Impact Factor (SJIF): 3.632

International Journal of Advance Research in Engineering, Science & Technology

e-ISSN: 2393-9877, p-ISSN: 2394-2444

Volume 3, Issue 5, May-2016

Computer Aided Design of Evaporative Condenser

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Abstract — ***Condenser is an important component of any refrigeration system. In this paper we have considered evaporative cooled condenser, its designing manually as well as using computer program via Scilab. We will note various benefits of this type of condenser along with computer aided design benefits.***

Keyword-Evaporative Condenser, Design of Condenser, Scilab, Program for Evaporative Condenser

I. INTRODUCTION

Evaporative condensers are used in medium to large capacity systems. These are normally cheaper compared to water cooled condensers, which require a separate cooling tower. Evaporative condensers are used in places where water is scarce. Since water is used in a closed loop, only a small part of the water evaporates. Make-up water is supplied to take care of the evaporative loss. The water consumption is typically very low, about 5 percent of an equivalent water cooled condenser with a cooling tower.

However, since condenser has to be kept outside, this type of condenser requires a longer length of refrigerant tubing, which calls for larger refrigerant inventory and higher pressure drops. Since the condenser is kept outside, to prevent the water from freezing, when outside temperatures are very low, a heater is placed in the water tank. When outside temperatures are very low it is possible to switch-off the water pump and run only the blowers, so that the condenser acts as an air cooled condenser.

II. PRINCIPLE[14]

Mollier *i-x* diagram demonstrates schematically how the state of air A (T_A , x_o. i_o) entering a condenser heat exchanger will change along its passage due to heat and mass transfer with the wetted surfaces. The surfaces are maintained at four selected wetted conditions. The state change (1) of the air in a dry heat exchanger is also shown for comparison. The heat transfer to the air under dry operation does not change its moisture content; therefore, the state change will be along the line x_0 until the temperature T_o is reached at an enthalpy value Δi_1 higher than $\Delta i_{0.}$ The temperature T_o is below the condensing temperature T_c to account for the finite overall heat transfer coefficient. The mass flow rate of the air multiplied by Δi, is equal to the rate of heat exchanged.

The state change (2) of the air in the wetted exchanger will again be at T_o, but at the saturated condition with moisture content x_1. The change of enthalpy Δi_2 is n_1 times larger than Δi_1, indicating that for a fixed condenser load either the mass flow rate of the air can be decreased or the air-side heat transfer area can be reduced. It is important to recognize that the heat removed from the condenser in case (2) can be larger than that of the dry heat exchanger even for zero. Heat transfer resistance at the air side is zero, i.e., infinite heat transfer coefficient of the air. This corresponds to the maximum possible augmentation. The same holds true for the state changes (3), (4), and (5).

The state change (3) is assumed to be isothermal along T_A and until saturation is reached at x_2, Δi_3 is n_2 times larger than Δi_1, and again mass flow rate and surface area can be reduced. In addition, two important facts should be noted. First, the higher heat transfer at the outer surfaces of the heat exchanger does not alter the condensing heat transfer internally nor the conductive transport through the walls. The condensing temperature can therefore be lowered to $T_{c1.}$ Second, the temperature of the air T_A is equal to the temperature of the water T_{W2} and is constant along its flow passage. The heat exchange is now equivalent to the exchange between two fluids of infinite heat capacities.

The state change (4) indicates the air being cooled to T_{01}, by moisture increase to saturation at x_3, $\Delta i_4 = \Delta i_1$. This demonstrates that the heat exchanger of the same surface area and the same mass flow rate of the air as for case (1) can now operate at an even lower condensing temperature $T_{C2.}$ Note that the condenser operates now with ambient air entering the heat exchanger at a temperature higher than T_{C2} of the condensing fluid.

In the limiting case (5), the air state changes along a line of constant enthalpy toward saturation at x_4, $\Delta i_5 = 0$ and heat cannot be removed from the condenser.

