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DESIGN OF A 10 TON CAPACITY STANDARD VAPOUR COMPRESSION AIR-CONDITIONING PLANT FOR APPLICATION IN A MINI AUDITORIUM, IN NIGERIA

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ABSTRACT
Most public auditoria, in Nigeria, are saddled with the problem of ill-ventilated atmosphere; and people who gather for programmes tend to suffer suffocation as an unavoidable sequel. Such buildings are exposed to the sun; the roofs and walls receive irradiation from the sun throughout the day. The principal aim of air-conditioning of a room, in Nigeria, is to establish a stable thermal environment which satisfies the majority of the occupants with respect to comfort, under all the climatic conditions to which the building or space, is subjected. In the present design, the plant is to operate on a standard vapour compression basis; using R-134a, as refrigerant and running between 2 bar and 8 bar respectively. Relying on these parameters, the design gave a C.O.P. of 3.05, and the refrigerating effect of 148.99 kJ/Kg. The horsepower per ton of refrigeration for the 10 Ton plant is equals 4.39 hp/ton. Increasing the temperature of the refrigerant increases the COP; whereas, the COP is decreased by decrease in the temperature of the refrigerant

Keywords: Air-Conditioning Plant, Vapour Compression Cycle, Coefficient of Performance, Climatic Conditions, Space to be Air-conditioned.

INTRODUCTION

Air-conditioning is the simultaneous control of temperature, humidity, motion and purity of the atmosphere, in confined space. The important factors which are involved in a complete air-conditioning installation are;

i. Temperature Control: In air-conditioning, the control of temperature means the maintenance of any desired temperature within an enclosed space, even though the temperature of the outside air is above (or below) the desired room temperature (Khurmi and Gupta, 2008; Rajput, 2013). This is accomplished either by the addition or removal of heat from the enclosed space, as and when demanded. It may be noted that human being feels comfortable when the air is at 21 ^{o}C, with 56 % relative humidity.

ii. Humidity Control: The control of the humidity of air means the increasing the moisture contents of air across the seasons of the year, as the weather in Nigeria can be overly hot. The control of humidity is not only necessary for human comfort, but it also increases the efficiency of workers, in well chilled factories with heavy human traffic. In general, for effective air-conditioning in a tropical country like Nigeria, the relative humidity should not be less than 60 %.

iii. Air Movement and circulation: The motion or circulation of air is another important factor which should be controlled, in order to keep constant temperature throughout the conditioned space. It is,

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therefore, necessary that there should be equi-distribution of air throughout the space to be air-conditioned.

iv. Air Filtering, Cleaning and Purification: It is an important factor for the comfort of the human body. It has been noticed that people do not feel comfortable when breathing contaminated air, even if it is within acceptable temperature and humidity ranges. It is, thus, obvious that proper filtration, cleaning and purification of air is essential, to keep it free from dust and other impurities.

Strictly speaking, the *human comfort* depends upon physiological and psychological conditions. Hence, it is difficult to define the term without some ambiguities. There are many definitions given for this concept by different bodies; but the most acceptable definition (from the subject point of view) is given by the American Society of Heating, Refrigeration and Air-conditioning Engineers (ASHRAE). According to this group (ASHRAE, 1997), human comfort refers to that condition of mind, which expresses satisfaction with thermal environment. A typical air-conditioning cycle is shown in Fig. 1; and comprises the following steps:

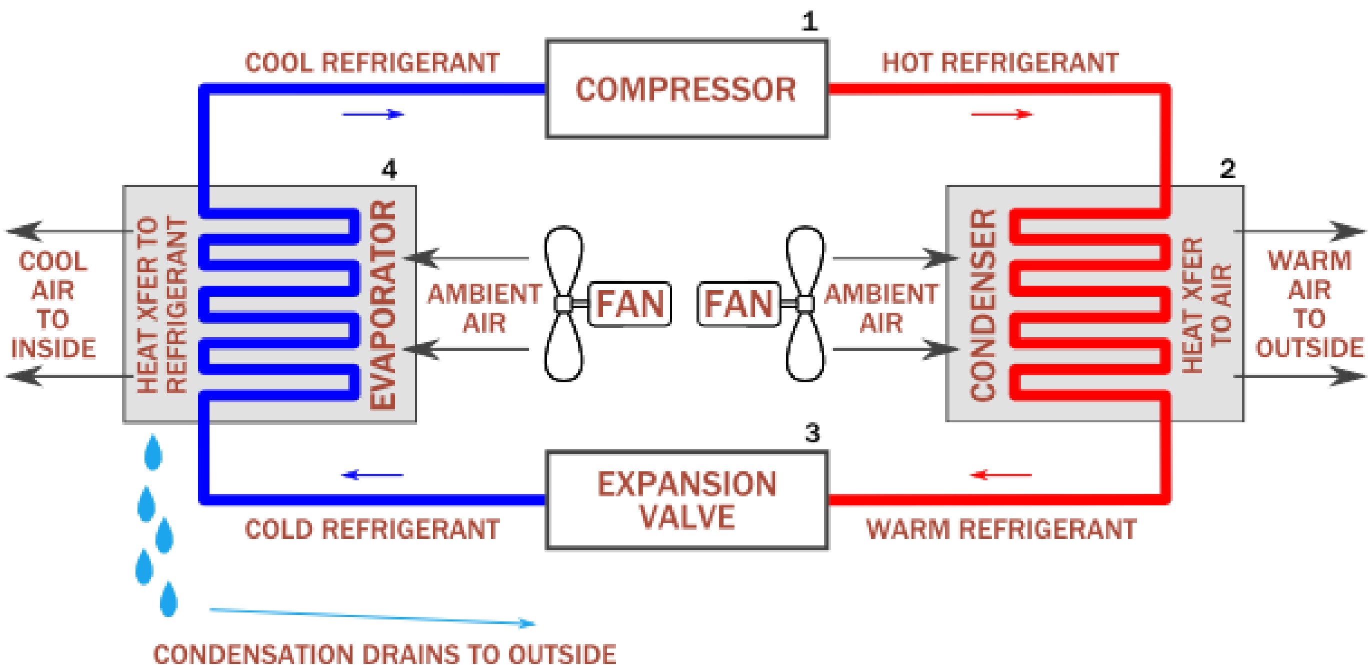

Figure 1: A Typical Air-conditioning Cycle

The fan forces air into duct work which is connected to openings, in the room. These openings are commonly called outlets or terminals. The duct work directs the air to the room through the outlets. The air enters the room, producing a cooling effect. Dust particles from the room enter the air-stream, and are carried along with it. Air flows from the room through a second outlet (sometimes called the return outlet), and enters the return duct work, where the dust particles are removed by a filter. After the air is cleaned, it is passed over the surface of a cooling coil. Finally, the air flows back to the fan, and the cycle is completed.

The main parts of the equipment in the air-conditioning cycle are explained below (Rajput, 2013)

Circulation Fan. The main function of this fan is to move air to and from the room. The air that a fan moves in air-conditioning system is made up of: all outdoor air; all indoor room air (called re-circulated air); and a combination of outdoor and indoor air. The fan pulls air from outdoors or from the room, but in most systems, it pulls air from both sources at the same time. The amount of air supplied by the fan must be regulated since drafts in the room causes discomfort, and poor air movement slows the heat rejection processes. This can be achieved by choosing a fan that can deliver the correct amount of air, and by controlling the speed of the fan so that air-stream in the room provides good circulation, but does not cause drafts. Of course, the fan is the only one of the pieces of equipment that contributes body comfort.

Air-conditioning Units. It is a unit which consists of cooling and dehumidifying processes for all-year round air-conditioning, in tropical countries, like Nigeria. The cooling coil can be located either ahead or after the fan, but should always be located after the filter. A filter ahead of the coil is necessary to prevent the excessive dirt, dust and air particles from covering the coil surfaces.

Supply Duct. It directs the conditioned air, from the circulating fan to the space to be air-conditioned, at proper point. In order that air may flow freely, it should be as short as possible, and have minimum number of turns.

Supply Outlets. These are grills, which distribute the conditioned air evenly, in a room. These outlets may fan the air, direct air in a jet-stream, and may do a combination of both. Since supply outlets can either be fan or jet the air-stream, therefore, they are able to exert some control on the direction of air delivered by the fan. This direction control plus the location and the number of outlets in the room, contribute a great deal to comfort or discomfort effect of air pattern.

