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Studies on Effect of Varying Geometric Parameters of Solar Receiver Tube on Thermal Loss Suffered By It

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Abstract

The radiative and convective heat losses from a solar receiver tube at a given operating temperature are a function of temperature of the outer glass envelope and outer surface area of the glass envelope. But the temperature of the outer glass envelope and its outer surface area are themselves a function of outer diameter of the heat pipe enclosed within the glass envelope and the annular gap between the outer surface of the heat pipe and the inner surface of the glass envelope. Hence the thermal losses associated with the receiver tube are a function of the size (outer diameter) of the heat pipe, annular gap between the glass envelope and the heat pipe, the operating temperature of the receiver tube. Theoretical thermal model is developed that describes various heat transfer phenomenon taking place in the solar receiver tube, under valid physical assumptions. Theoretical heat loss investigation is conducted by varying the values of outer diameter of the heat pipe (25–75 mm) and annular gap (25–75 mm) at three different operating temperatures of the receiver tube: 150, 300 and 450°C respectively, employing the thermal model developed. The radiative, convective and hence total losses associated with a solar receiver tube are quantified. The effect of varying the geometric parameters of the receiver tube on the heat loss suffered by it is studied.

Keywords: *Solar receiver tube, operating temperature, thermal losses, annular gap*

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INTRODUCTION

Solar receiver tube is that component of a concentrating solar collector that is responsible for absorption of solar energy and its conversion into heat. A typical concentrating solar collector consists of a reflector and a heat collecting element (Solar Receiver Tube) placed at the reflector's focus. A heat transfer fluid circulates through a metal tube (absorber tube) with an external selective surface that absorbs solar radiation reflected from the mirror surfaces of the reflector. In order to reduce the heat losses, the absorber tube is covered by an envelope and the enclosure is usually kept under vacuum pressure (<1 Torr) [1].

The Solar receiver tube consists of an absorber tube surrounded by a glass envelope. The absorber tube is typically a stainless steel tube with a selective radiation absorbing surface. Selective surfaces ensure a high absorbance for incoming solar radiation and low emittance for infrared radiation at the temperature range in which the surface emits radiation shown in Figure 1. The glass envelope protects the absorber tube from environmental degradation and reduces heat losses suffered by it. The vacuum enclosure is used primarily to reduce heat losses at high operating temperatures and to protect the solar selective absorber surface from oxidation. The receiver tube uses conventional glass to metal seals with metal bellows at either end to achieve the necessary vacuum enclosure and to account for thermal expansion difference between the steel tubing and the glass envelope [1].

Absorber tube of a typical solar receiver tube is made up of a material having high thermal conductivity, in general copper or stainless steel. The absorbing surface is coated with a solar selective coating to obtain high solar absorbance. Some commonly employed coatings are black nickel, black chrome, black copper oxide and black cobalt oxide.

The most successful and stable coating so far developed is black chrome [2]. The glass envelope is made up of borosilicate glass with low iron content which keeps it highly transparent to incoming solar light (visible range) and almost opaque to infrared radiation escaping out of the absorber tube. The outer surface of the envelope is provided with an antireflecting coating to minimize reflection suffered by the incoming solar radiation [3]. Heat transfer fluids (HTFs) are working fluids in the solar receiver tube. It is primarily water. Improvements in design and higher operating temperatures in concentrating solar power plants have necessitated the use of other HTFs like molten salts, thermic oil, and ionic fluids in medium temperature operations (up to 500°C) [4] and HTFs like solid particle receivers (solid), small air particle receivers (air), and falling film receivers in high temperature operations (up to 700°C) [5]. The geometry (size) of the solar receiver tube is defined by the diameter of absorber tube, the annular gap between the outer surface of the absorber tube and inner surface of the glass envelope, the length of the receiver tube. The thermal losses suffered by the receiver tube in the form of radiative and convective heat transfers from the outer surface of the glass envelope to the surrounding atmosphere depends on the above mentioned geometric parameters of the receiver tube. Since radiative and convective heat transfers from a cylindrical surface are directly proportional to length of the cylindrical surface, the thermal loss suffered by the solar receiver tube can be ascertained to be proportional its length. The variation of thermal loss with the other two geometric parameters is complex to analyze. Thermo-mathematical modeling of the solar receiver system and solving the same is required to analyze effect of variation of other two geometric parameters on thermal losses suffered by the receiver tube. Analyzing the same would help in designing the receiver tube with optimum size for desired applications.

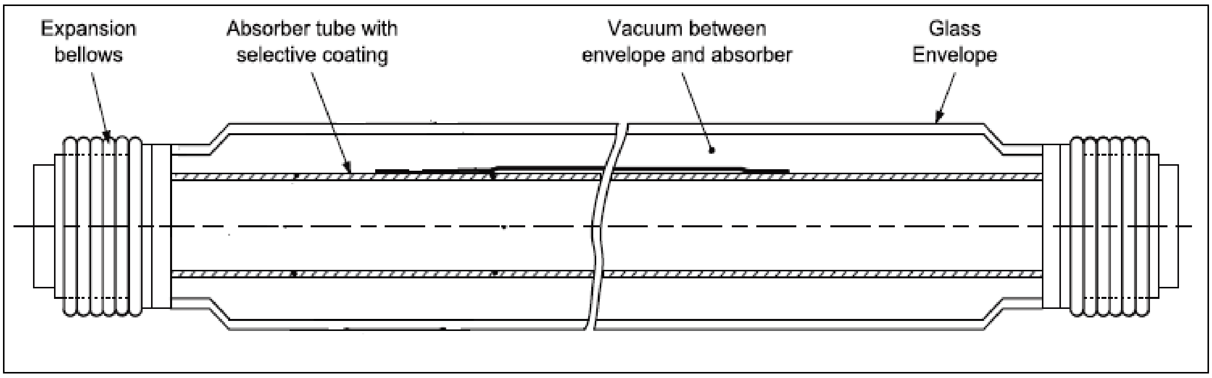


Fig. 1: *Schematic of a Solar Receiver Tube* [1].

THERMO-MATHEMATICAL MODELING

Assumptions

1. Steady state, 1-dimensional heat transfer (in radial direction of the tube only).
2. Variation of radiative properties with temperature is negligible in the operating temperature range.
3. Conduction heat loss from heat pipe to surroundings through metal bellow and supporting bracket is negligible.
4. The temperature difference between inner and outer surface of the glass envelope is negligibly small so that it can be analyzed as a system at single steady temperature.

