THERMODYNAMIC IMPROVEMENTS ON THE DIRECT-STEAM PLANT

Ronald DiPippo

Mechanical Engineering Department
University of Massachusetts Dartmouth
North Dartmouth, Massachusetts 02727

ABSTRACT

This paper offers two methods to improve the performance of a direct-steam geothermal power plant. One concept - Direct-Steam with Reheat - borrows a principle of operation from nuclear power plants, while the other - Direct-Steam with Interstage Moisture Removal - applies a well-known turbine design feature to excellent advantage. Both show improvements over a Basic Direct-Steam plant based on power output and utilization efficiency. Over a range of main steam temperatures from 150-200 C (300-400 F), the gain is 3-4 percent for the Reheat design and 5-7 percent for the Moisture Removal design.

INTRODUCTION

Direct-Steam (or Dry-Steam) plants account for 70 percent of the total worldwide installed geothermal power capacity [1]. The largest complex is at The Geysers in California where serious problems have recently caused a significant reduction in output [2,3]. One way to restore some of the lost output would be to modify existing plants to increase the utilization efficiency, i.e., to generate more electricity per unit mass of available steam.

An effective technique is to lower the condenser pressure, i.e., improve the waste heat rejection system so as to reduce the temperature of the cooling water and thus lower the turbine exhaust pressure. This has a dramatic impact on plant performance as can be seen, for example, by comparing PG&E Geysers Unit 18 and SMUDGEO No.1: The former consumes 25 percent more steam per unit of output than the latter using inlet steam of about the same conditions. The main difference between the plants is the turbine exhaust pressure: The former runs at 9.62 kPa (2.84 in Hg) while the latter runs at about 3.4 kPa (1 in Hg). However, major modifications to key plant elements would be required to retrofit an existing plant to operate at a significantly lower condenser pressure.

The two conceptual designs presented here offer modest improvements in performance but would involve fewer and less costly modifications to existing plant. It is not our intention to present complete designs, but rather to compare the new schemes with a basic plant to determine any possible thermodynamic gain.

THE SYSTEMS

Three systems will be compared: (1) the reference plant - a Basic Direct-Steam plant; (2) an improved basic plant - a Direct-Steam plant with Interstage Moisture Removal; and (3) a new design concept - a Direct-Steam plant with Internal Reheat.

<u>Basic Direct-Steam</u> This familar design is shown in simplified form in Fig. 1. In its simplest form, it consists of a production well(s), a turbine-generator, a condenser, cooling tower and injection well(s). The process diagram is given in Fig. 2 in temperature-entropy coordinates. Note that the expansion process, 1-2, takes place entirely within the wet region leading to a reltively low turbine efficiency according to the widely used Baumann rule [4].

<u>Direct-Steam with Moisture Removal</u>
Figure 3 depicts a plant having a turbine fitted with three stages of moisture removal. Traps built into the steam path collect water droplets and divert them away from the turbine blades resulting in a drier and more efficient overall expansion process. With several extraction points, the actual expansion line can be shifted toward the saturated vapor curve (see Fig. 4) and the turbine efficiency will be improved. Of course the water that is removed is no longer available to generate output. This loss must be weighed against the increase in turbine efficiency.

<u>Direct-Steam with Internal Reheat</u> The idea of using moisture separation and reheat via a side stream of main steam is used in some advanced nuclear power plants [5]. Since the thermodynamic inlet steam conditions in a nuclear plant are similar to that in a direct-steam plant, a reheat design may likewise be beneficial for geothermal applications. DiPippo and Vrane have presented the concept of using internal reheat for geothermal plants of the double-flash type [6].

Figure 5 shows a schematic of a direct-steam plant modified to incorporate an internal reheater. The turbine is divided into two sections, T1 and T2, and the reheater, RH, (actually, a superheater) is placed between T1 and T2. A side stream of the main steam is used to evaporate any moisture and superheat the steam before admission to T2. The condensed steam (state 6) may be reinjected.

The process diagram is given in Fig. 6. Note that the turbine T2 (process 3-A-4) consists of a dry section (3-A) and a wet section (A-4) with different efficiencies. The relative size of these segments depends on the main steam temperature, the reheat pressure, inlet temperature to turbine T2, and the condenser temperature.

