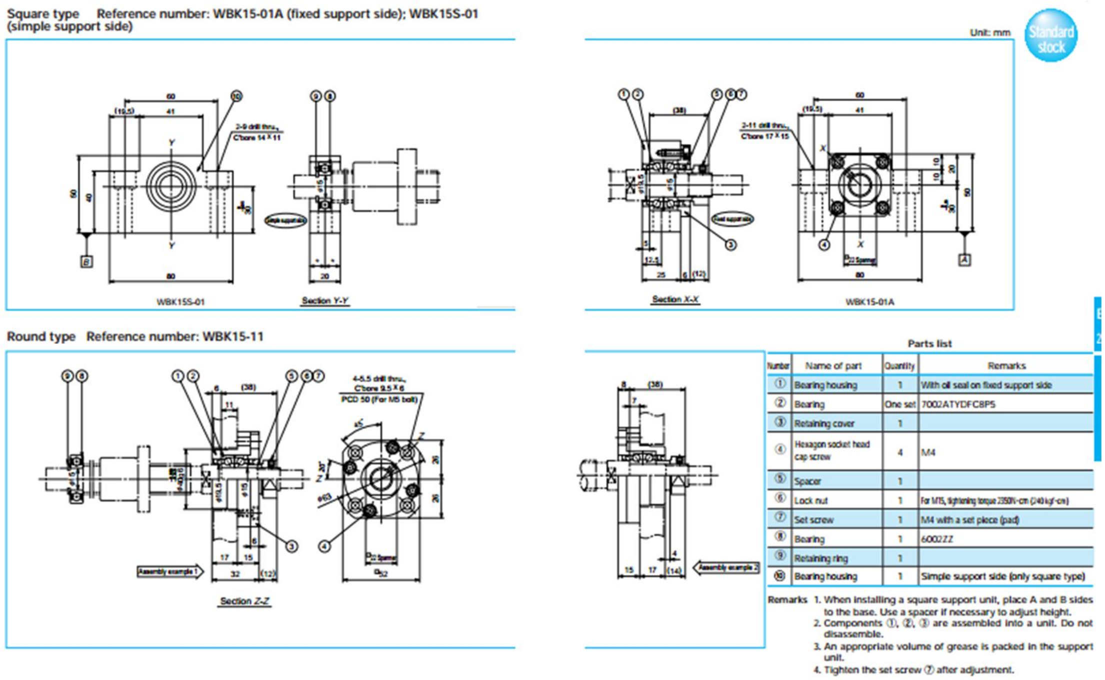

Figure (3.5): The support unit feature

Table (3.13): ball screw code numbers

Screw shaft	**RS2010A**
Nut Ball screw	**RNST2010A2.5S**
Drive side support unit	**WBK15-01A**
Non drive side support unit	**WBK15S-01**

The ball screw code numbers which are needed are known as shown in table (3.13) as well as they are used to request them from the manufacture.

3.3.3 Portable iron frame

A portable iron frame is a main part of the system, whereby carries all other parts of the system such as pumps, electric motor, ball screw…etc. and make them movable from one location to another which the targeted drums is positioned.

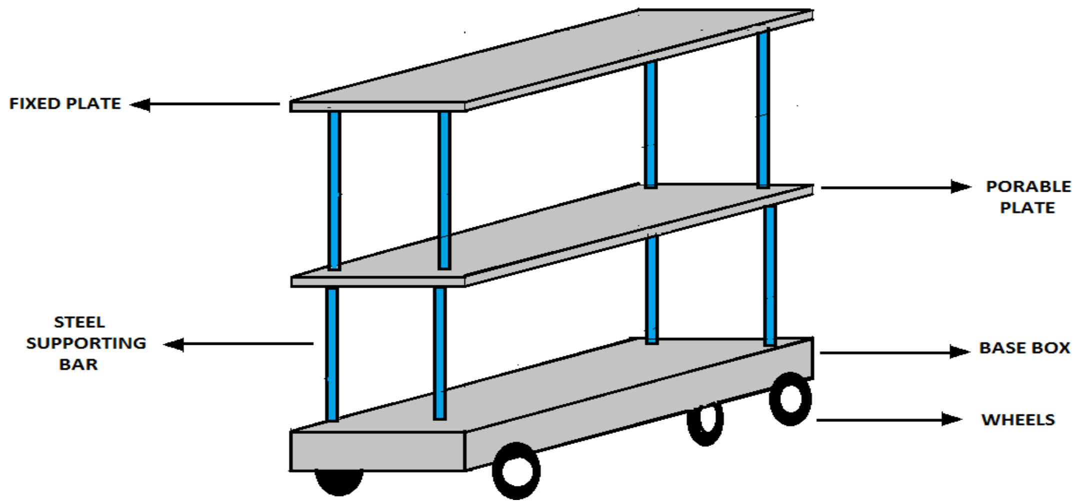

Figure (3.6): The portable iron frame

The frame consists of four wheels which assist in movement in all directions easily considering its ability to handle all essential parts along with additional equipment needed.

The main parts of a portable frame as illustrate in figure (3.6) as follow:

- ✓ Supporting vertical bar:
  4 supporting steel bars which perform as a guide for the process of lifting the pump up and down.
- ✓ Fixed plate:
  It's a plate placed at the top of the frame. Which fits the four supporting bars with each other and also fits the ball screw.

- ✓ Base box:

  It's a base acts as supporting box to install the bars and the ball screw and the wheels which drive the frame.
- ✓ Portable plate:

  It's the plate which holds the pumps and lift them up and down via the ball screw that connected in it.
- ✓ Frame handle:

  A steel handle used to facilitate the system movement in multiple directions depending on the underneath frame wheels.

3.3.4 PLC programming

The PLC is controlling the whole system based on the installed program which act like a guideline to perform the needed process in a sequential method and to ensure high efficiency in implementation.

The pictures below illustrate the system's PLC program that depends on Omron brand as follow:

Step (01):

This step illustrate the over view page of the program

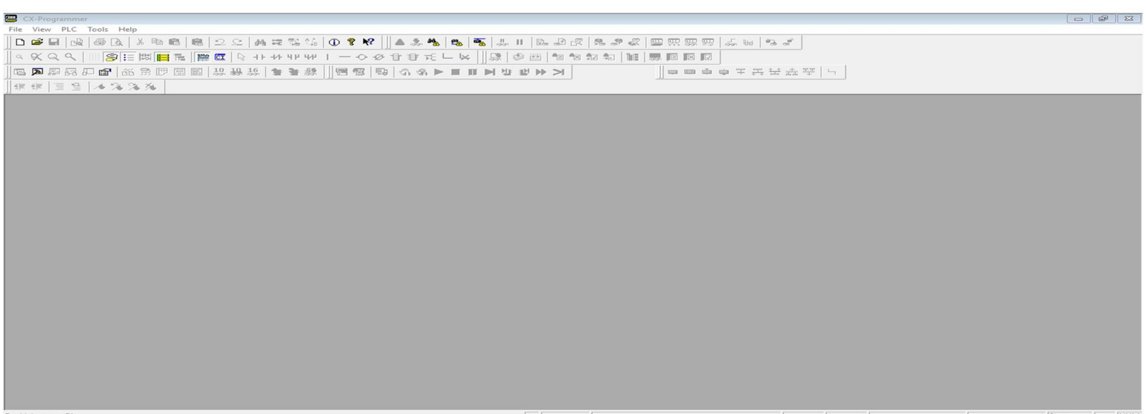

Figure (3.7): page of programming page

Step (02):