   Figure 2 shows the cross section of the receiver tube with associated dimension.

Nomenclature

A=surface area m^2
b=interaction coefficient
d=diameter of the heat pipe m
D=diameter of the glass envelope m
h=convective heat transfer coefficient W/m^2-K
l=length of the receiver tube m
P=annulus pressure Pa
Q=heat transfer W
T=temperature °C

Greek letters

α=absorptivity of surface
ϵ=emissivity of surface
λ=mean free path m
σ=Stefan-Boltzman constant=$5.67 \times 10^{-8} W/m^2\text{-}K^4$
τ=transmittivity of surface

Subscripts

a: Heat pipe
e: Glass envelope
i: Inner
o: Outer
s: Surrounding atmosphere
a-e: heat transfer from heat pipe to glass envelope
e-s: heat transfer from glass envelope to surrounding atmosphere
rad: radiation
conv: convection

The thermal loss is investigated by considering the outer glass envelope as a control mass at steady state.

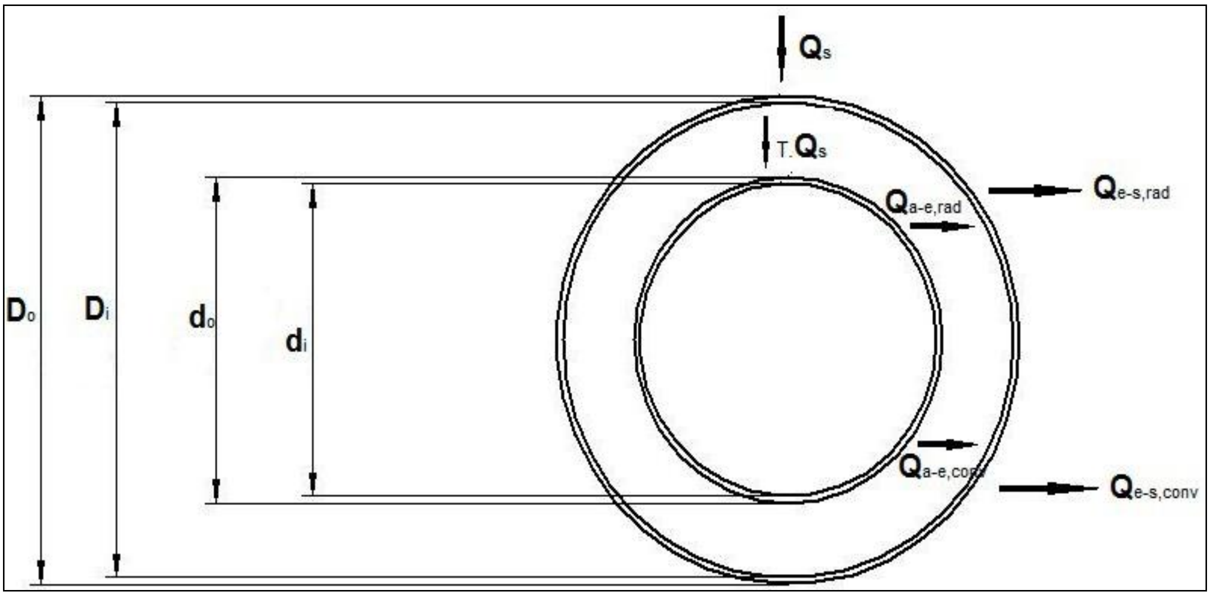


Fig. 2: *Figure Showing Dimensions and Heat Transfers Involved in the Receiver Tube Drawing Tool: Solid Edge V19.*

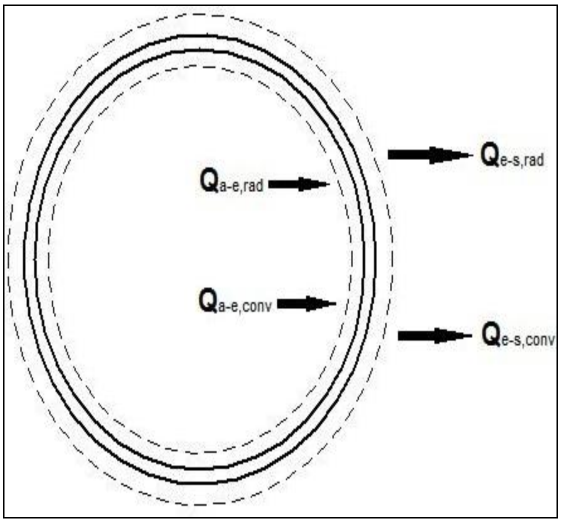


Fig. 3: *Control Mass Model for the Glass Envelope of the Receiver Tube with the Optical Loss Term $\alpha_e.Q_s$ Neglected, Drawing Tool: Solid Edge V19.*

First law of thermodynamics for the control mass at steady state gives,

$$Q_{a-e,rad}+Q_{a-e,conv}=Q_{e-s,rad}+Q_{e-s,conv} \quad (1)$$

Radiative Heat Transfer from Heat Pipe to Glass Envelope $Q_{a-e,rad}$

The net radiative heat transfer from the outer surface of the heat pipe to the inner surface of the glass envelope is given by,

$$Q_{a-e,rad}=\sigma.A_a.Ƒ_{a-e}[(T_a+273)^4-(T_e+273)^4] \text{ [6]. } (2)$$

Where; $Ƒ_{a-e}$ is the radiation transfer factor between the outer surface of the heat pipe and the inner surface of the glass envelope.

Assuming the heat pipe and the glass envelope to form long, concentric cylinders, $Ƒ_{a-e}$ is given by,

$$Ƒ_{a-e}=[1/\epsilon_a+(1-\epsilon_e).d_o/(\epsilon_e.D_i)]^{-1} \text{ [6]}$$

Convective Heat Transfer from Heat Pipe to Glass Envelope $Q_{a-e,conv}$

The diagrammatic view of control mass model for the glass envelope is shown in Figure 3. The convective heat transfer phenomenon at pressures less than 1 Torr is relatively complex. Gas flow regimes and hence convective heat transfers are characterized by the nature of the gas and by the relative quantity of gas present in the enclosure. Gas flow regimes are classified based on the Knudsen number which is the ratio of mean free path between the gas molecules to the characteristic length of the enclosure. Gas flow regimes are classified into the following three major regimes based on the Knudsen number:

Table 1: *Flow Regimes based on Knudsen Number [7].*

Flow Regime	Knudsen Number (Kn)
Continuum flow	Kn<0.01
Transition flow	0.01<Kn<1
Molecular flow	Kn>1

When pressure in the annular gap is less than 1 Torr, the mean free path encountered is in the order of 10^{-1}–10^{-2} m, the flow regime is essentially molecular.