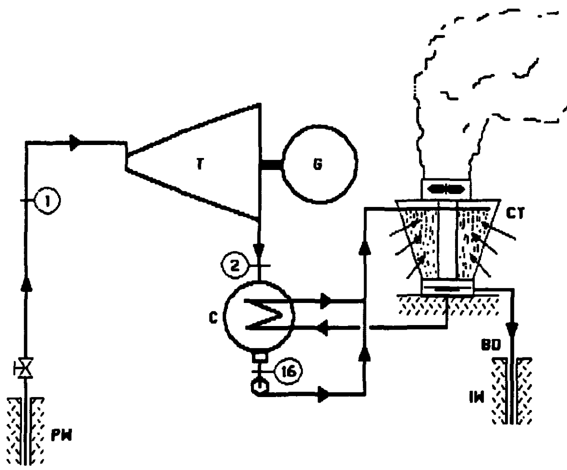

Fig.1 Flow diagram: Basic Direct-Steam Plant. Nomenclature: PW = production well; T,G = turbine and generator; C = condenser; CT = cooling tower; BD = blowdown; IW = injection well.

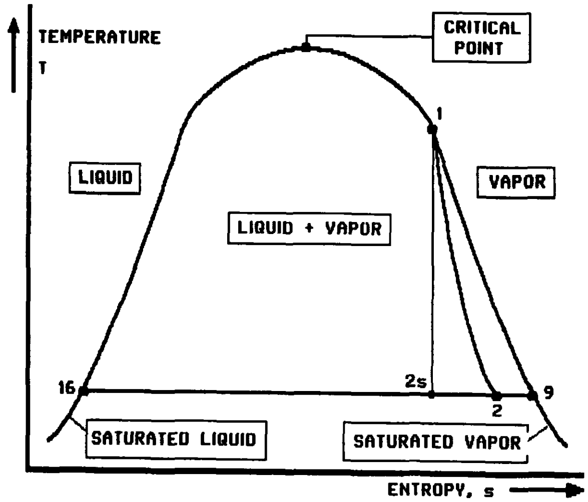

Fig.2 Temperature-entropy diagram for Basic Direct-Steam Plant. See Fig.1 for location of state points.

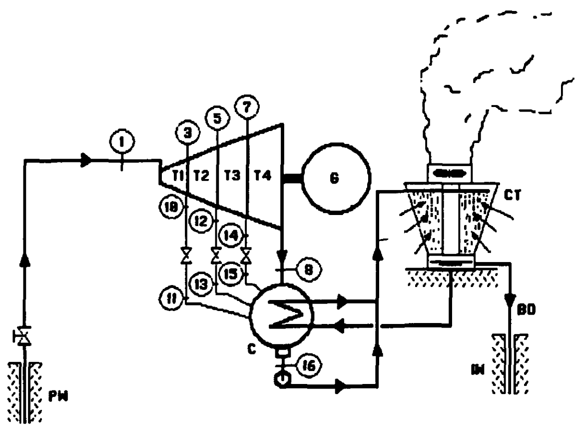

Fig.3 Flow diagram: Direct-Steam with Interstage Moisture Removal. Nomenclature: PW = production well; T1,T2,T3,T4,G = turbine and generator; C = condenser; CT = cooling tower; BD = blowdown; IW = injection well.

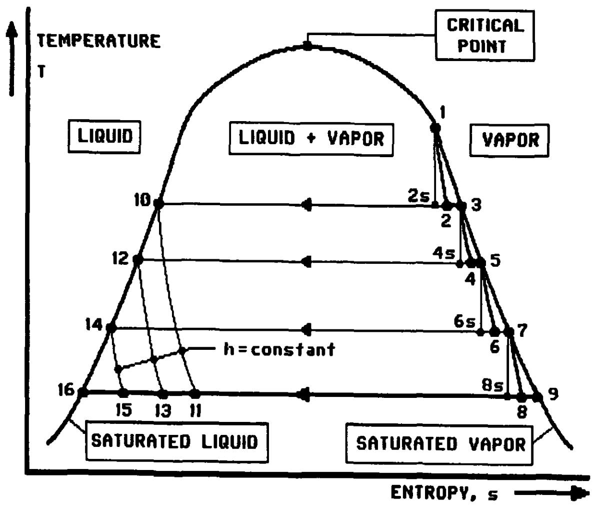

Fig.4 Temperature-entropy diagram for Direct-Steam with Interstage Moisture Removal. See Fig.3 for location of state points.