Return Outlets. These are openings in a room surface which allow the room air to enter the return duct (i.e., return outlets allow air to pass from the room). They are usually located at opposite extreme of the wall or room, from the supply outlet.

Filters. The main function of the filters is to remove dust, dirt and other harmful bacteria from the air. They are usually located at some point in the return air-duct. They are made of many materials from spun glass to composite plastic. Other types operate on electrostatic principles.

Space. It is very important to have an enclosed space (i.e., room) since if it does not exist, it would be impossible to complete the air cycle. Lest, contained air from supply outlets would flow into the atmosphere.

Evaporative cooling is one of the oldest forms of cooling known, and has been used by man since 2500 B.C. Air cooling provision in buildings, especially in Nigeria, is undergoing a period of rapid expansion (Harris, 1983). Building cooling load components are; direct solar radiation, transmission load, ventilation/infiltration load and internal load. Calculating all these loads individually and adding them up gives the estimate of total cooling load. The load, thus calculated, constitutes total sensible load. Normal practice is that depending on the building type, certain percent of it is added to take care of latent load. Step by step calculation procedure has been adequately scripted in this paper (Stoecker and Jones, 1982). The requisite data of building material properties, climate conditions and ventilation standard are also established and reported by Prasad (1989). New types of office hall-planning and making air cooling are essential elements in the total environment design. The object of air cooling is to establish a stable thermal environment which satisfies the majority of occupants with respect to comfort under all the climatic conditions to which the building is subjected. The building of most auditoria in Nigeria, posed the problem of ill-ventilated atmosphere and suffocation in most of its part during the programs. Major parts of the building exposed to sun; the roof being heated throughout the day. Hence it was decided to provide the false ceiling and design air cooling system for mini college auditorium for better comforts to the occupants during longer program (Wane and Nagdeve, 2012). Factors directly affecting thermal comfort of the human are: air temperature, moisture content of the air, radiant exchange and air movement. It is the job of the thermal engineer to decide on the values of these factors, and design a system to maintain them within practical and economical limits; when the outside environment for most of the time (and in some cases continuously) will be hostile to this endeavour (Thornley, D. L., Partner, Roger and Partners, 1969).

MATERIALS AND METHOD

A 10 Ton air-conditioning system is to be designed for application in a mini auditorium, in Nigeria. It shall be air-cooled, with all-weather enclosure of ample capacity to meet the indoor requirements under adverse weather conditions. The plant should be capable of functioning round the clock. However, its usage would be casual. The plant is to operate on a standard vapour compression basis. It is expected to use R-134a, as refrigerant; operating between 2 bar and 8 bar respectively. The compression process is assumed to be isentropic; and the refrigerant leaves the evaporator dry saturated. Other losses in the system are neglected.

The thermodynamics of the vapour compression cycle can be analyzed on the temperature versus entropy diagram as depicted in Fig. 2. At point 1 in the diagram, the circulating refrigerant enters the compressor as a saturated vapour. From point 1 to 2, the vapour is isentropically compressed, and exits the compressor as superheated vapour. From point 2 to point 3, the vapour travels through part of the condenser, which removes the superheat by cooling the vapour. Between point 3 and point 4, the vapour travels through the remainder of the condenser, and is condensed into a saturated liquid. The condensation process occurs at essentially constant pressure. Between points 4 and 5, the saturated liquid refrigerant passes through the expansion valve, and undergoes an abrupt decrease of pressure. That process results in the adiabatic *flash evaporation* and *auto-refrigeration* of a portion of the liquid (typically, less than half of the liquid flashes). The adiabatic flash evaporation process is isenthalpic (occurs at constant enthalpy.