The heat transfer coefficient in this regime is given by,

$h_{a-e}=k_{std}/[(d_o/2).\ln(D_i/d_o)+b\lambda(d_o/D_i+1)]$ W/m^2–K [8]

Where k_{std}=thermal conductivity of air at standard 1 atm. pressure,

b=interaction coefficient, for air-glass surface interaction b=1.571 [8],

λ=mean free path of collision between air molecules given by,

$\lambda=R_o.T/(\sqrt{2}.\pi.d^2.N_A.P)$ m,

R_o=universal gas constant=8.314J/mole–K,

T=absolute temperature of the air K,

d=molecular diameter of air=3.53x10^{-10} m [8],

N_A=Avagadro number=6.023x10^{23}/mole,

P=annulus pressure Pa

The convective heat transfer from outer surface of the absorber to the inner surface of the glass envelope is given by,

$$Q_{a-e,conv}=h_{a-e}.A_a.(T_a-T_e) \quad (3)$$

Radiative Heat Transfer from Glass Envelope to Surroundings $Q_{e-s,rad}$

The gas molecules and the suspended particles in the surrounding atmosphere emit radiation as well as absorbing it. Although this emission least resembles the distribution of radiation from a blackbody, it is found convenient in radiation calculations to treat the atmosphere as a blackbody at some lower fictitious temperature that emits an equivalent amount of radiation energy. This fictitious temperature is called the effective sky temperature T_{sky}.

The value of T_{sky} depends on the atmospheric conditions. It ranges from about 230 K for cold, clear-sky conditions to about 285 K for warm, cloudy-sky conditions.

An empirical relation that connects the effective sky temperature with the actual atmospheric temperature is,

$T_{sky}=0.0553T_s^{1.5}$K [9],

Where; T_s=atmospheric temperature °C.

The net radiative heat transfer between the glass envelope and the atmosphere is given by,

$$Q_{e-s,rad}=\sigma.\epsilon_e.A_e.[(T_e+273)^4-T_{sky}^4] \quad (4)$$

where; A_e=outer surface area of the glass envelope=$\pi D_o l$m^2.

Convective Heat Transfer from Glass Envelope to Surroundings $Q_{e-s,conv}$

The convective heat loss from the outer surface of the glass envelope to the atmosphere is given by,

$$Q_{e-s,conv}=h_{e-s}.A_e.(T_e-T_s) \quad (5)$$

In general, the convective heat transfer coefficient from horizontal cylindrical surface in presence of wind is given by,

$h=k_{std.}Nu/D_o$ W/m^2–K [8],

Where Nu=average Nusselt number based on outer surface area of glass envelope.

A simple empirical correlation for heat transfer coefficient from the outer surface of the glass envelope is developed by Mullik and Nanda,

$h_{e-s}=4.v_w^{0.58}.D_o^{-0.42}$W/m^2–K [9],

Hence the simplified expression for convective heat transfer from the outer surface of the glass envelope to the atmosphere is given by,

$$Q_{e-s,conv}=4.\pi.l.(D_o v_w)^{0.58}(T_e-T_s) \quad (6)$$

HEAT LOSS INVESTIGATION

The total heat loss from a solar receiver tube to the atmosphere consists of the following:

1. Radiative loss $Q_{e-s,rad}$
2. Convective loss $Q_{e-s,conv}$

The radiative loss $Q_{e-s,rad}$ and the convective loss $Q_{e-s,conv}$ are calculated using Eq. (1–6).

Thermal Conductivity of Air

The thermal conductivity of air at standard 1atm. pressure is used in the heat loss calculations. There is a necessity to express thermal conductivity of air explicitly as a function of temperature in order to simplify the equations to obtain T_e and hence $Q_{e-s,rad}$ and $Q_{e-s,conv}$. The values of thermal conductivity of air in the temperature range of

30°C to 300°C are obtained from Refs. [6–9]. A curve fit that explicitly expresses the thermal conductivity of air as a function of temperature is obtained using MATLAB (Figure 4).

Table 2: *Thermal Conductivity of Air at Various Temperatures in the Range 30°C to 300°C, 1 atm Pressure* [10].

Temperature t (°C)	Thermal Conductivity of Air at 1 atm. Pressure kstd (W/m–K)
30	0.02675
40	0.02756
50	0.02826
60	0.02896
70	0.02966
80	0.03047
90	0.03128
100	0.03210
120	0.03338
140	0.03489
160	0.03640
180	0.03780
200	0.03931
250	0.04268
300	0.04605

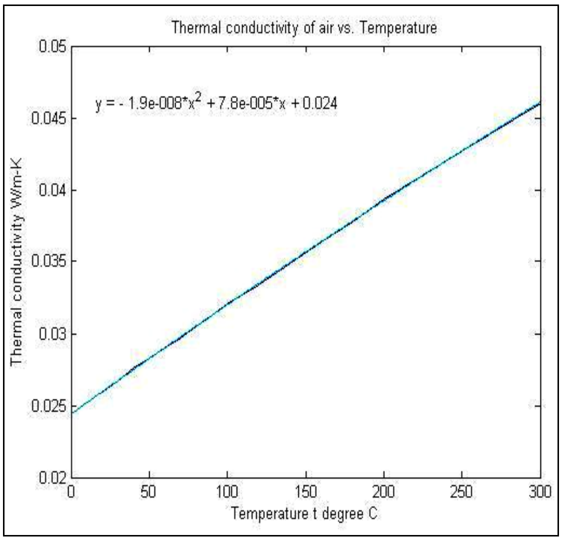


Fig. 4: *MATLAB Generated Curve Fit of Thermal Conductivity vs. Temperature at 1 atm. Pressure.*

Investigation is conducted assuming T_s=25°C, and P_s=1 atm., ϵ_a=0.21, ϵ_e=0.86 [8].

Investigation at Operating Temperature of 150°C

The simplified expressions of $Q_{a–e,rad}$, $Q_{a–e,conv}$, $Q_{e–s,rad}$ and $Q_{e–s,conv}$ for various values of outer diameter of heat pipe and annular gap at T_a=150°C is tabulated in Tables 3–5 and Figure 5.