This step illustrates the configuration the device type and network type

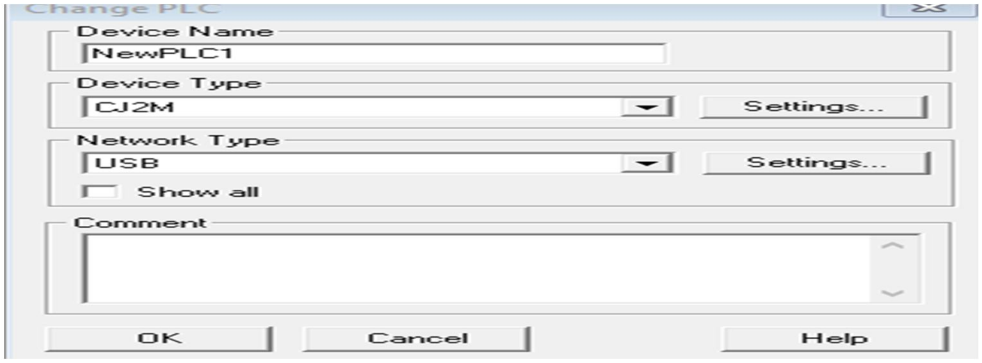

Figure (3.8): Configuration the device type and network type

Step (03):

This step illustrates the selecting CPU type

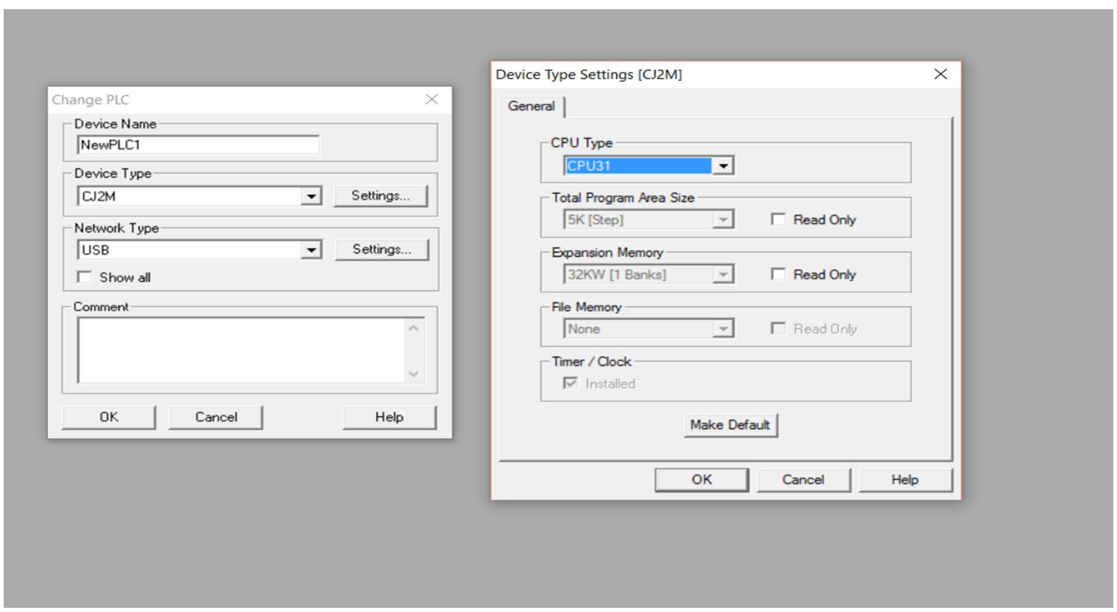

Figure (3.9): CPU type selection

Step (04):

This step illustrates the new page that is opened

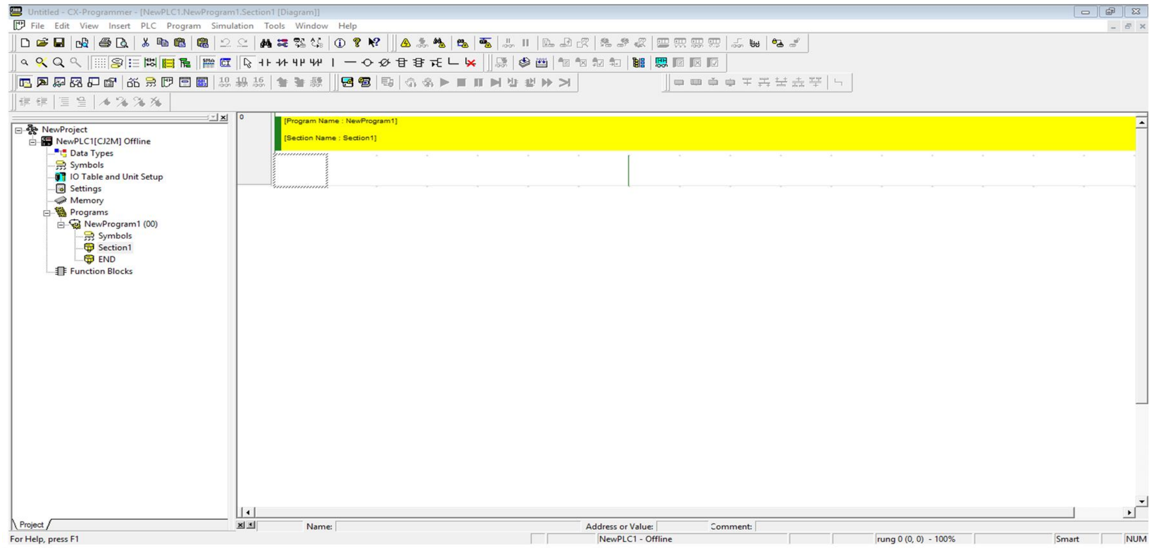

Figure (3.10): New page of the program

Step (05):