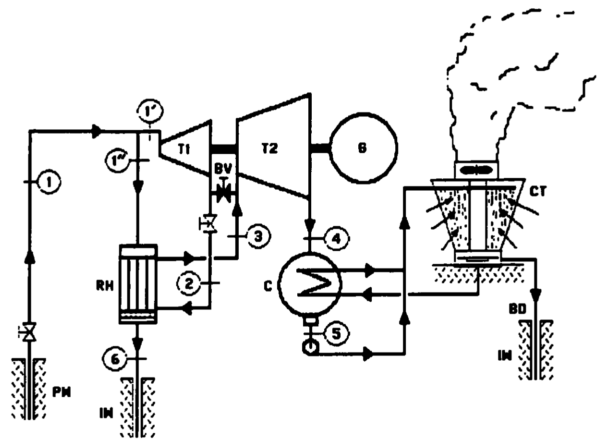

Fig.5 Flow diagram: Direct-Steam with Internal Reheat. Nomenclature: PW = production well; RH = reheater; T1, T2,G = turbine and generator; C = condenser; CT = cooling tower; BD = blowdown; BV = bypass valve; IW = injection well.

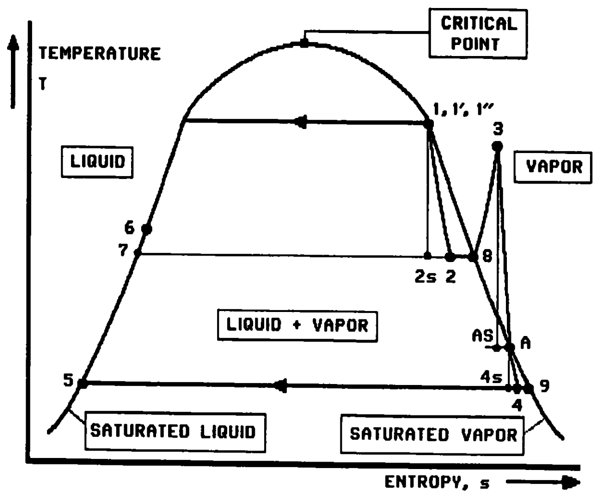

Fig.6 Temperature-entropy diagram for Direct-Steam with Internal Reheat. See Fig.5 for location of state points.

As with the Direct-Steam plant with Moisture Removal, this design involves a trade-off between the reduction of steam flow through the turbine (in this case due to main steam bleed to the reheater) and the increase in turbine efficiency (in this case due mainly to superheating).

THERMODYNAMIC PERFORMANCE ANALYSIS

The analysis of the plants is straightforward involving the application of the First and Second Laws of thermodynamics. Detailed examples of the analysis of several types of geothermal plant may be found in Ref. [1]. Applications of the Second Law to geothermal systems are given in Refs. [6] and [7].

The performance of the plants strongly depends on the performance of the turbines. The Baumann rule is used to determine the turbine efficiency when the expansion is in the wet region. The rule states that the wet efficiency equals the dry efficiency (assumed to be 85 percent) times the average quality during the actual expansion process. Thus, a Basic Direct-Steam plant will be significantly penalized due to the relatively high moisture content (i.e., low quality) at the turbine exhaust.

When analyzing the Reheat plant, state A (Fig. 6) is located such that the expansion line 3-A has an efficiency of 85 percent; the remaining portion, A-4, is governed by the Baumann rule.

A computer code was written to carry out the required calculations, including all thermodynamic property values using simple but accurate correlations for pure water. Each of the three systems was studied over a range of main steam temperatures from 150-200 C (300-400 F), saturated vapor. The condenser temperature was fixed at 40 C (104 F). This corresponds to a pressure of 7.38 kPa (2.2 in Hg).

The Direct-Steam plant with Moisture Removal was analyzed as follows: For all main steam temperatures, the inlet temperature for the last turbine section T4 was arbitrarily fixed at 100 C (212 F), and the inlet temperatures for sections T2 and T3 were calculated assuming equal temperature differences between each section. This simple scheme was checked against a lenghty multi-variable optimization routine and produced essentially identical results.