Table 3: *Simplified Expression of $Q_{a-e,rad}$ and $Q_{a-e,conv}$ for Varying Values of Outer Diameter of Heat Pipe (d_o) and Annular Gap (D_i–d_o) at T_a=150°C.*

d_o (mm)	(D_i–d_o) (mm)	D_i (mm)	$Q_{a–e,rad}$ (W)	$Q_{a–e,conv}$ (W)
25	25	50	$29.4897–9.1980x10^{-10} T^4$	.0785xX/(.0087+1.50xY)
25	50	75	$29.6561–9.2499x10^{-10} T^4$	.0785xX/(.0137+1.33xY)
25	75	100	$29.7400–9.2761x10^{-10} T^4$	.0785xX/(.0173+1.25xY)
50	25	75	$58.6503–1.8293x10^{-10} T^4$	.1571xX/(.0101+1.67xY)
50	50	100	$58.9794–1.8396x10^{-10} T^4$	.1571xX/(.0173+1.50xY)
50	75	125	$59.1786–1.8458x10^{-10} T^4$	.1571xX/(.0229+1.40xY)
75	25	100	$87.7307–2.7364x10^{-10} T^4$	.2356xX/(.0108+1.75xY)
75	50	125	$88.1723–2.7501x10^{-10} T^4$	.2356xX/(.0192+1.60xY)
75	75	150	$88.4691–2.7594x10^{-10} T^4$	.2356xX/(.0260+1.50xY)

$X=[–4.75x10^{-9}(T_e+150)^2+3.9x10^{-5}(T_e+150)+.024]x(150–T_e)$
$Y=[1.136x10^{-4}+1.634x10^{-7}T_e]$

Table 4: *Simplified Expression of $Q_{e\text{-}s,rad}$ and $Q_{e\text{-}s,conv}$ for Varying Values of Outer Diameter of Heat Pipe (d_o) and Annular Gap (D_i–d_o) at T_a=150°C.*

d_o (mm)	(D_i–d_o) (mm)	D_i (mm)	$Q_{e-s,rad}$ (W)	$Q_{e-s,conv}$ (W)
25	25	50	$8.5786x10^{-9}T^4$–056.2015	$3.0329T_e$–075.8215
25	50	75	$1.2400x10^{-8}T^4$–081.2914	$3.7569T_e$–093.9215
25	75	100	$1.6238x10^{-8}T^4$–106.3813	$4.3912T_e$–109.7794
50	25	75	$1.2400x10^{-8}T^4$–081.2914	$3.7569T_e$–093.9215
50	50	100	$1.6238x10^{-8}T^4$–106.3813	$4.3912T_e$–109.7794
50	75	125	$2.0068x10^{-8}T^4$–131.4713	$4.9650T_e$–124.1255
75	25	100	$1.6238x10^{-8}T^4$–106.3813	$4.3912T_e$–109.7794
75	50	125	$2.0068x10^{-8}T^4$–131.4713	$4.9650T_e$–124.1255
75	75	150	$2.3898x10^{-8}T^4$–156.5612	$5.4943T_e$–137.3586

$T=(T_e+273)$.

Table 5: *Envelope Temperature, Radiative Heat Loss, Convective Heat Loss and Total Heat Loss for Varying Values of Outer Diameter of Heat Pipe (d_o) and Annular Gap (D_i–d_o) at T_a=150°C.*

d_o (mm)	(D_i–d_o) (mm)	T_e (°C)	$Q_{rad\text{-}loss}$ (W)	$Q_{conv\text{-}loss}$ (W)	$Q_{total\text{-}loss}$ (W)
25	25	35.33	21.3307	31.3295	052.6602
25	50	30.15	23.5049	19.3478	042.8527
25	75	27.84	26.6270	12.4709	039.0979
50	25	39.88	37.6210	55.9021	093.5230
50	50	33.64	37.1847	37.9398	075.1245
50	75	30.90	39.6976	29.2936	068.9912
75	25	42.69	54.8985	77.6799	132.5784
75	50	36.08	51.6698	55.0124	106.6822
75	75	32.99	52.9389	43.8998	096.8387

Investigation at Operating Temperature 300°C

The simplified expressions of $Q_{a–e,rad}$, $Q_{a–e,conv}$, $Q_{e–s,rad}$ and $Q_{e–s,conv}$ for various values of outer diameter of heat pipe and annular gap at T_a=300°C is tabulated in Tables 6–8 and Figure 6.

Table 6: *Simplified Expression of $Q_{a\text{-}e,rad}$ and $Q_{a\text{-}e,conv}$ for Varying Values of Outer Diameter of Heat Pipe (d_o) and Annular Gap (D_i–d_o) at T_a=300°C.*

d_o (mm)	(D_i–d_o) (mm)	D_i (mm)	$Q_{a–e,rad}$ (W)	$Q_{a–e,conv}$ (W)
25	25	50	$103.8618–9.6246x10^{-10}T^4$	.0785xX/(.0087+1.50xY)
25	50	75	$104.4745–9.6810x10^{-10}T^4$	.0785xX/(.0137+1.33xY)
25	75	100	$104.7836–9.7100x10^{-10}T^4$	.0785xX/(.0173+1.25xY)
50	25	75	$206.5126–1.9137x10^{-9}T^4$	.1571xX/(.0101+1.67xY)
50	50	100	$207.7237–1.9249x10^{-9}T^4$	.1571xX/(.0173+1.50xY)
50	75	125	$208.4571–1.9317x10^{-9}T^4$	.1571xX/(.0229+1.40xY)
75	25	100	$308.8686–2.8622x10^{-9}T^4$	.2356xX/(.0108+1.75xY)
75	50	125	$310.4930–2.8770x10^{-9}T^4$	.2356xX/(.0192+1.60xY)
75	75	150	$311.5855–2.8874x10^{-9}T^4$	.2356xX/(.0260+1.50xY)

$X=[-4.75x10^{-9}(T_e+300)^2+3.9x10^{-5}(T_e+300)+.024]x(300-T_e)$,
$Y=1.38x10^{-4}+1.6312x10^{-7}T_e$