This step illustrates the configuration of the input & output

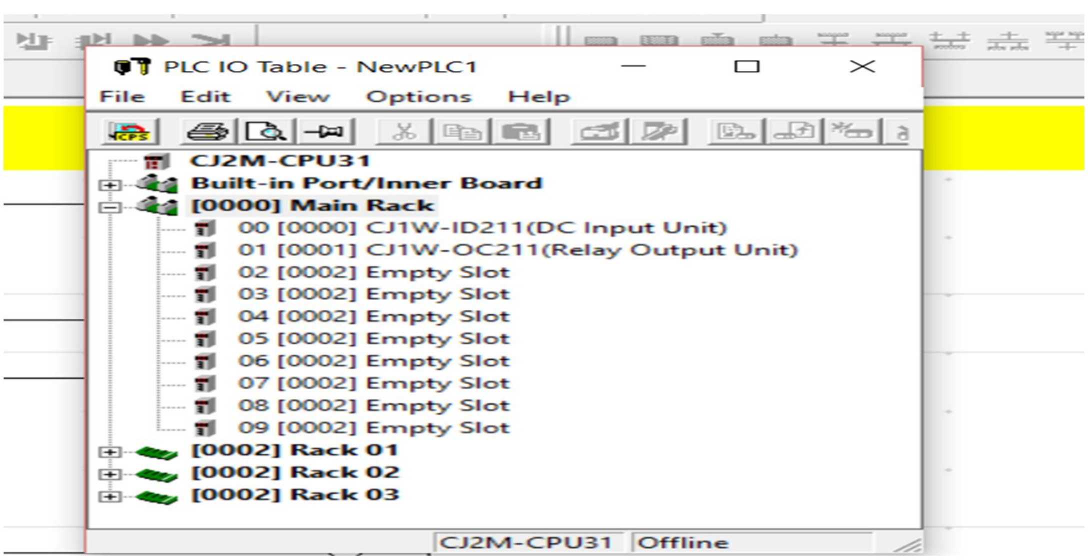

Figure (3.11): Configuration of the input & output

Step (06):

This step illustrate the input output data type & address number

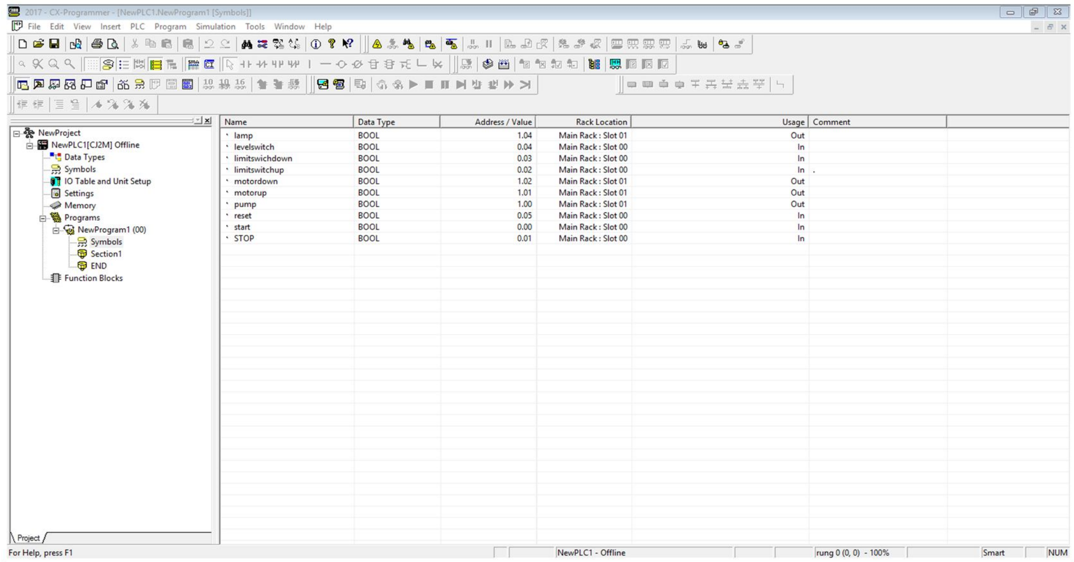

Figure (3.12): Input/output data type & address number

Step (07):

This step illustrate the ladder diagram of the system

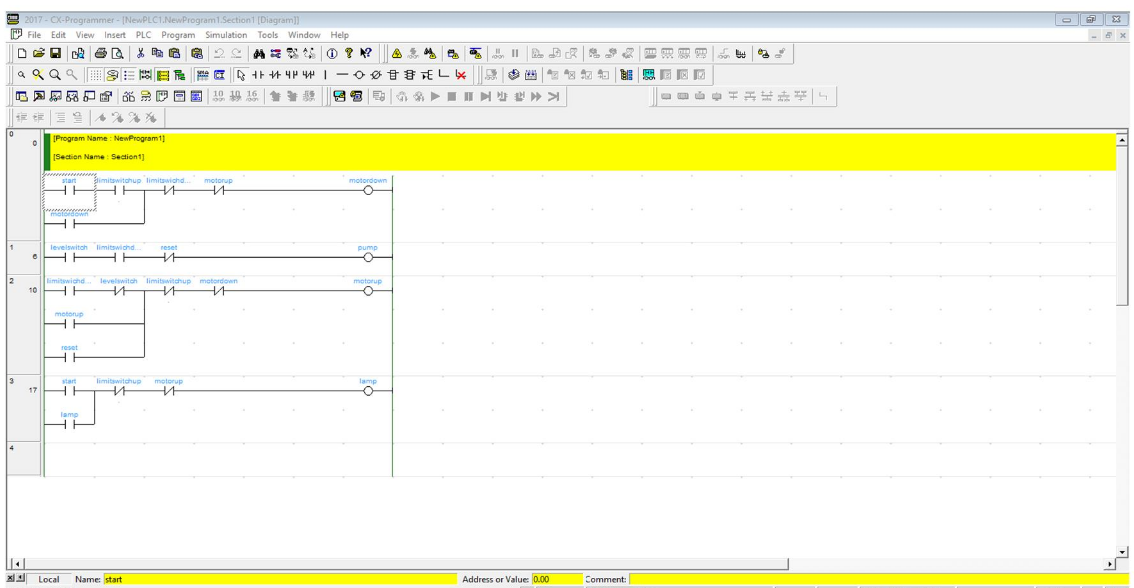

Figure (3.13): Ladder diagram of the system

Step (08):

This step illustrates the system at idle position

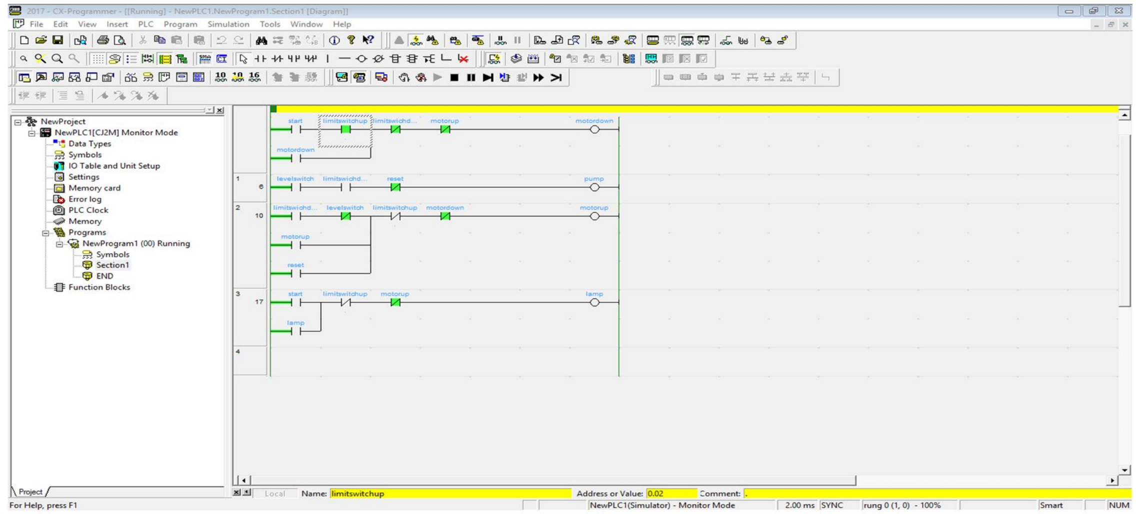

Figure (3.14): System at idle position

Step (09):