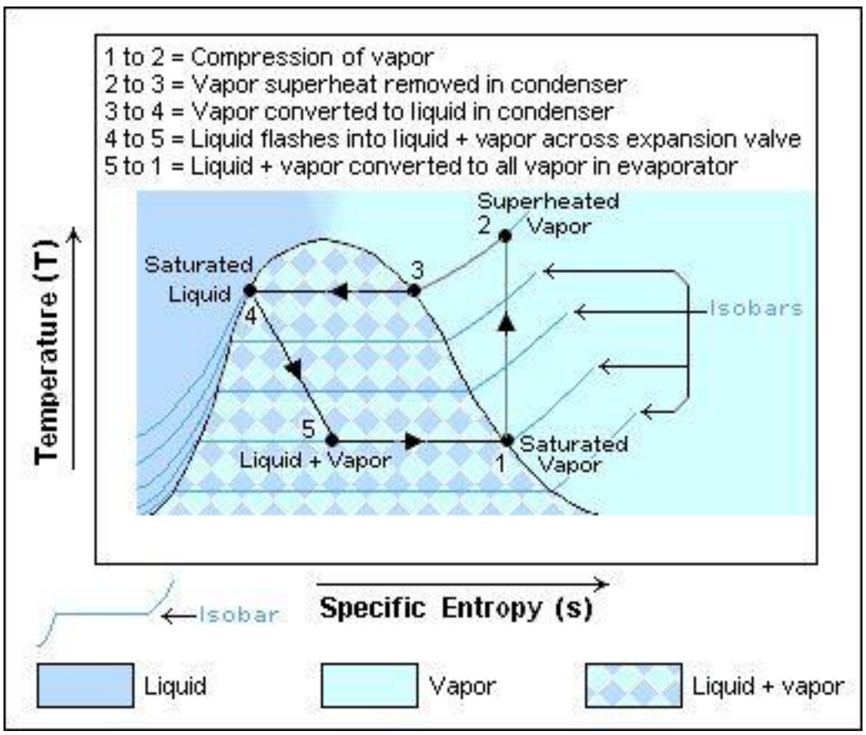

Fig. 2: T-S Diagram of a Vapour Compression Cycle

Between points 5 and 1, the cold and partially vaporized refrigerant travels through the coil or tubes in the evaporator, where it is totally vaporized by the warm air (from the space being air-conditioned) that a fan circulates across the coil or tubes, in the evaporator. The evaporator operates at essentially constant pressure, and boils off all available liquid there, after adding 4-8 Kelvin of superheat to the refrigerant as a safeguard for the compressor (as it cannot pump liquid). The resulting refrigerant vapour returns to the compressor inlet at point 1, to complete the thermodynamic cycle. A corresponding P-V diagram of this cycle is shown in Figure3 (Ballaney, 2011).

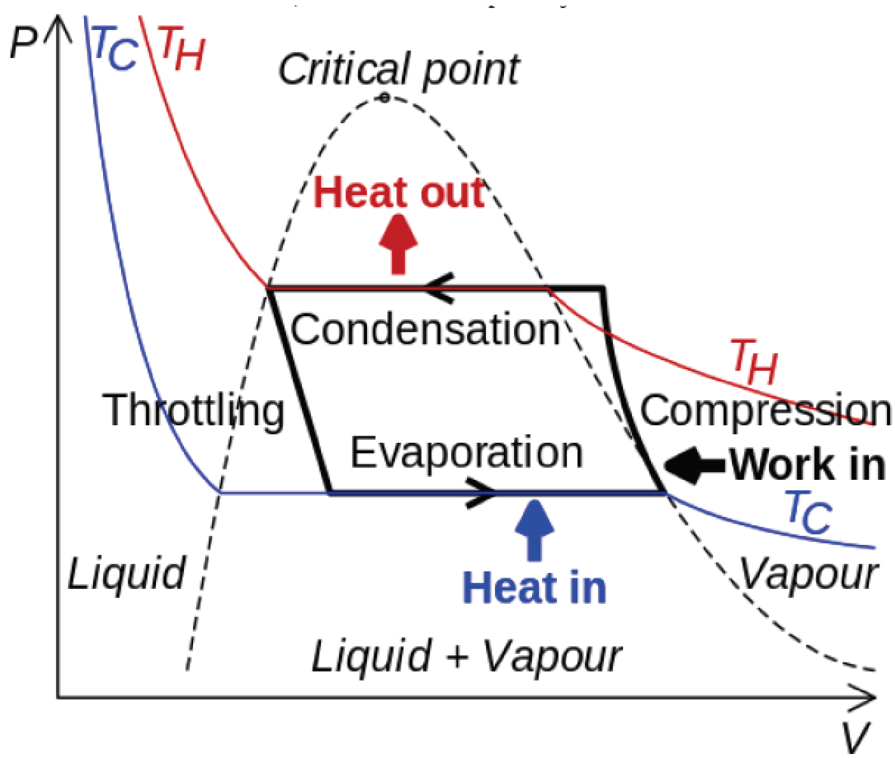

Fig 3: P-V Diagram of a Vapour Compression Cycle

RESULTS AND DISCUSSION

In the analysis of the system, we make use of the steady state flow energy equation. For ease of analysis, it is assumed that changes in kinetic and potential energies are negligible.