Table 7: *Simplified Expression of $Q_{e\text{-}s,rad}$ and $Q_{e\text{-}s,conv}$ for Varying Values of Outer Diameter of Heat Pipe (d_o) and Annular Gap (D_i–d_o) at T_a=300°C.*

d_o (mm)	(D_i–d_o) (mm)	D_i (mm)	$Q_{e–s,rad}$ (W)	$Q_{e–s,conv}$ (W)
25	25	50	$8.5786x10^{-9}T^4$–056.2015	$3.0329T_e$–075.8215
25	50	75	$1.2400x10^{-8}T^4$–081.2914	$3.7569T_e$–093.9215
25	75	100	$1.6238x10^{-8}T^4$–106.3813	$4.3912T_e$–109.7794
50	25	75	$1.2400x10^{-8}T^4$–081.2914	$3.7569T_e$–093.9215
50	50	100	$1.6238x10^{-8}T^4$–106.3813	$4.3912T_e$–109.7794
50	75	125	$2.0068x10^{-8}T^4$–131.4713	$4.9650T_e$–124.1255
75	25	100	$1.6238x10^{-8}T^4$–106.3813	$4.3912T_e$–109.7794
75	50	125	$2.0068x10^{-8}T^4$–131.4713	$4.9650T_e$–124.1255
75	75	150	$2.3898x10^{-8}T^4$–156.5612	$5.4943T_e$–137.3586

Table 8: *Envelope Temperature, Radiative Heat Loss, Convective Heat Loss and Total Heat Loss for Varying Values of Outer Diameter of Heat Pipe (d_o) and Annular Gap (D_i–d_o) at T_a=300°C.*

d_o (mm)	(D_i–d_o) (mm)	T_e (°C)	$Q_{rad\text{-}loss}$ (W)	$Q_{conv\text{-}loss}$ (W)	$Q_{total\text{-}loss}$ (W)
25	25	63.2	053.489	116.067	169.557
25	50	49.7	053.317	092.907	146.224
25	75	43.4	056.374	080.841	137.216
50	25	77.8	106.770	198.625	305.395
50	50	62.3	098.960	163.966	262.927
50	75	54.7	100.093	147.709	247.803
75	25	86.5	164.845	270.057	434.902
75	50	70.6	148.340	226.553	374.894
75	75	62.7	147.012	207.246	354.259

Investigation at Operating Temperature 450°C

The simplified expressions of $Q_{a–e,rad}$, $Q_{a–e,conv}$, $Q_{e–s,rad}$ and $Q_{e–s,conv}$ for various values of outer diameter of heat pipe and annular gap at T_a=450°C is tabulated in Tables 9–11 and Figure 7.

Table 9: *Simplified Expression of $Q_{a\text{-}e,conv}$ for Varying Values of Outer Diameter of Heat Pipe (d_o) and Annular Gap (D_i–d_o) at T_a=450°C.*

d_o (mm)	(D_i–d_o) (mm)	D_i (mm)	$Q_{a–e,rad}$ (W)	$Q_{a–e,conv}$ (W)
25	25	50	321.7121–$1.1764x10^{-9}T^4$	.0785xX/(.0087+1.50xY)
25	50	75	324.0376–$1.1849x10^{-9}T^4$	.0785xX/(.0137+1.33xY)
25	75	100	325.2129–$1.1892x10^{-9}T^4$	.0785xX/(.0173+1.25xY)
50	25	75	638.8396–$2.3360x10^{-9}T^4$	.1571xX/(.0101+1.67xY)
50	50	100	643.4242–$2.3528x10^{-9}T^4$	.1571xX/(.0173+1.50xY)
50	75	125	646.2067–$2.3630x10^{-9}T^4$	.1571xX/(.0229+1.40xY)
75	25	100	954.8575–$3.4910x10^{-9}T^4$	.2356xX/(.0108+1.75xY)
75	50	125	960.9983–$3.5141x10^{-9}T^4$	.2356xX/(.0192+1.60xY)
75	75	150	965.1363–$3.5292x10^{-9}T^4$	.2356xX/(.0260+1.50xY)

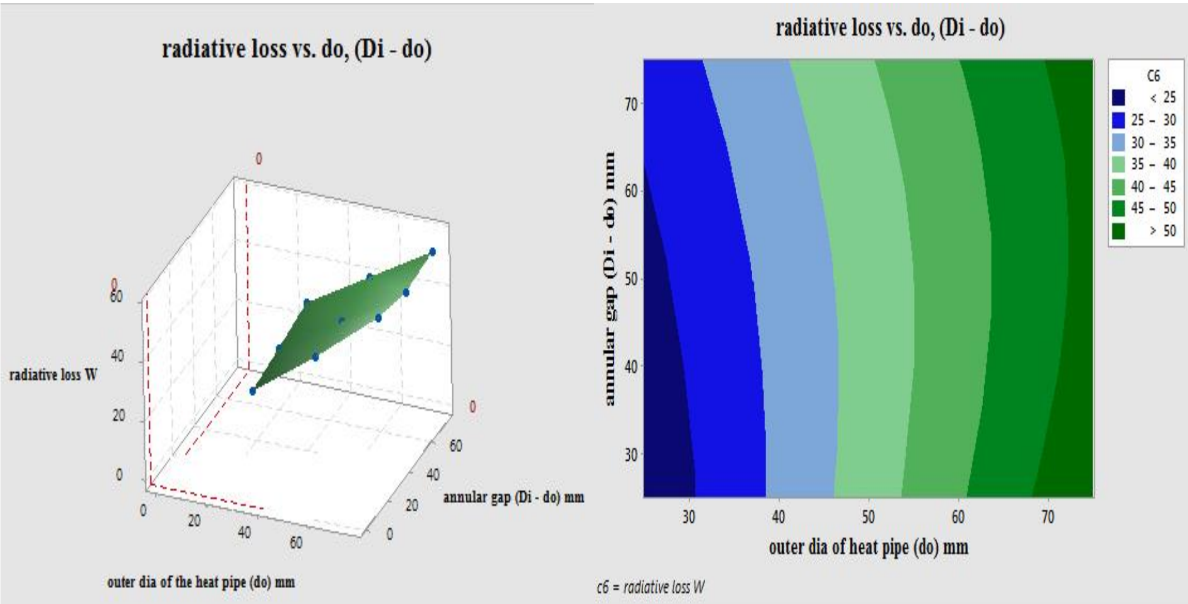
Table 10: *Simplified Expression of $Q_{e\text{-}s,conv}$ for Varying Values of Outer Diameter of Heat Pipe (d_o) and Annular Gap (D_i–d_o) at T_a=450°C.*

d_o (mm)	(D_i–d_o) (mm)	D_i (mm)	$Q_{e–s,rad}$ (W)	$Q_{e–s,conv}$ (W)
25	25	50	$8.5786x10^{-9}T^4$–056.2015	$3.0329T_e$–075.8215
25	50	75	$1.2400x10^{-8}T^4$–081.2914	$3.7569T_e$–093.9215
25	75	100	$1.6238x10^{-8}T^4$–106.3813	$4.3912T_e$–109.7794
50	25	75	$1.2400x10^{-8}T^4$–081.2914	$3.7569T_e$–093.9215
50	50	100	$1.6238x10^{-8}T^4$–106.3813	$4.3912T_e$–109.7794
50	75	125	$2.0068x10^{-8}T^4$–131.4713	$4.9650T_e$–124.1255
75	25	100	$1.6238x10^{-8}T^4$–106.3813	$4.3912T_e$–109.7794
75	50	125	$2.0068x10^{-8}T^4$–131.4713	$4.9650T_e$–124.1255
75	75	150	$2.3898x10^{-8}T^4$–156.5612	$5.4943T_e$–137.3586