This step illustrates the system when the operator presses the start push button

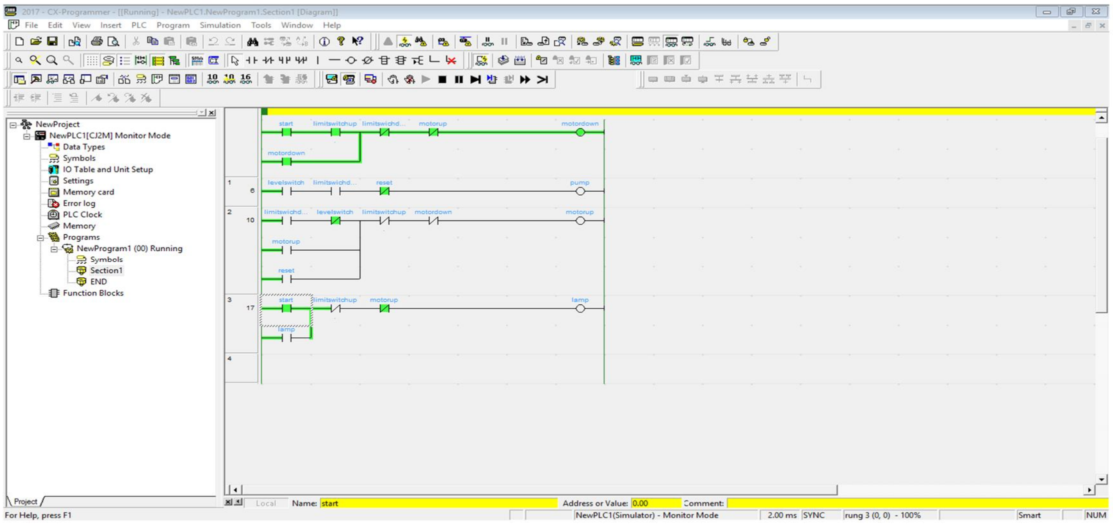

Figure (3.15): System when the operator pressed the start push button

Step (10):

This step illustrate the system when the operator released the start push button

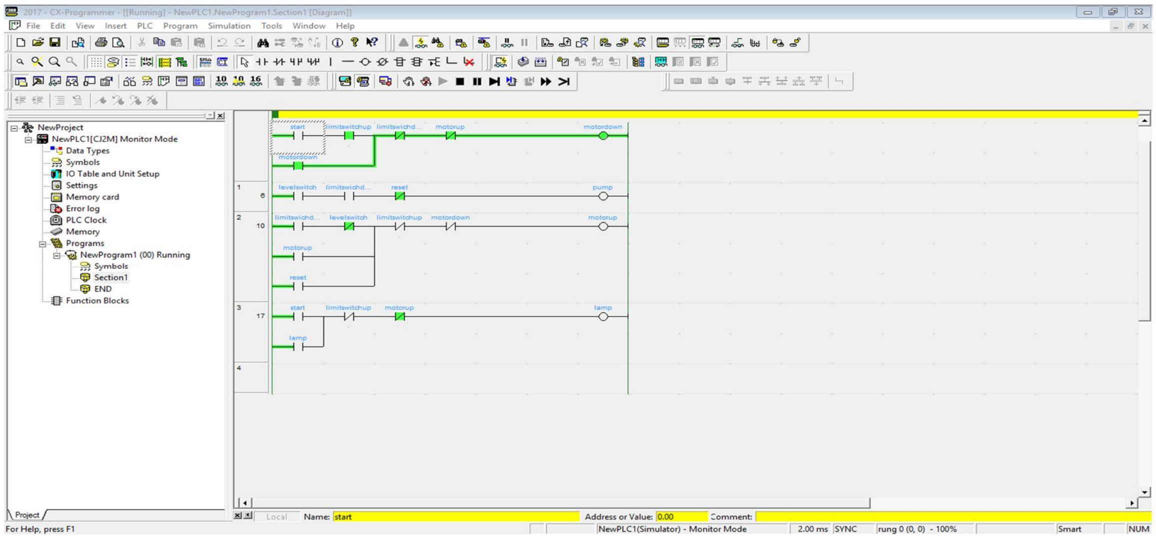

Figure (3.16): System when the operator released the start push button

Step (11):

This step illustrates the system when the motor running and the up limit switch signal disappeared

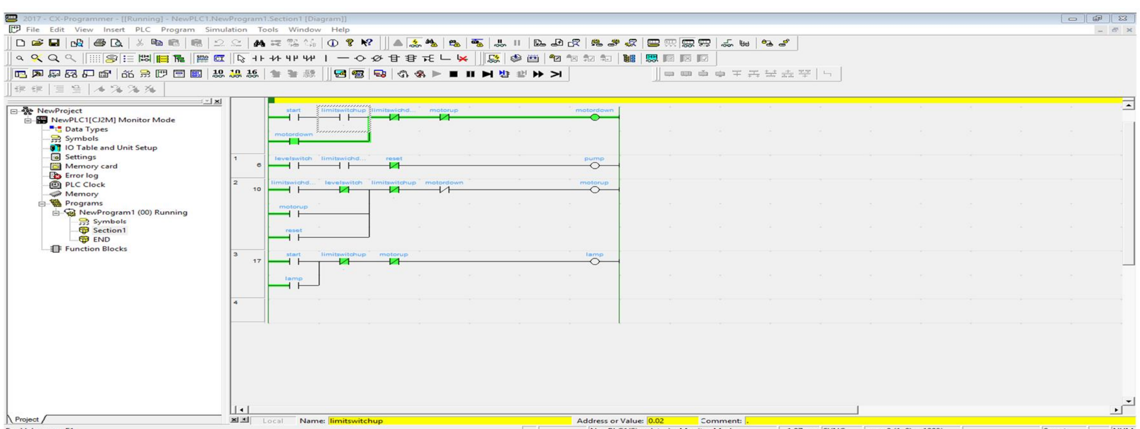

Figure (3.17): System when the motor running and the up limit switch signal disappeared

Step (12):

This step illustrates the level switched signal was received (the pipe touch the liquid surface)