At point 1 (refer to Fig. 3), and from the saturated refrigerant-134a – Pressure Table (Appendix 1[10])

P_1= 2 bars = 200 kPa = 0.2 MPa
T_1= -10.09 ^oC
S_1= 0.93773 kJ/KgK
v_1= 0.099867 m^3/Kg
h_1 = 244.46 kJ/Kg

At point 2
P_2 = 8 bars = 800 kPa = 0.8 MPa
$S_1 \equiv S_2$ = 0.93773 kJ/KgK
We interpolate to get the value of h_2

$$\frac{S_2-S_2^I}{S_2^{II}-S_2^I} = \frac{h_2-h_2^I}{h_2^{II}-h_2^I}$$

$$\frac{0.93773-0.9183}{0.9480-0.9183} = \frac{h_2-267.29}{276.45-267.29}$$

h_2 = 293.28 kJ/Kg

At point 3
P_3 = 800 kPa
$h_3 = h_f$ = 95.47 kJ/Kg

At point 4
$h_4 = h_3$ = 95.47 kJ/Kg

Refrigerating Effect = $h_1 - h_4$
= 244.46 – 95.47
= 148.99 kJ/Kg

To determine the mass flow rate for the 10-ton plant, the following formula is used;

$$\dot{m}_1 = \frac{Q}{h_1-h_4}$$; Recall, 1 ton = 3.52 kW

Therefore, $\dot{m}_1 = \frac{10 \ x \ 3.52}{244.46 - 95.47}$ = 0.2363 Kg/min

The mass flow rate per KW of refrigeration is calculated from the equation;

$$\dot{m}_2 = \frac{1}{h1 - h4} = \frac{1}{244.46 - 95.47}$$

$= 6.71 \times 10^{-3}$ Kg/hr

The volumetric flow rate of the refrigerant through the evaporator, for the 10 ton plant, is determined thus;

$\tilde{V} = \dot{m}_1 \times v_1$
$= 0.2363 \times 0.099867$
$= 0.02360\ m^3/min$

To determine the power input in the system, in horse-power, for the 10 ton plant, we employ the following equation;

$P_{in} = \dot{m}_1(h_2 - h_1)$
$= 0.2363\ (293.28 - 244.46)$
$= 11.54$ KW

We multiply this value with a conversion factor of 1.34, to get the value of power input in hp, thus;

$P_{in} = 11.54 \times 1.34$
$= 15.46$ hp

To design the condenser capacity for the 10 ton plant, we refer to the equation;

$Q_c = \dot{m}_1(h_2 - h_3)$
$= 0.2363(293.28 - 95.47)$
$= 46.74$ KW

After compression, the temperature of the refrigerant increases; to determine this value, we make some interpolation, employing the equation;

$$\frac{h_2 - h_2^I}{h_2^{II} - h_2^I} = \frac{T_2 - T_2^I}{T_2^{II} - T_2^I}$$

$$\frac{293.28 - 267.29}{276.45 - 267.29} = \frac{T_2 - 31.31}{40 - 31.31}$$

$T_2 = 55.97\ ^oC$

The horsepower per ton of refrigeration for the 10 ton plant is calculated using the equation;

$$\text{hp/ton} = \frac{P_{in}}{3.52}$$

$$\text{hp/ton} = \frac{15.46}{3.52}$$

$= 4.39$ hp/ton

A certain percentage of the refrigerant is actually doing the cooling; and this is determined thus