Table 11: *Envelope Temperature, Radiative Heat Loss, Convective Heat Loss and Total Heat Loss for Varying Values of Outer Diameter of Heat Pipe (d_o) and Annular Gap (D_i–d_o) at T_a=450°C.*

d_o (mm)	(D_i–d_o) (mm)	T_e (°C)	$Q_{rad\text{-}loss}$ (W)	$Q_{conv\text{-}loss}$ (W)	$Q_{total\text{-}loss}$ (W)
25	25	117.6	143.546	280.934	424.4800
25	50	092.0	139.113	251.972	391.0857
25	75	078.8	142.542	236.552	379.0948
50	25	148.3	309.809	463.408	773.2175
50	50	122.3	290.200	427.349	717.5492
50	75	107.7	290.196	410.756	700.9529
75	25	164.8	490.650	614.281	1104.900
75	50	140.5	455.724	573.906	1029.600
75	75	125.9	448.638	554.489	1003.100

RESULTS AND DISCUSSIONS



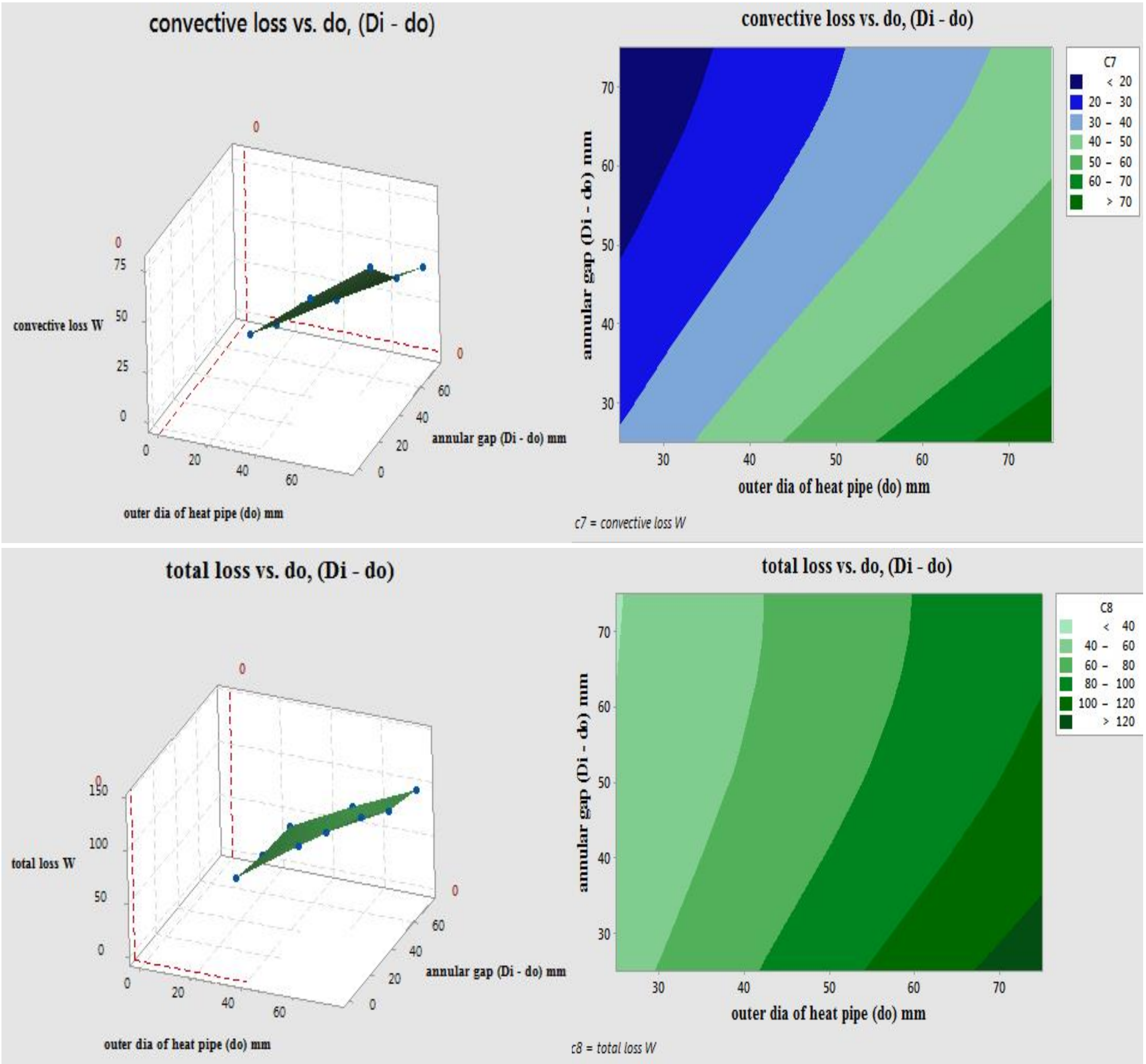
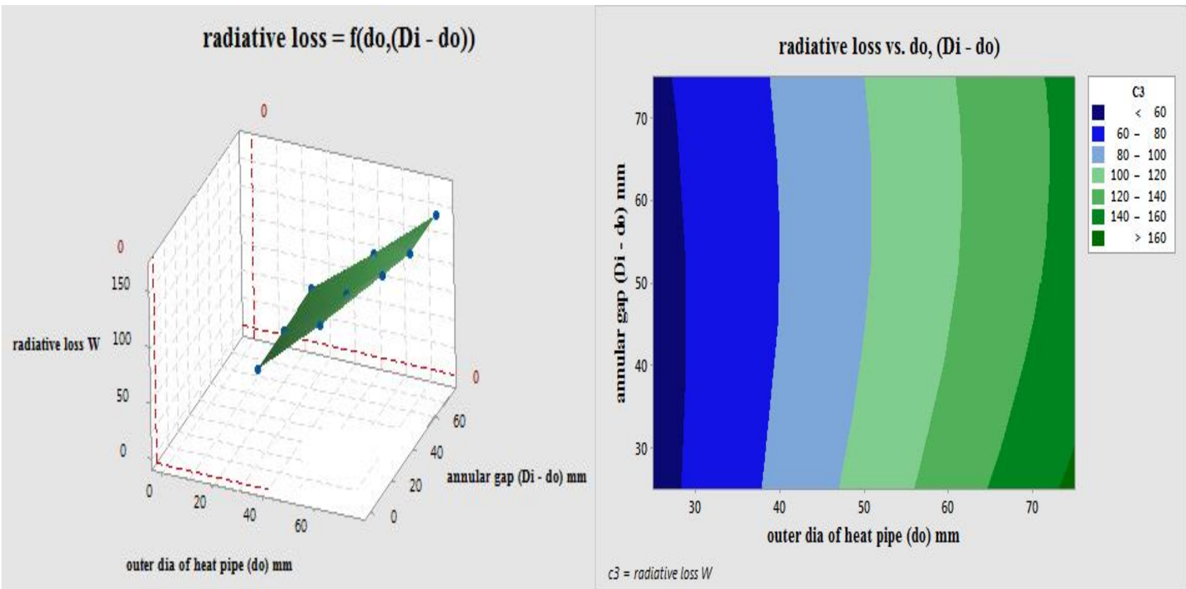