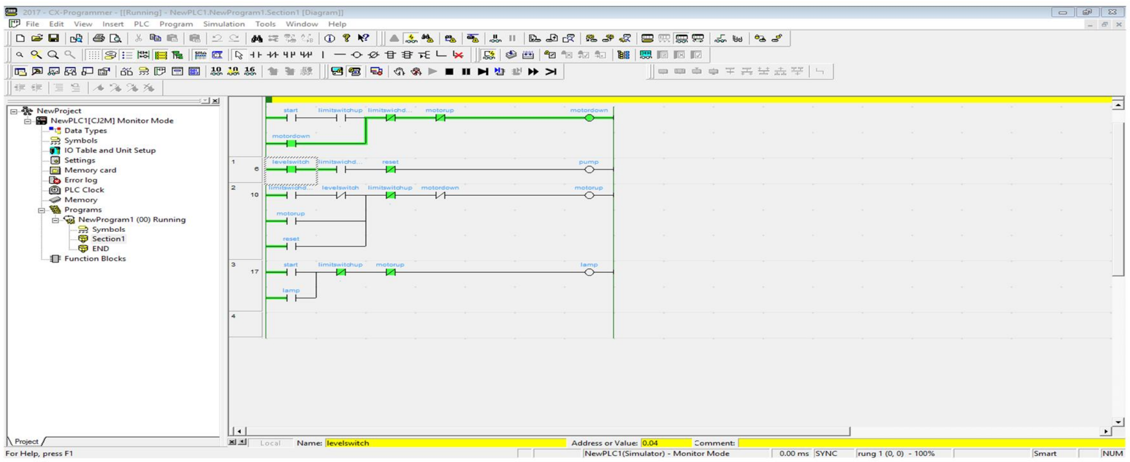

Figure (3.18): The level switch signal is received (when the pipe touches the liquid surface)

Step (13):

This step illustrates the system when both signals (level switch & down limit switch) are received

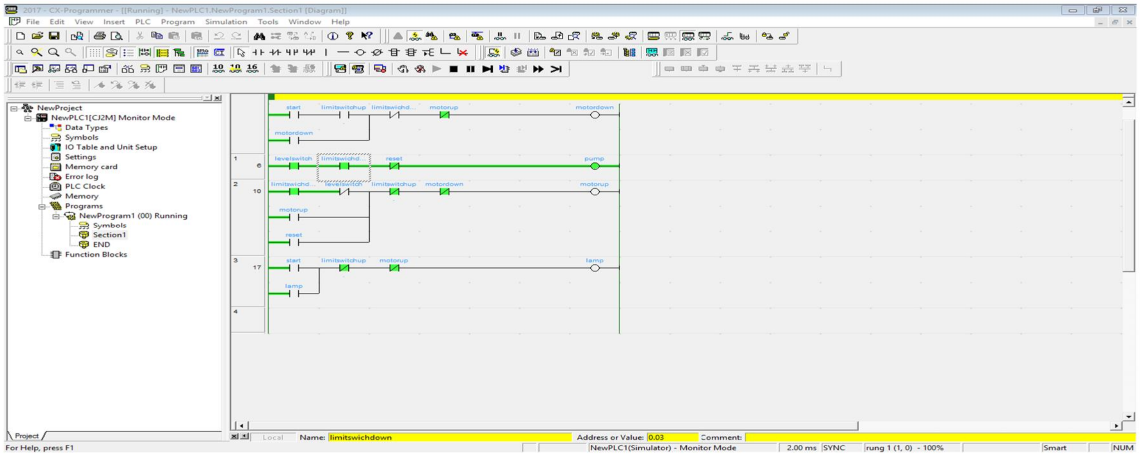

Figure (3.19): System when level switch & down limit switch are received

Step (14):

This step illustrates the system when the target liquid is discharged (level switch signal is disappeared)

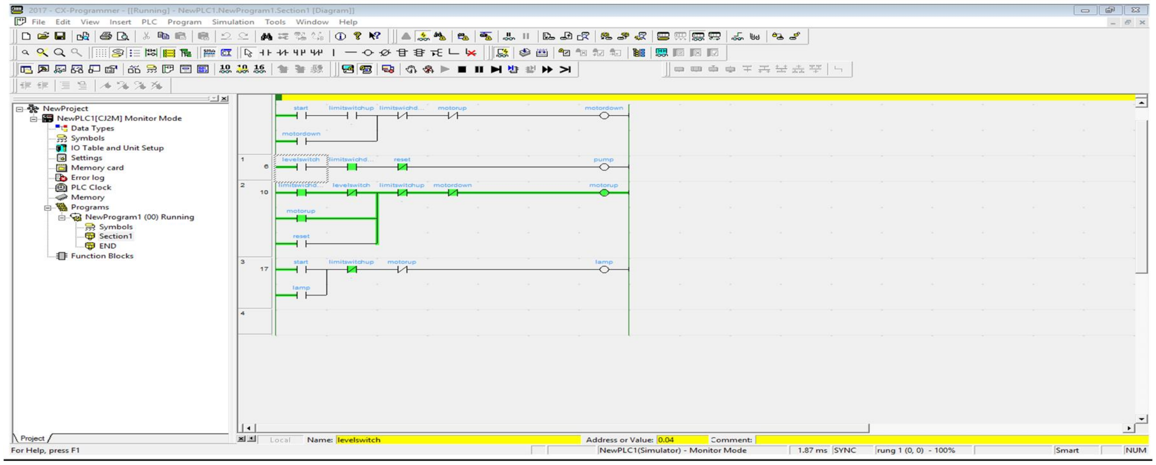

Figure (3.20): System when the target liquid is discharged (level switch signal is disappeared)

Step (15):

This step illustrates the system at moving up period and the down limit switch is disappeared.

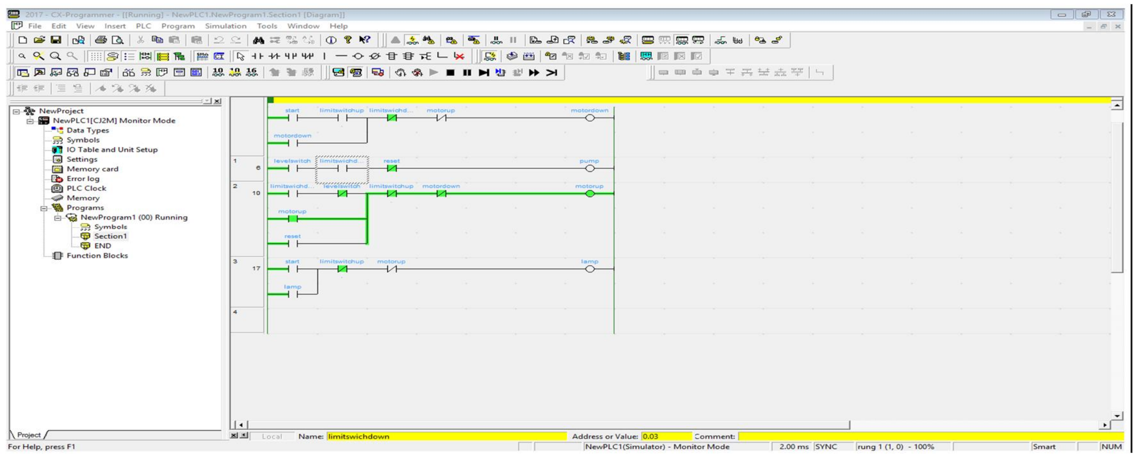

Figure (3.21): System at moving up period and the down limit switch is disappeared.