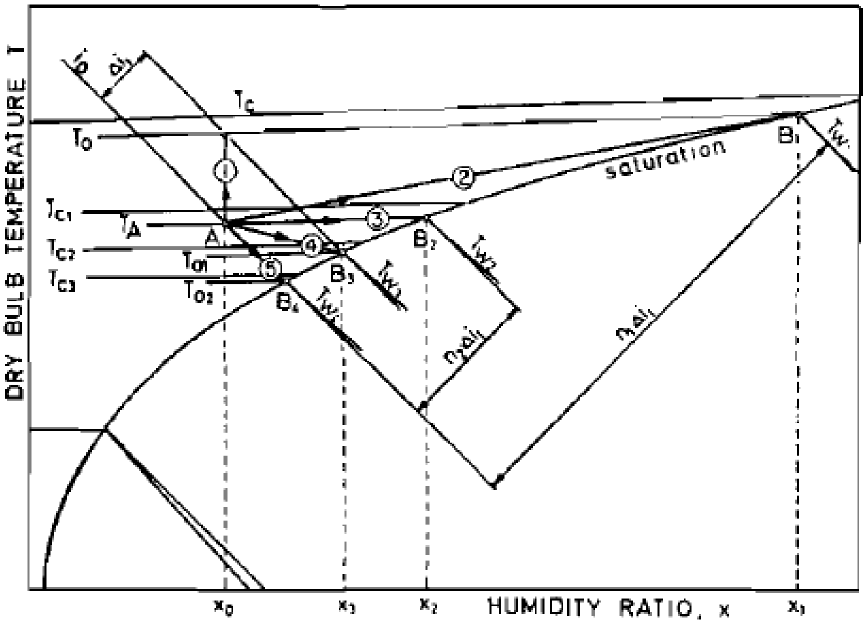

Figure 1: Mollier *i-x* diagram

For the cases (3), (4), and (4) → (5), lower condensing temperatures are feasible, generally to a new value T_c^*. Commonly refrigeration systems are designed to maintain a constant cooling load; therefore the increase of the load resulting from lowering the condensing temperature must be compensated for by a decrease of mass flow rate. This in turn lowers the compressor power and the heat rejected by the condenser. Therefore, the condensing temperature can be lowered further to T_c^{**}, which again could be compensated for as indicated above.

$$\alpha = \frac{i_{R1} - i_{R4}n'}{i_{R1} - i_{R4}(n+1)'}$$

$$[(i_{R2} - i_{R3})\dot{m}_R]_{\text{final}} = (\dot{m}_A \; \Delta i_A)_{\text{final}}$$

The condensing temperature 'T_C' is usually in the neighborhood of the dry bulb temperature of the surrounding or even considerably lower. The removal of heat from the condenser coil becomes impossible when the temperature of the water wetting the surfaces is equal or exceeds the condensing temperature.

The cases (2) → (5) just discussed consume water. The amount of water evaporating is given by the respective products of the mass flow rate of air and the change of moisture content. Despite the increase of the air flow rates (2) → (5), the water consumption decreases with decreasing moisture content of the exiting air.

III. WORKING[13]

The principal components of an evaporative condenser include condensing coil/tube bundle, fan, spray water system, water distribution system, cold water sump, drift eliminators, and water makeup assembly. Figure 2 shows the schematic diagram of an evaporative condenser.

Evaporative condensers use condensing coils or tube bundles fabricated from bare pipe or tubing. Coils are used for low capacity systems and tube bundles are used for high capacity systems. The coil section is arranged in a serpentine fashion, whereas tubes are arranged in a bundle. Compressed refrigerant vapors are condensed inside coil/tube section. This section is supplied with spray headers and nozzles.

The coil/tube bundle is continuously sprayed with water from the top. Water spurge headers are used in the condenser which gives a fine spray of water over the condenser tubes. Fans are mounted in this condenser for providing air. The fan section is one of the most important components for smooth functioning of the condenser. The performance of this condenser is fully dependent upon a sufficient amount of air passing through the coils to carry away the water vapor containing heat.

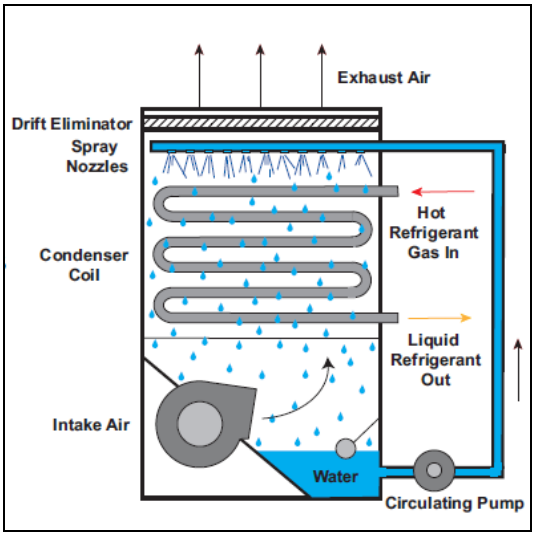

Figure 2: Schematic diagram of Evaporative condenser

The role of air is to increase the rate of evaporation of water. The fan can be either induced draft or forced draft. Natural draft condenser can also be used but it requires regular cleaning and maintenance. The condenser also has eliminator plates placed on top of the condenser. Eliminator plates serve to prevent spray water from being drawn into the fans and discharged to the atmosphere. In other words, they reduce carryout of water droplets from condenser. One of the biggest advantages of evaporative condenser is reduced water consumption. A water storage or water recirculating tank is employed for recirculation of water, thereby reducing water consumption.