$h_{f_3} = 95.47$ KJ/Kg; and $h_{f_4} = 38.43$ KJ/Kg

Therefore, $\Delta h_f = h_{f_3} - h_{f_4}$
$= 95.47 - 38.43$
$= 57.04$ KJ/Kg

But $h_{g_4} = 244.46$ KJ/Kg

$$\therefore \%\ \text{Vapour} = \frac{57.04}{244.46} = 0.2333 = 23.33\ \%$$

% Liquid = 1- 0.2333 = 0.7667 = 76.67 %

The system under design has a co-efficient of performance; this is calculated using the following relation:

$$COP = \frac{\dot{m}(h_1 - h_4)}{\dot{m}(h_2 - h_1)} = \frac{h_1 - h_4}{h_2 - h_1}$$

$$= \frac{244.46 - 95.47}{293.28 - 244.46}$$

$= 3.05$

If the system is superheated by 8.06 ^oC, the C.O.P. changes; and to determine this value, we perform another interpolation, thus:

$T_2 = 31.31 + 8.06 = 39.37\ ^oC$

$$\frac{T_2 - T_2^I}{T_2^{II} - T_2^I} = \frac{h_2^* - h_2^I}{h_2^{II} - h_2^I}$$

$$\frac{39 - 37}{40 - 37} = \frac{h_2^* - 267.29}{276.45 - 267.29}$$

$h_2^* = 275.76$ KJ/Kg

$$C.O.P_2^* = \frac{h_1 - h_4}{h_2^* - h_1} = \frac{244.46 - 95.47}{275.76 - 244.46} = 4.76$$

Assuming the system is to be under-cooled to 21.55 ^oC, a significant change is noticed in the C.O.P., this is determined thus:

After interpolation, we get; h_3^*, and h_4^* = 81.51 KJ/Kg

$$C.O.P_3^* = \frac{h_1 - h_4^*}{h_2 - h_1}$$

$$= \frac{244.46 - 81.51}{293.28 - 244.46} = 3.34$$

Sometimes, there are simultaneous processes of superheating and under-cooling in one system. At such moments, a variation in the C.O.P. of the system is noticed. This is calculated using the following mathematical relation;

$$C.O.P_4^* = \frac{h_1 - h_4^*}{h_2^* - h_1}$$

$$= \frac{244.46 - 81.51}{275.76 - 244.46} = 5.21$$

It has been noticed that a condition of superheating increases the C.O.P. of the system. Conversely, if the system is under-cooled, the C.O.P. decreases. Under the conditions of simultaneous superheating and under-cooling of the system, the C.O.P. undergoes a significant increase. One can, then, infer that the C.O.P. of the plant is linearly correlated with the temperature of the refrigerant. The plant's capacity to transfer heat to the outside is appreciably high, making the system efficient and reliable. This makes the contrivance able to function smoothly in a tropical region, like Nigeria.

CONCLUSION

The above discussion is based on the ideal vapour-compression refrigeration cycle, and does not take into consideration real-world effects like frictional pressure drop in the system, slight thermodynamic irreversibility during the compression of the refrigerant vapour, or non-ideal gas behaviour (if any). The performance of the system, as discussed, corresponds to the parameters that govern the design. The above calculations are hinged around some assumption of no system heat loss.

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APPENDIX 1

The saturated refrigerant-134a—pressure table (in SI units) is given below.