Step (16):

This step illustrates the system when the up limit switch signal is received as second time (complete the process).

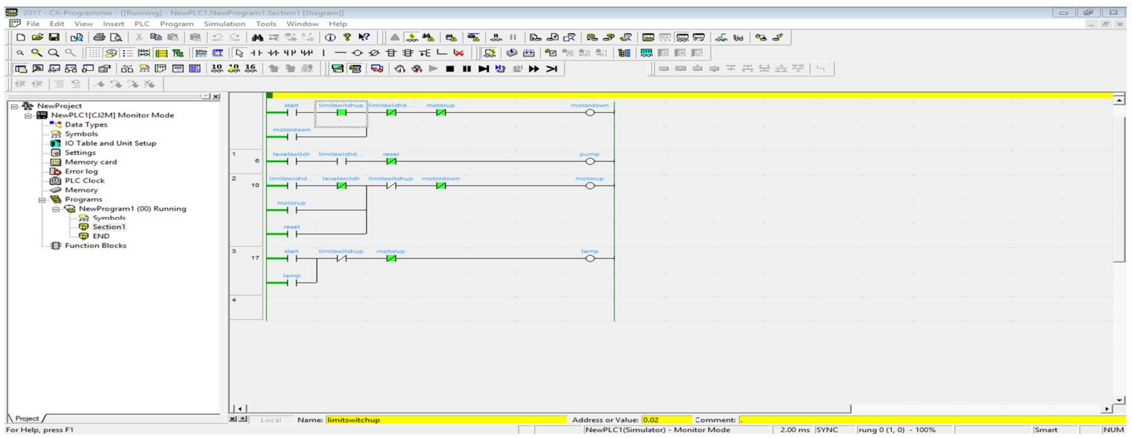

Figure (3.22): System when the up limit switch signal is received as second time (complete the process).

Step (17):

This step illustrates the system when it doesn’t receive limit switch signal (the pump in the middle).

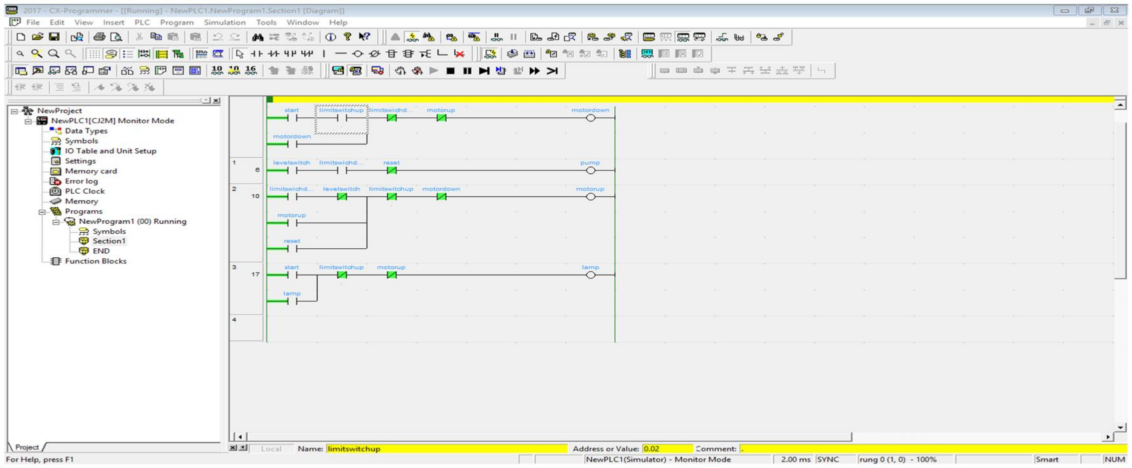

Figure (3.23): System on the pump in the middle.

Step (18):

This step illustrates the system when it doesn't receive limit switch signal as well as the operator press the start push button (fault lamp is light)

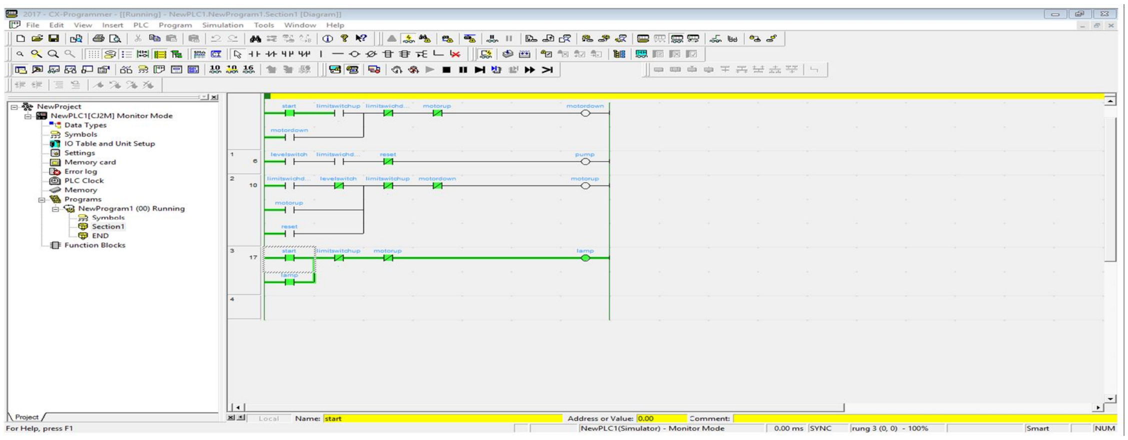

Figure (3.24): System isn't received each limit switch signal as well as the operator press the start push button (fault lamp was light)

Step (19):

This step illustrate the fault lamp is lighting after the start push button is released.

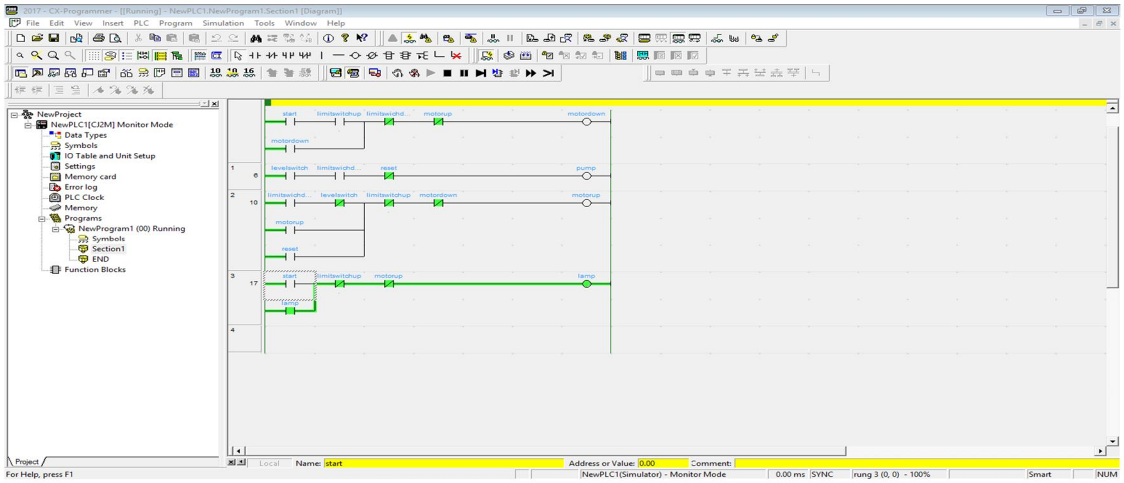

Figure (3.25): The fault lamp is lighting after the start push button is released.