The Direct-Steam plant with Reheat was analyzed as follows: Reheat pressure P_2 (Fig. 6) was varied to find the optimum overall plant efficiency; for each trial the reheat temperature T_3 was set at a value 5 C (9 F) lower than the main steam temperature T_1, and the condensate temperature T_6 was set 5 C (9 F) higher than the saturation temperature T_2 corresponding to the reheat pressure. In other words, the terminal temperature differences at both ends of the reheater were set equal to 5 C (9 F).

RESULTS

<u>Overall Comparison</u> Table 1 compares the three systems on the basis of specific output and utilization efficiency [7]. The latter is defined as the ratio of the specific output, w, to the specific exergy, e_1, carried by the main steam:

Table 1
OVERALL COMPARISON
Basic System vs. Optimized
Reheat and Moisture Removal Systems

	Basic System		Reheat			Moisture Removal		
T_1	w	N_u	w	N_u	Advan.	w	N_u	Advan.
C	kW/ (kg/s)	%	kW/ (kg/s)	%	%	kW/ (kg/s)	%	%
150	489.5	65.7	505.1	67.8	3.2	513.0	68.9	4.8
160	518.6	66.3	536.7	68.6	3.5	546.4	69.9	5.4
170	545.8	66.8	565.1	69.1	3.5	577.8	70.7	5.9
180	571.2	67.1	593.0	69.7	3.8	607.2	71.4	6.3
190	594.9	67.4	618.1	70.1	3.9	634.6	72.0	6.7
200	616.7	67.7	641.9	70.5	4.1	660.0	72.5	7.0

$$N_u = w/e_1, \qquad (1)$$

with

$$e_1 = h_1 - h_o - T_o \times (s_1 - s_o), \qquad (2)$$

where h_1 and s_1 are the specific enthalpy and entropy at main steam conditions, T_o is the dead-state temperature (in K or R), and h_o and s_o are the enthalpy and entropy at the dead-state temperature. The dead-state temperature was fixed at 20 C or 293.15 K (68 F or 527.67 R).

The Reheat system holds a 3-4 percent advantage over the Basic system; the Moisture Removal system is even better, having roughly a 5-7 percent advantage over the Basic system. The gain relative to a Basic system ranges from 15-25 kW/(kg/s) of main steam for the Reheat system, and from 23-43 kW/(kg/s) for the Moisture Removal system. The higher the main steam temperature, the greater the advantage for the two alternatives. The same information is conveyed graphically in Fig. 7.

<ins>Direct-Steam</ins> <ins>with</ins> <ins>Reheat</ins> Table 2 lists the findings for the optimum split temperature, T_2^*, the output of each turbine section, the total output, and the utilization efficiency based on the Second Law [7]. The split temperature optimized at three different values: 120 C (248 F) for low-temperature main steam; 134 C (248 F) for medium-temperature main steam; and 144 C (291 F) for high-temperature main steam. In the neighborhood of these optimum points, the output was relatively insensitive to the split temperature. Table 3 contains the optimum thermodynamic state properties for the case of 175 C (347 F) main steam temperature.

Table 4 lists the Second Law efficiencies for the high-pressure turbine, the low-pressure turbine, and the reheater, as functions of the main steam temperature.

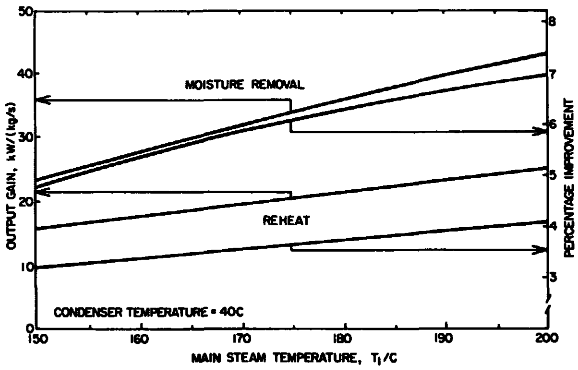

Fig.7 Output gain and percentage improvement as functions of main steam temperature. Moisture Removal and Reheat Systems vs. Basic Direct-Steam System.