Fig. 5: *Radiative, Convective and Total Loss as a Function of Outer Diameter of Heat Pipe and Annular Gap at T_a=150°C, Plotting Tool: Minitab-17.*



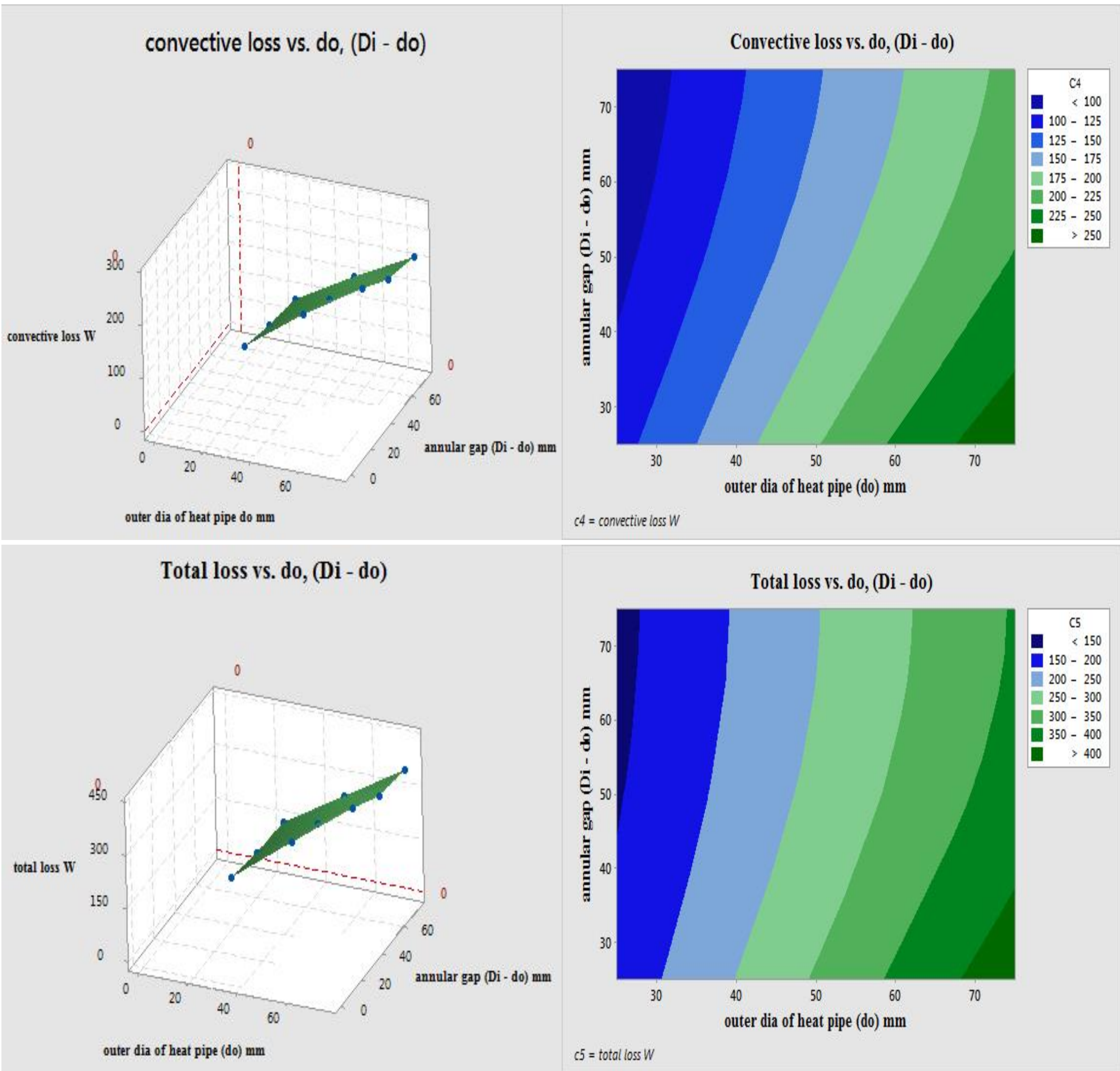
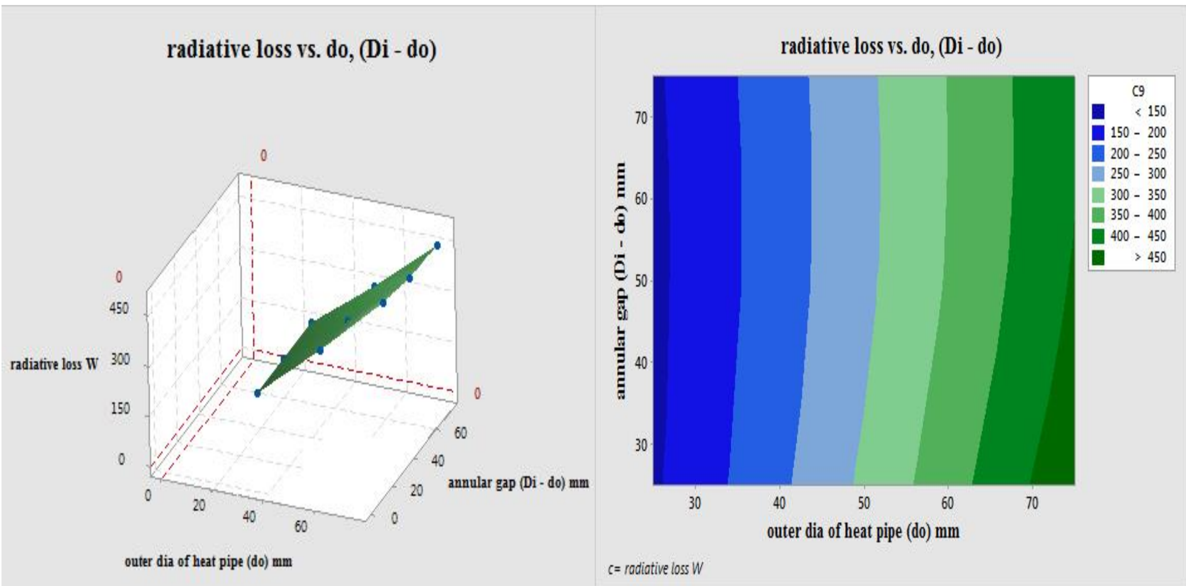