Saturated refrigerant-134a—Pressure table

		Specific volume, m^3/kg		*Internal energy,* kJ/kg			*Enthalpy,* kJ/kg			*Entropy,* kJ/kg·K		
Press., P kPa	Sat. temp., T_{sat} °C	Sat. liquid, v_f	Sat. vapor, v_g	Sat. liquid, u_f	Evap., u_{fg}	Sat. vapor, u_g	Sat. liquid, h_f	Evap., h_{fg}	Sat. vapor, h_g	Sat. liquid, s_f	Evap., s_{fg}	Sat. vapor, s_g
60	−36.95	0.0007097	0.31108	3.795	205.34	209.13	3.837	223.96	227.80	0.01633	0.94812	0.96445
70	−33.87	0.0007143	0.26921	7.672	203.23	210.90	7.722	222.02	229.74	0.03264	0.92783	0.96047
80	−31.13	0.0007184	0.23749	11.14	201.33	212.48	11.20	220.27	231.47	0.04707	0.91009	0.95716
90	−28.65	0.0007222	0.21261	14.30	199.60	213.90	14.36	218.67	233.04	0.06003	0.89431	0.95434
100	−26.37	0.0007258	0.19255	17.19	198.01	215.21	17.27	217.19	234.46	0.07182	0.88008	0.95191
120	−22.32	0.0007323	0.16216	22.38	195.15	217.53	22.47	214.52	236.99	0.09269	0.85520	0.94789
140	−18.77	0.0007381	0.14020	26.96	192.60	219.56	27.06	212.13	239.19	0.11080	0.83387	0.94467
160	−15.60	0.0007435	0.12355	31.06	190.31	221.37	31.18	209.96	241.14	0.12686	0.81517	0.94202
180	−12.73	0.0007485	0.11049	34.81	188.20	223.01	34.94	207.95	242.90	0.14131	0.79848	0.93979
200	−10.09	0.0007532	0.099951	38.26	186.25	224.51	38.41	206.09	244.50	0.15449	0.78339	0.93788
240	−5.38	0.0007618	0.083983	44.46	182.71	227.17	44.64	202.68	247.32	0.17786	0.75689	0.93475
280	−1.25	0.0007697	0.072434	49.95	179.54	229.49	50.16	199.61	249.77	0.19822	0.73406	0.93228
320	2.46	0.0007771	0.063681	54.90	176.65	231.55	55.14	196.78	251.93	0.21631	0.71395	0.93026
360	5.82	0.0007840	0.056809	59.42	173.99	233.41	59.70	194.15	253.86	0.23265	0.69591	0.92856
400	8.91	0.0007905	0.051266	63.61	171.49	235.10	63.92	191.68	255.61	0.24757	0.67954	0.92711
450	12.46	0.0007983	0.045677	68.44	168.58	237.03	68.80	188.78	257.58	0.26462	0.66093	0.92555
500	15.71	0.0008058	0.041168	72.92	165.86	238.77	73.32	186.04	259.36	0.28021	0.64399	0.92420
550	18.73	0.0008129	0.037452	77.09	163.29	240.38	77.54	183.44	260.98	0.29460	0.62842	0.92302
600	21.55	0.0008198	0.034335	81.01	160.84	241.86	81.50	180.95	262.46	0.30799	0.61398	0.92196
650	24.20	0.0008265	0.031680	84.72	158.51	243.23	85.26	178.56	263.82	0.32052	0.60048	0.92100
700	26.69	0.0008331	0.029392	88.24	156.27	244.51	88.82	176.26	265.08	0.33232	0.58780	0.92012
750	29.06	0.0008395	0.027398	91.59	154.11	245.70	92.22	174.03	266.25	0.34348	0.57582	0.91930
800	31.31	0.0008457	0.025645	94.80	152.02	246.82	95.48	171.86	267.34	0.35408	0.56445	0.91853
850	33.45	0.0008519	0.024091	97.88	150.00	247.88	98.61	169.75	268.36	0.36417	0.55362	0.91779
900	35.51	0.0008580	0.022703	100.84	148.03	248.88	101.62	167.69	269.31	0.37383	0.54326	0.91709
950	37.48	0.0008640	0.021456	103.70	146.11	249.82	104.52	165.68	270.20	0.38307	0.53333	0.91641
1000	39.37	0.0008700	0.020329	106.47	144.24	250.71	107.34	163.70	271.04	0.39196	0.52378	0.91574
1200	46.29	0.0008935	0.016728	116.72	137.12	253.84	117.79	156.12	273.92	0.42449	0.48870	0.91320
1400	52.40	0.0009167	0.014119	125.96	130.44	256.40	127.25	148.92	276.17	0.45325	0.45742	0.91067
1600	57.88	0.0009400	0.012134	134.45	124.05	258.50	135.96	141.96	277.92	0.47921	0.42881	0.90802
1800	62.87	0.0009639	0.010568	142.36	117.85	260.21	144.09	135.14	279.23	0.50304	0.40213	0.90517
2000	67.45	0.0009887	0.009297	149.81	111.75	261.56	151.78	128.36	280.15	0.52519	0.37684	0.90204
2500	77.54	0.0010567	0.006941	167.02	96.47	263.49	169.66	111.18	280.84	0.57542	0.31701	0.89243
3000	86.16	0.0011410	0.005272	183.09	80.17	263.26	186.51	92.57	279.08	0.62133	0.25759	0.87893