Table 2
DIRECT-STEAM WITH INTERNAL REHEAT
Optimum Performance Results

T_1	T_2^*	w_{T1}	w_{T2}	w_{T3}	w_{TOT}	N_u
C	C	....	kW/(kg/s)	....		%
150	120	121.4	69.6	314.0	505.1	67.8
160	120	153.7	95.2	287.8	536.7	68.6
170	134	137.5	89.5	338.1	565.1	69.1
180	134	162.6	112.2	318.2	593.0	69.7
190	144	156.4	113.8	347.9	618.1	70.1
200	144	180.9	138.2	322.8	641.9	70.5

The turbine efficiency N_T is defined as the actual output divided by the drop in exergy across the turbine or section:

$$N_{T1} = w_{T1}/(e_1 - e_2) \qquad (3)$$

$$= (h_1 - h_2)/(e_1 - e_2) \qquad (4)$$

$$N_{T2} = w_{T2}/(e_3 - e_4) \qquad (5)$$

$$= (h_3 - h_4)/(e_3 - e_4). \qquad (6)$$

The reheater efficiency N_{RH} is defined as the ratio of the increase in steam exergy from state 2 to 3 (Figs. 5 and 6) to the decrease in exergy from state 1'' to 6:

$$N_{RH} = (\dot{m}_2/\dot{m}_{1''})x[(e_3-e_2)/(e_{1''}-e_6)], \qquad (7)$$

Table 3
DIRECT-STEAM WITH INTERNAL REHEAT
State Properties for Optimum Performance:
T_1 = 175 C (347 F)

state	T C	x -	h kJ/kg	s kJ/(kg·K)	E kW/(kg/s)
1	175	1.0	2773.7	6.6257	834.37
2	134	0.9473	2612.1	6.7084	596.49
3	170	0.0	2802.8	7.1684	647.94
4	40	0.9004	2334.6	7.4916	130.05
5	40	0.0	167.6	0.5725	2.51
6	139	-	584.8	1.7287	6.49
7	134	0.0	563.3	1.6765	68.84
8	134	1.0	2726.0	6.9882	625.83
A	111.9	1.0	2694.4	7.2180	454.82
9	40	1.0	2574.3	8.2570	144.15

Table 4
DIRECT-STEAM WITH INTERNAL REHEAT
Second Law Efficiencies for Components

T_1 C	N_{T1} %	N_{T2} %	N_{RH} %
150	87.0	83.1	86.3
160	86.6	83.6	83.2
170	87.1	83.0	85.8
180	86.8	83.4	84.0
190	87.1	83.1	85.3
200	86.7	83.6	83.4
Avgs.	86.9	83.3	84.7

where $\dot{m}_{1''}$ and $\dot{m}_2$ are the mass flow rates through the bleed line and the turbines, respectively. All values are fairly uniform over this temperature range and the average values shown may be used.

Direct-Steam with Moisture Removal
Table 5 shows the results for this design, including split temperatures, output from each section of the turbine, total output, and utilization efficiency. Table 6 gives state point property data for the case of 175 C (347 F) main steam.

Turbine Performance Comparison Table 7 compares the turbines for all three systems for the case of 175 C (347 F) main steam. The First Law (or isentropic) efficiency n_T is defined as the actual output divided by the ideal isentropic output for any given turbine or section. The Second Law (or exergy) efficiency N_T was defined above. Compared to the Basic system, the Reheat and Moisture Removal systems have significantly higher efficiencies of both types.

Table 5
DIRECT-STEAM WITH MOISTURE REMOVAL
Optimum Performance Results

T_1 C	$T_3/T_5/T_7$ .. C ...	$w_{T1}/w_{T2}/w_{T3}/w_{T4}$ ..... kW/(kg/s)	w_{TOT}	N_u %
150		71.4		
	133.3	74.2		
	116.7	77.0		
	100.0	290.5	513.1	68.9
160		82.7		
	140.0	86.6		
	120.0	90.5		
	100.0	286.7	546.5	69.9
170		93.1		
	146.7	98.2		
	123.3	103.6		
	100.0	283.0	577.9	70.7
180		102.6		
	153.3	109.2		
	126.7	116.0		
	100.0	279.3	607.1	71.4
190		111.3		
	160.0	119.5		
	130.0	128.0		
	100.0	275.8	634.6	72.0
200		119.1		
	166.7	129.2		
	133.3	139.4		
	100.0	272.3	660.0	72.5

CONCLUSIONS

Both the Moisture Removal and the Reheat systems offer improved performance over a Basic Dierct-Steam plant. The gain comes about through an increase in turbine efficiency. In both cases some working fluid is sacrificed in order to effect the turbine improvement. For example, when the main steam is at 175 C (347 F), using the Moisture Removal system, the loss of working fluid at points 10, 12 and 14 amounts to 9.9 percent of the main steam flow but is more than compensated by the improvement in turbine efficiency. The same conclusion holds for the Reheat system where 8 percent of the main steam is diverted to the reheater.