Step (20):

This step illustrate the system returned to ward the idle point when the reset push button is pressed as well as the fault lamp is turned off.

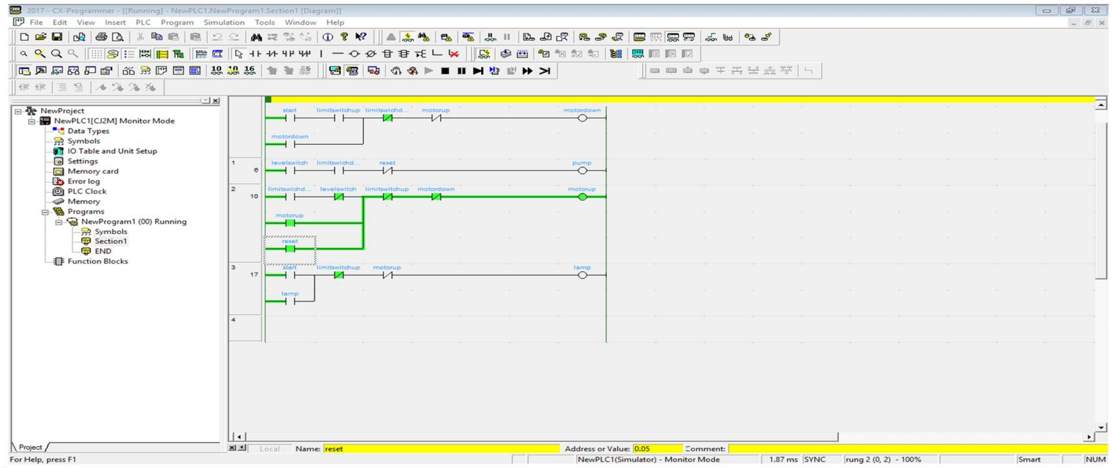

Figure (3.26): System returned to ward the idle position when the reset push button is pressed as well as the fault lamp is turned off.

Step (21):

This step illustrate the system returned to ward the idle position after the reset push button is released.

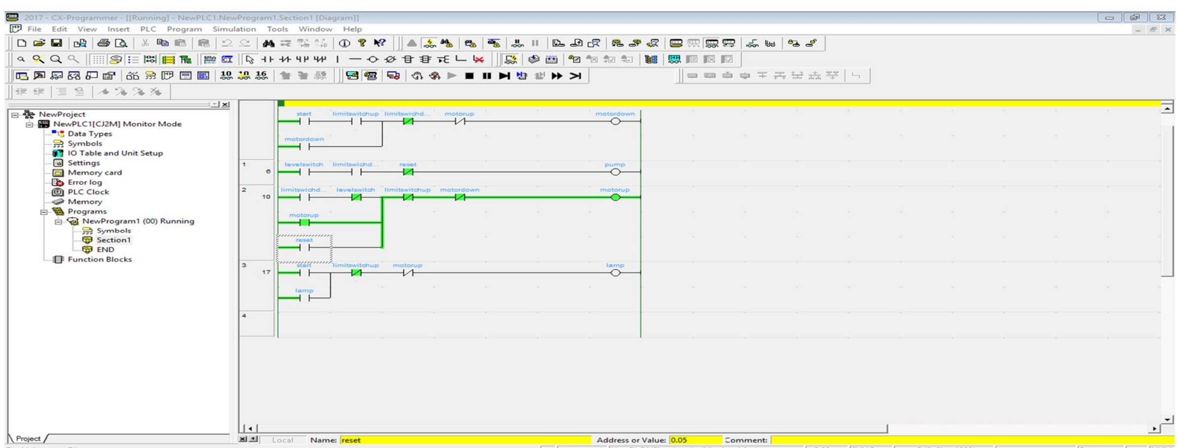

Figure (3.27): System returned to ward the idle position after the reset push button is released.

Step (22):

This step illustrate the system was on ready mode when the pump returns back to the idle point.

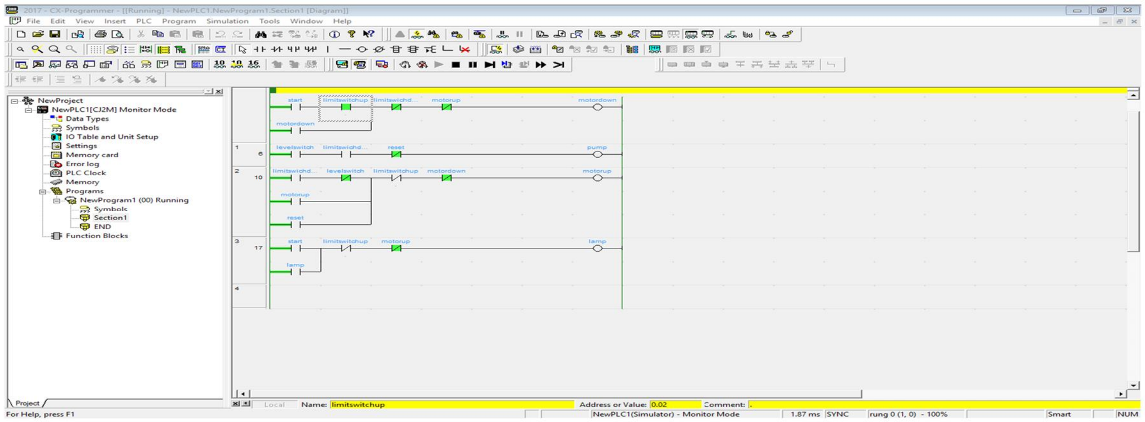

Figure (3.28): System is on ready mode when the pump returns back to the idle point.

Step (23):

This step illustrate the system at the discharging period.