IV. ADVANTAGES AND DISADVANTAGES[13,6]

The various advantages of evaporative cooled condenser are as follows:

1. As minimum driving force or heat transfer required in an evaporative cooling type condenser is 1°, while the same for water cooled condenser (shell and tube heat exchanger) 3-5 °C and for air cooled condenser (finned tube heat exchanger) 10-15 °C.
2. To reduction in compression ratio i.e. reduction in power consumption of refrigeration cycle for the given refrigeration duty or it is resulted in the improvement in the value of COP.
3. This modification eliminates to use of separate piping required for cooling tower for connecting cooling tower and condenser of refrigeration cycle.
4. This modification decreases a pressure drop in cooling water circuit. Hence, it decreases a power consumption of cooling water circulating pump.
5. Evaporative condensers reduce the water pumping and chemical treatment requirements associated with cooling tower / refrigerant condenser system.
6. In addition they require substantially less fan power than the air cooled condenser of comparable capacity.
7. The evaporative condenser can operate at a lower condensing temperature than the air cooled condenser.
8. Compared to Shell & Tube Condensers and cooling tower, about 15% electrical consumption can be saved due to low pressure condensation of refrigerant.
9. Lower energy consumption.
10. Saving of investment cost.
11. Smaller plan area.
12. Smaller driving power for compressors.
13. It decreases to compression ratio require for vapor compression refrigeration cycle.

The various disadvantages of evaporative cooled condenser are as follows:

1. Reliable design co-relations are not available.
2. The water may freeze at low operating temperatures.

3. Impurities in the vapors may cause corrosion.
4. Cost of water consumption and water treatment method the saving on total cost may not be significant.
5. Spread of Legionnaires disease. Precautionary measures needed.
6. Air intakes of ventilation and air conditioning system
7. Away from them due to water mist and noise's emission.
8. A small amount of the cooling water must be continually purged to prevent the build-up of contaminants.

V. COMPUTER AIDED DESIGN OF EVAPORATIVE CONDENSER

CASE STUDY - 1

An evaporative condenser is considered for condensing ammonia. As shown in below figure evaporative condenser is basically a rectangular tube bundle installed in cooling tower. Cooling tower falls over the tubes at a required rate at essentially a constant temperature; 4.5 °C above the design wet bulb temperature of air. Ambient air at 40 °C dry bulb temperature and 28 °C wet bulb temperature (WB) is supplied from sides of the tower. An induced draft fan sucks the air from the louvers and discharges it from top with 95% relative humidity (RH). Ammonia gas enters at 120 °C the tubes and condenses at 1.5 °C higher that the cooling water temperature.
Design the tube bundle to be placed in the evaporative condenser. For the evaporation of water film over tube bundle, heat transfer coefficient can be determined by Kallam's equitation.

$h_o = 7.3 \times 10^{-9} \times N^{0.05} \left(\frac{G^{0.3}\ YZ\ d_o}{\mu P_v}\right)^{4.4}$ where, h_o = heat transfer coefficient, W/(m^2. °C). Here, evaporation temperature of water film can be taken as 34 °C. evaporation rate of water can be taken as 2% of circulation rate. [3]

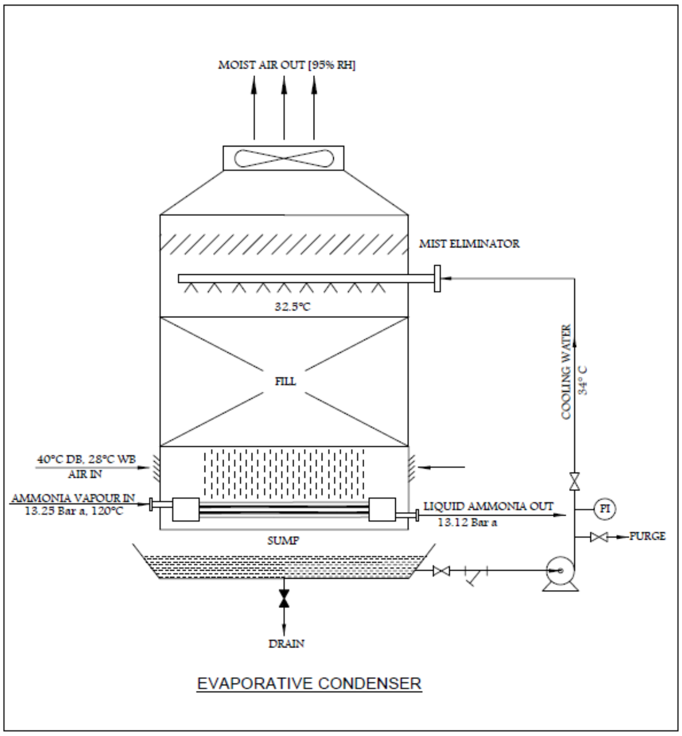

Figure 3: Schematic Diagram of Evaporative Condenser

CASE STUDY – 2

10, 900 kg/h of nearly pure saturated methyl ethyl ketone vapor at 13.73 kPa g is to be condensed and cooled to 60 °C by cooling water which is available in plant at 32 C. consider pressure drops of 13.7 kPa for vapor and 68.7 kPa for the water as permissible. Design the suitable shell and tube heat exchanger for the heat duty.[3]

The program structuring of evaporative condenser done with help of Scilab programming software. The result is obtained from the programming as follows:

Table 1: Design data of Evaporative Condenser

Parameters to be calculated	Actual Literature Data of Case Study-2		Calculated Literature Data for Case Study-2 using Scilab	Manually Calculated Data for Case Study-1	Structured Program Data for Case Study-1 using Scilab	Units
Heat duty for desuperheating zone	ϕ_{des}	166	166.084	155.303	155.307	kW
Heat duty for condensation zone	ϕc	1327	1326.99	742.45	742.469	kW
Total heat duty of condenser	ϕ	1493	1493.078	897.45	897.776	kW
Mass flowrate of cooling water	$\dot{m}_w$	44.5746	44.5769	0.37	0.37026	kg/s
Tube side heat transfer coefficient	hi	7155.5	7159.7396	370.45	370.467	$W/m^2.°C$
Shell side heat transfer coefficient for desuperheating zone	h_{co}	1920.9	1864.6703	6556.2	6556.84	$W/m^2.°C$
Shell side heat transfer coefficient for condensation zone	h_o	283.77	1864.6703	12071	12070.8	$W/m^2.°C$
Overall heat transfer coefficient for desuperheating zone	U_{ode}	237.72	237.722	248.5	248.508	$W/m^2.°C$
Overall heat transfer coefficient for condensation zone	U_{oc}	831	820.41385	1327.4	1327.45	$W/m^2.°C$
Mean temp. difference for desuperheating zone	ΔTm_{de}	38.349	38.349	21.15	21.1501	°C
Mean temp. difference for condensation zone	ΔTm_c	47.336	47.336	1.5	1.5	°C
Area required for desuperheating zone	Ar_{de}	18.21	18.218	29.55	29.5487	m^2
Area required for condensation zone	Ar_c	33.73	34.1699	372.88	372.879	m^2
Total area required	A_r	51.94	52.388002	402.43	402.428	m^2
Area provided	A_{pr}	61.95	63.156	457.64	457.41	m^2
Excess heat transfer area		19.27%	20.55%	13.72%	13.66%	Acceptable

VI. CONCLUSION

From the results we can conclude that the program made is well suited for any system incorporating Evaporative Condenser, and has been validated with two different case studies. The result of a case study has been validated with the literature data, satisfying all the conditions.

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VIII. APPENDIX

Given Inputs:

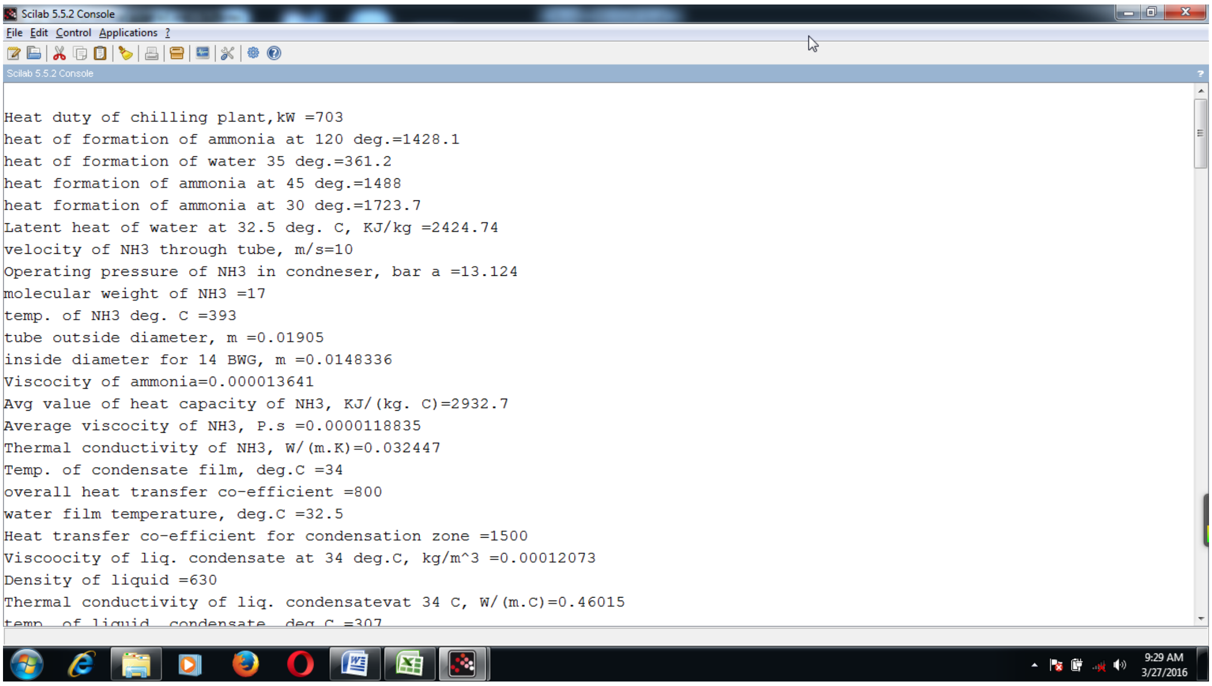

Figure 4: Output Window – 1

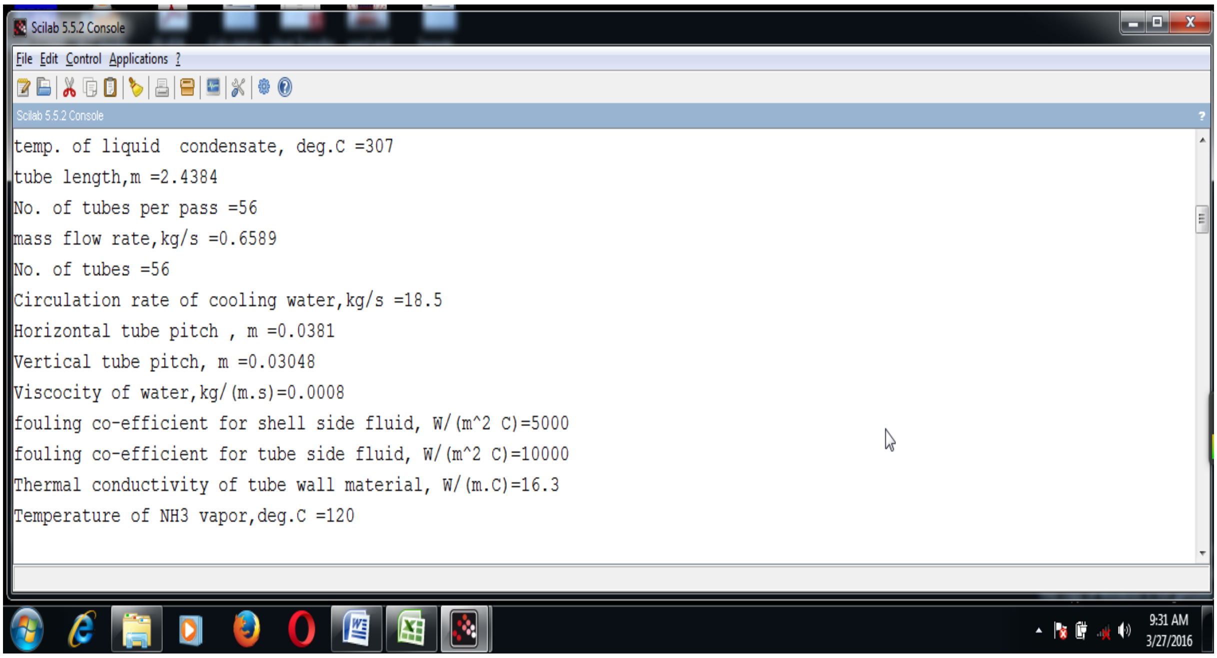

Figure 5: Output Window – 2

OUTPUT WINDOWS:

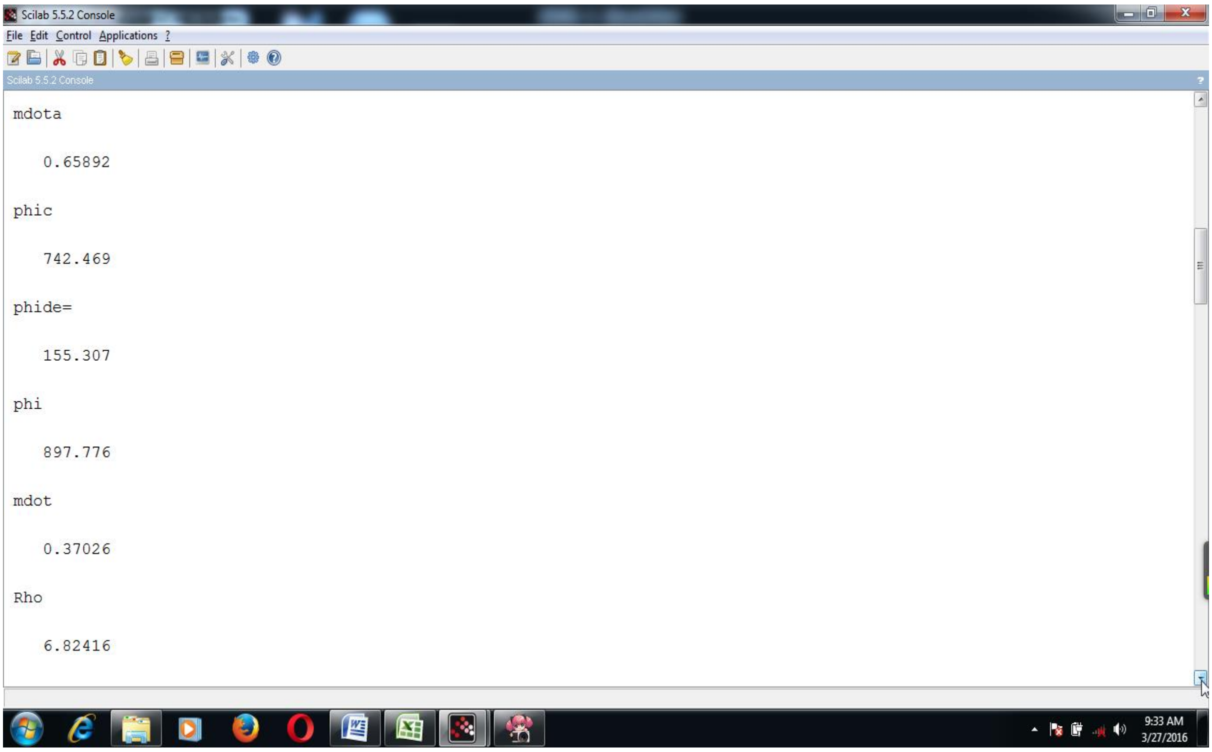

Figure 6: Output Window – 3

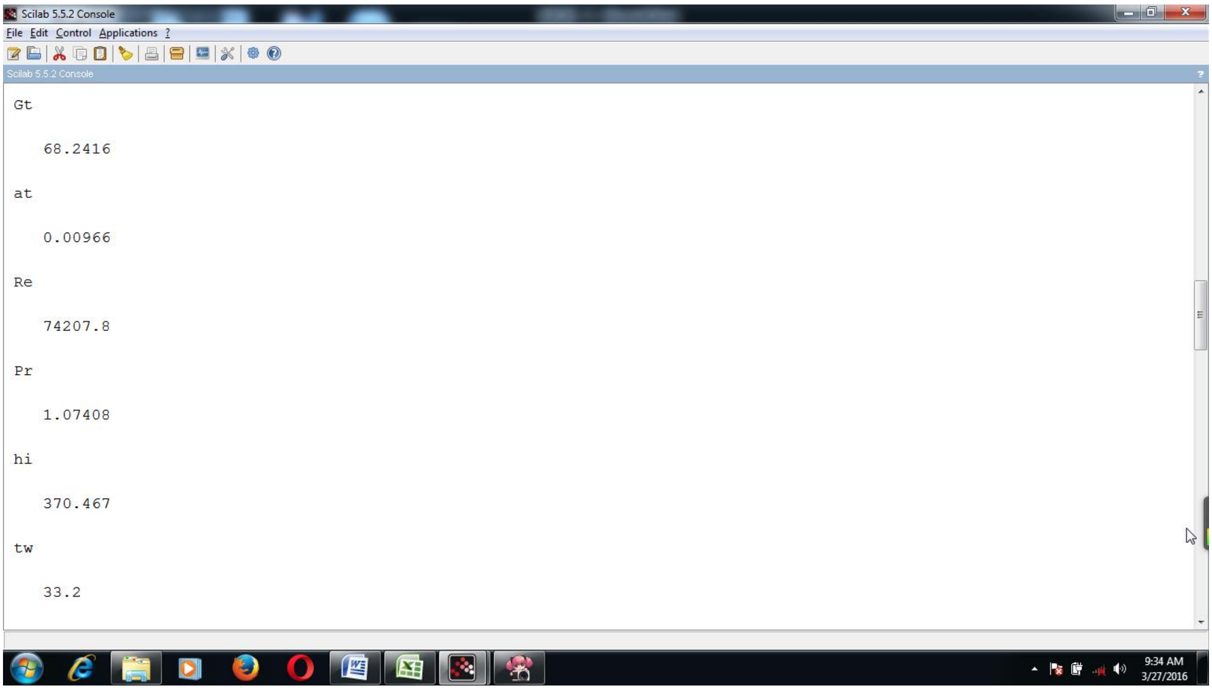

Figure 7: Output Window – 4