Fig. 6: *Radiative, Convective and Total Loss as a Function of Outer Diameter of Heat Pipe and Annular Gap at T_a=300°C, Created using Minitab-17.*



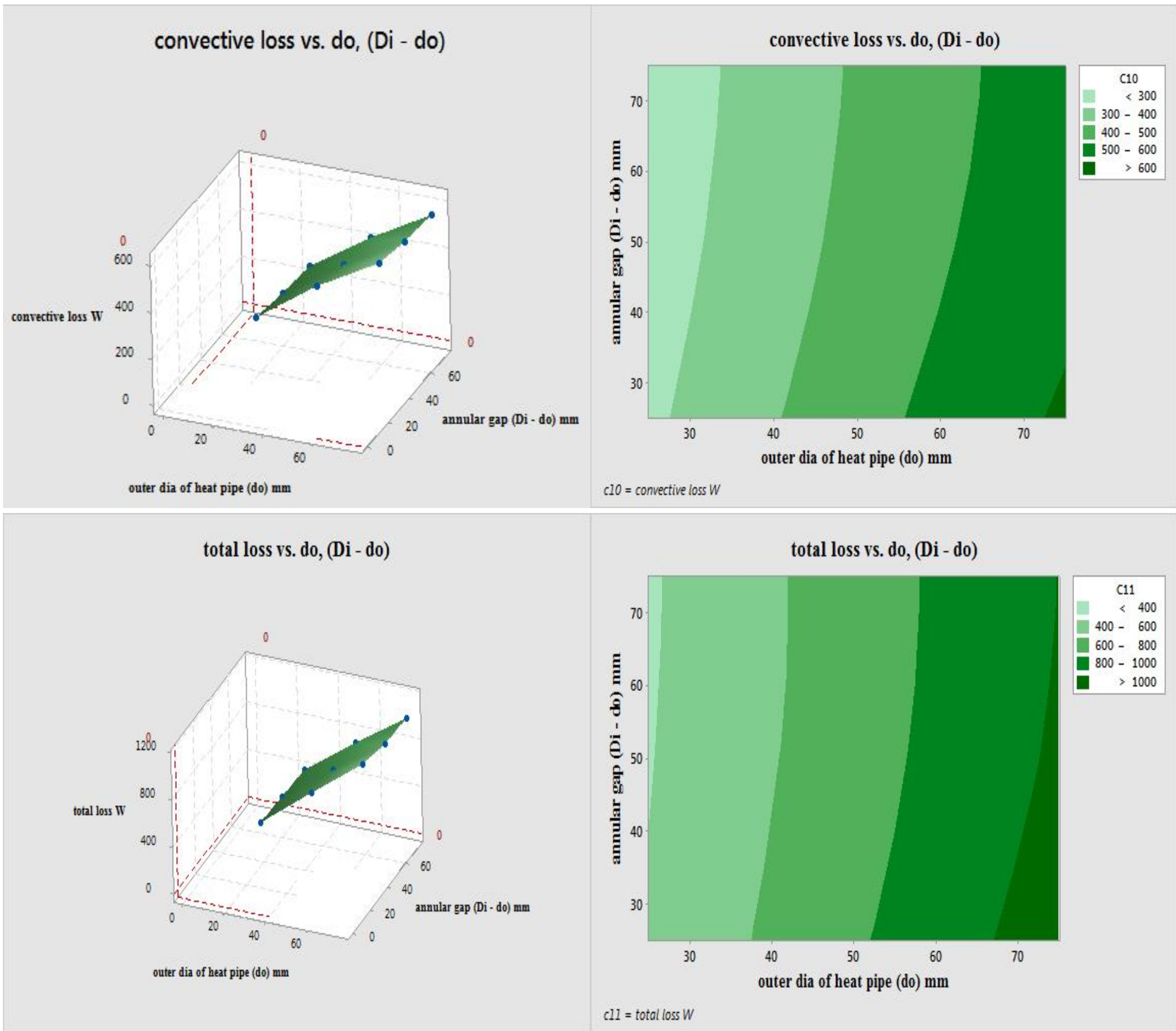


***Fig 7:** Total Loss as a Function of Outer Diameter of Heat Pipe and Annular Gap at T_a=450°C, Created using Minitab-17.*

CONCLUSIONS

1. At a given temperature and for a given size (outer diameter) of the heat pipe, the radiation loss increases with the annular gap while the convective loss decreases with the annular gap. The effect of decrease in convection loss supersedes the effect of increase in the radiation loss leading to decrease in overall heat loss. Hence, the total heat loss decreases with the increase in annular gap.
2. At a given temperature, both radiative and convective losses tend to increase with the size of the heat pipe. Hence, the total heat loss increases with the increase in size of the heat pipe.
3. As the operating temperature is increased, both radiative and convective losses tend to increase. Hence the total heat loss increases with the increase in operating temperature.

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