Although a detailed economic analysis is not possible here (site-specific factors will strongly influence the costs of incorporating the proposed modifications into an existing plant), the enhanced revenues can be calculated as a function of the price of electricity and capacity factor. Table 8 shows the increase in annual revenues for plants receiving a steam flow of 200 kg/s (1,600,000 lbm/h) at a temperature of 175 C (347 F). This corresponds to a Basic Direct-Steam plant output of about 112 MW. Depending on the electricity price and the capacity factor, the revenue enhancement can be dramatic. For example, assuming a price of $ 0.10/kWh and a capacity factor of 80 percent, a Reheat plant would generate $ 2,900,000 more than a Basic plant, and a Moisture Removal plant would produce a gain of $ 4,768,000. Improved revenues of this magnitude may justify the installation of the proposed systems in certain cases.

REFERENCES

[1] DiPippo, R. and Ellis, P.F., II, "Geothermal Power Cycle Selection Guidelines", Geoth. Info. Series, Part 2, Electric Power Research Institute, DCN 90-213-142-02-02, Palo Alto, CA, July 1990.

[2] __, "The Geysers - Three Decades of Achievement: A Window on the Future", GRC TRANS., v. 13, 1989.

[3] Hodgson, S.F., "The Geysers Geothermal Field: What Next?", Geothermal Hot Line, v. 19, 1989, pp. 21-23.

[4] Baumann, K., "Some Recent Developments in Large Steam Turbine Practice", J. Inst. Elec. Eng., v. 59, 1921, p. 581.

[5] Li, K.W. and Priddy, A.P., Power Plant System Design, J. Wiley & Sons, 1985.

[6] DiPippo, R. and Vrane, D.R., "A Double-Flash Plant with Interstage Reheat", GRC TRANS., v. 15, 1991, pp. 381-386.

[7] DiPippo, R. and Marcille, D.F., "Exergy Analysis of Geothermal Power Plants", GRC TRANS., v. 8, 1984, pp. 47-52.

Table 6
DIRECT-STEAM WITH MOISTURE REMOVAL
State Properties for Optimum Performance:
T_1 = 175 C (347 F)

state	T (C)	x (-)	h (kJ/kg)	s (kJ/(kg·K))	E (kW/(kg/s))
1	175	1.0	2773.7	6.6257	834.37
2	150	0.9666	2675.8	6.6712	723.11
3	150	1.0	2746.4	6.8381	719.92
4	125	0.9660	2639.0	6.8912	601.07
5	125	1.0	2713.6	7.0784	598.96
6	100	0.9646	2595.9	7.1407	472.04
7	100	1.0	2675.8	7.3549	470.75
8	40	0.9125	2363.6	7.5845	129.00
9	40	1.0	2574.3	8.2570	---
16	40	0.0	167.6	0.5725	2.73

Table 7
TURBINE PERFORMANCE COMPARISON
T_1 = 175 C (347 F)

Turbine section	w kW/(kg/s)	η_T %	N_T %
	BASIC SYSTEM		
T1	558.7	78.7	79.7
	REHEAT SYSTEM		
T1	148.7	82.8	87.0
T2	430.7	84.5	83.2
TOTAL	579.4		
	MOISTURE REMOVAL SYSTEM		
T1	97.9	83.6	88.0
T2	103.8	83.6	87.3
T3	109.9	83.5	86.6
T4	281.1	81.3	82.3
TOTAL	592.7		

Table 8
ADDED ANNUAL REVENUES (in $1000)
RELATIVE TO BASIC DIRECT-STEAM PLANT
T_1 = 175 C (347 F)

Price of Electricity $/kWh	Capacity Factor 60 %	80 %	100 %
	REHEAT SYSTEM		
0.05	1087	1450	1812
0.10	2175	2900	3625
0.15	3262	4350	5437
	MOISTURE REMOVAL SYSTEM		
0.05	1788	2384	2980
0.10	3576	4768	5960
0.15	5364	7152	8940