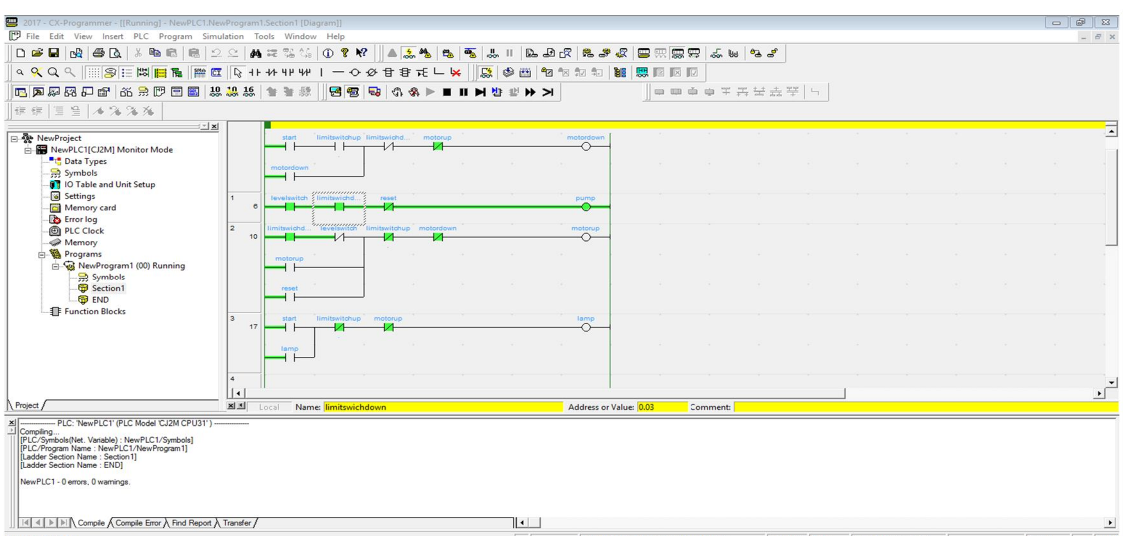

Figure (3.29): System at the discharging period.

Step (24):

This step illustrate the pump is stopped and the system is returns back to idle point when the reset push button is pressed.

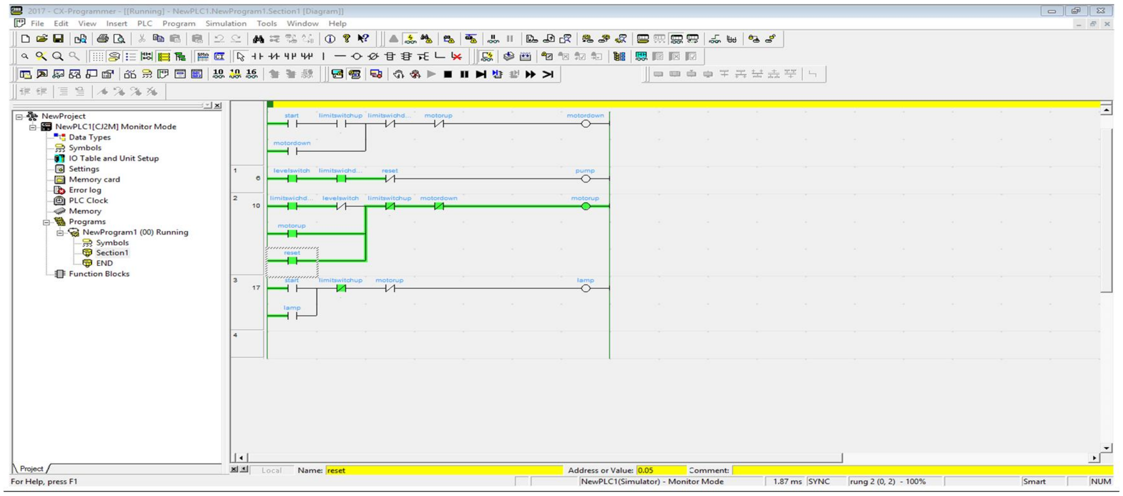

Figure (3.30): Pump is shut off and the system is returns back to idle point when the reset push button is pressed.

Step (25):

This step illustrate the system was still returned back to idle point when the reset push button was released.

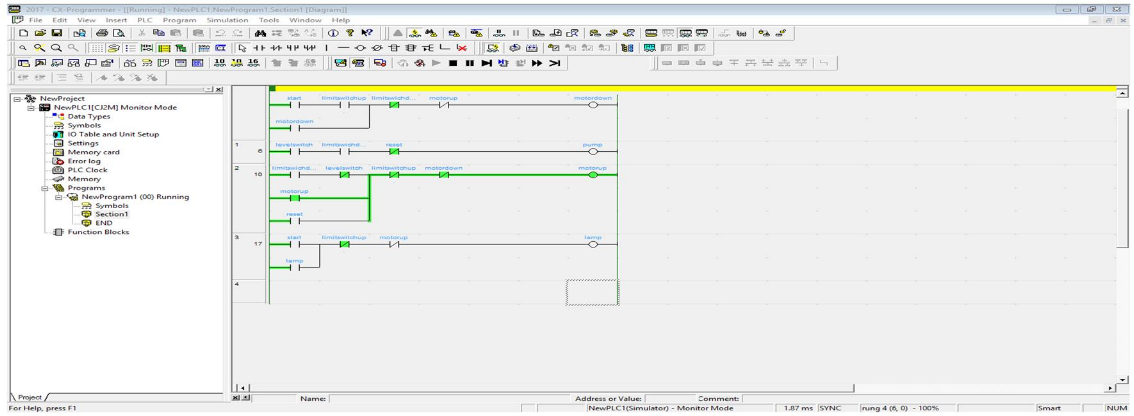

Figure (3.31): System is still returning back to idle point when the reset push button is released.

<u>Step (26):</u>

This step illustrate the system was on ready mode when the pump returned back to the idle point.

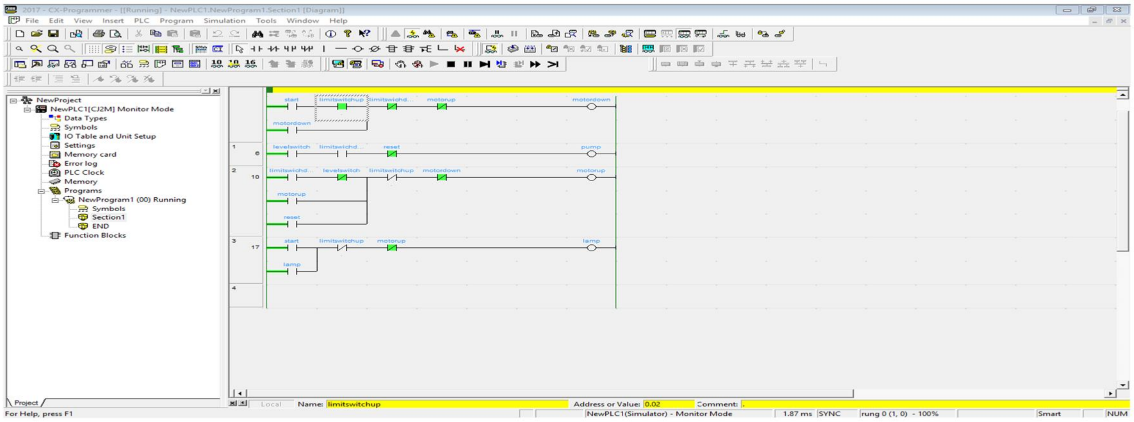

Figure (3.32): System is on ready mode when the pump returned back to the idle point.