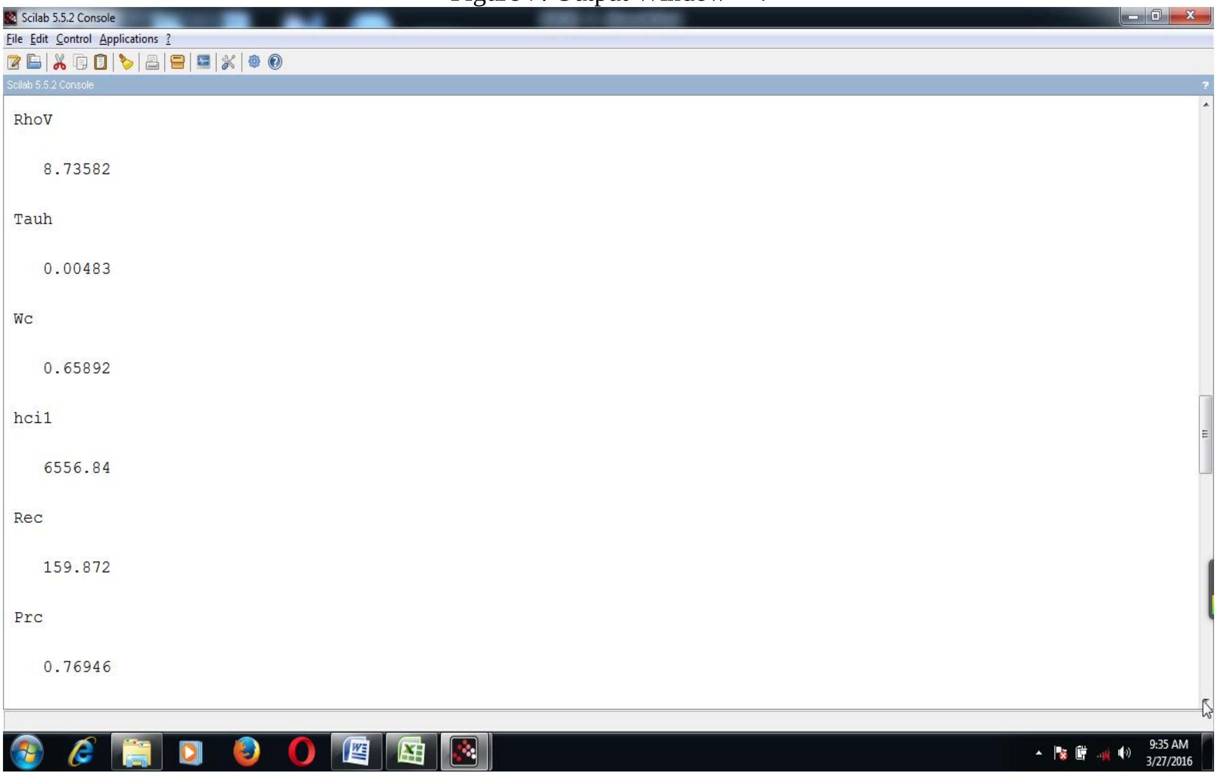

Figure 8: Output Window – 5

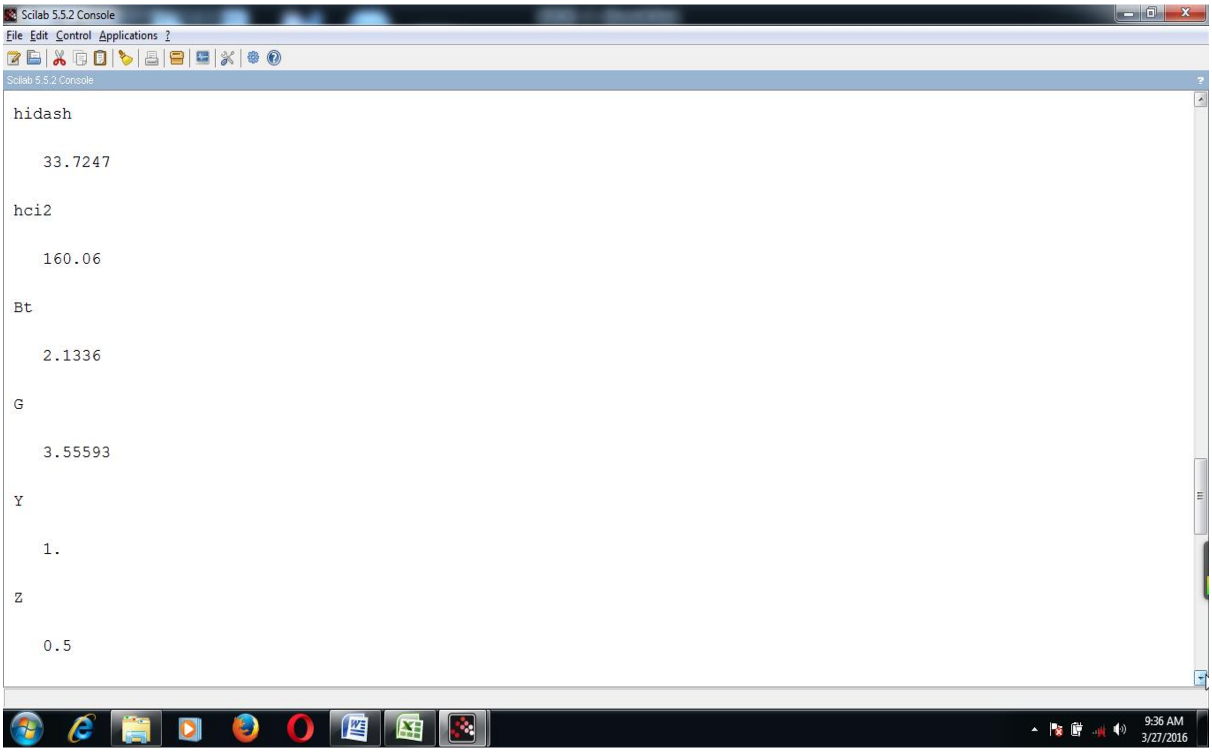

Figure 9: Output Window – 6

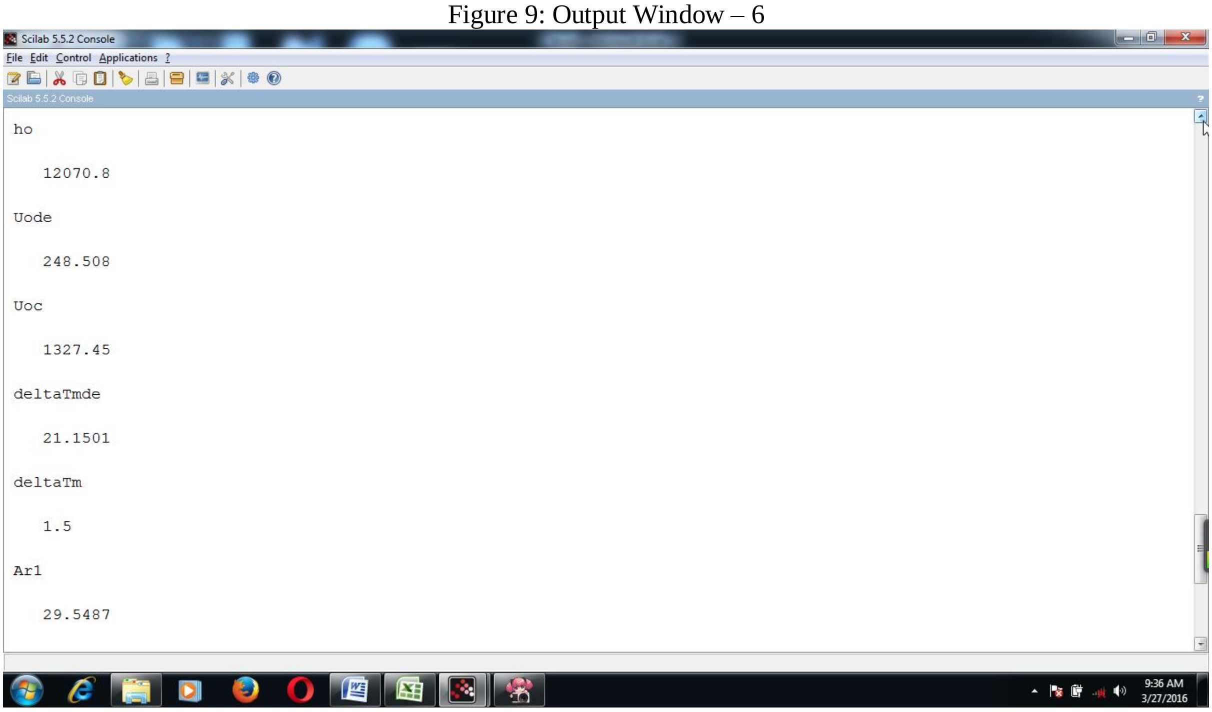

Figure 10: Output Window – 7

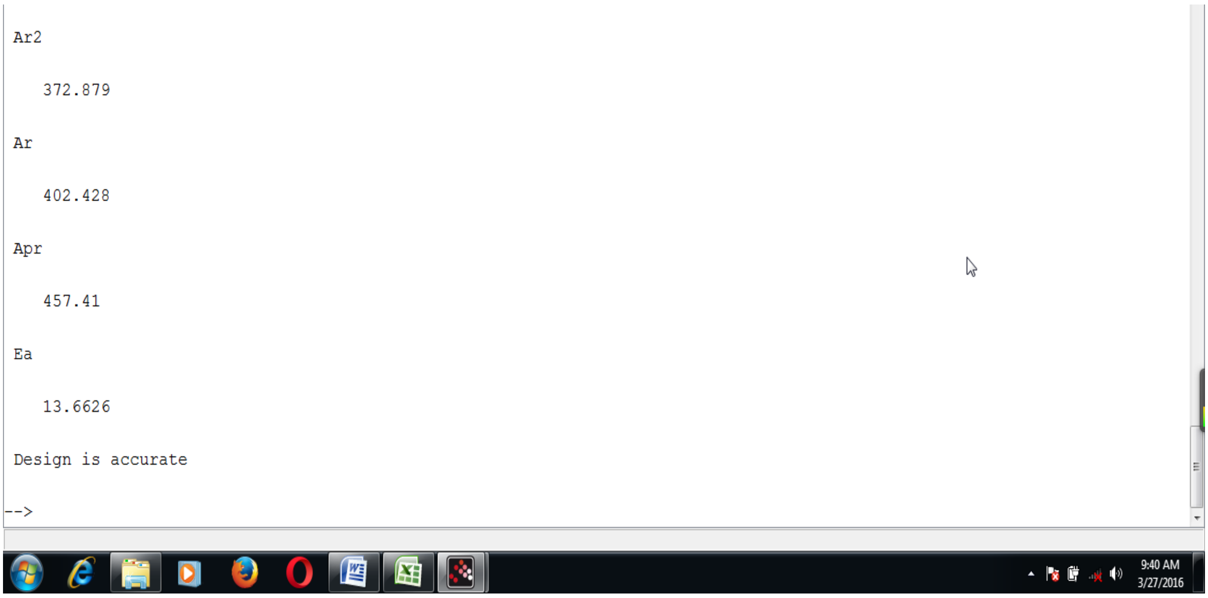

Figure 11: Output Window – 8