Variable Fill Fluid Coupling Radiator Fan Drive with Fuzzy Logic Flow Control for AFV Power Pack

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***Abstract*— Armed Force Vehicles (AFV) in today's scenario faces a more increasingly stringent cooling requirements. It is required to provide only the required amount of power to the cooling fan to meet the vehicles ambient operating conditions. In the normal automobile engines, the radiator fan is driven by the engine crankshaft. In this paper a Variable Fill Fluid Coupling (VFFC) is used to couple the radiator fan with the engine crankshaft. By controlling the fluid flow in to the VFFC i.e. Percentage fill, the output speed of the coupling which in turns drives the radiator fan is controlled, this reduces the parasitic losses. A Fuzzy Logic Controller (FLC) is used for precisely controlling the flow rate in accordance to the engine temperature. The complexity of the system and its control can be better analyzed by modelling and simulation. The obtained results from the simulation shows that for a prescribed engine temperature there is a change in the radiator fan speed.**

Keywords—VFFC, Percentage Fill, AFV, FLC, Volume Fraction,

I. INTRODUCTION

Thermal management system in traditional automobiles depends on the radiator fan driven by the engines crankshaft. In certain cases the fan speed may over cool the fluid, which decreases the efficiency [1]. Later invention used an electrical actuator to control and drive the radiator fan speed, thus making the radiator fan to rotate at variable speed independent of the engine speed [2]. Hydraulic motors where also used to drive the radiator fan at variable speeds and it became attractive because of the hydraulic properties such as compact spacing and power density [3]. Electronically controlled hydraulic motor driven radiator fan systems identifies the optimum fan speed according to the engines operating conditions [4].

Fluid coupling is a hydrokinetic device which is used to couple two shafts by means of a fluid medium, here the power is transmitted by means of fluid only. It consists of a pump, turbine and working fluid as illustrated in Fig. 1. The pump is driven by the prime mover which is an engine in our case, and the turbine drives the radiator fan. Here the power transmitted depends on the amount of fluid filled inside the coupling. They are widely used in locomotives, marine, industrial machines where there is a need of variable speed operation.

Armed Force Vehicles (AFV) are vehicles which are specially designed for defense and military purpose. They are also known as combat vehicles. The AFV power pack includes a modular power train that contains an internal combustion diesel engine along with the engine drive components, transmission system and various other supporting components.

Here in our case a Variable fill Fluid coupling, couples the engine crankshaft and the radiator fan. As the engine coolant temperature increases, the output speed of the VFFC which drives the fan is increased by increasing the control flow rate supplied to the VFFC by means of a proportional flow control valve. The proportional flow control valve in turn is modulated by means of a Fuzzy Logic Controller according to the engine coolant outlet temperature. The converse is true for decrease in coolant temperature. Thus the fan speed can be varied in accordance to the engine temperature.

II. THEORY AND MODELIING OF VFFV

Techniques used for modelling the fluid coupling in the literature can be extended and implemented for variable fill fluid coupling with continuous filling and emptying. By using the general rotor dynamics and the conventional design formula a mathematical model for the fluid coupling was developed and analyzed by Rolfe [5]. The experimental data obtained from the test rig had a close match with the results obtained from the model developed by him. Performance of the fluid coupling was predicted by Qulaman` [6], in which he used fundamental equations of a 1D flow. Later this model was extended by Wallace [7] and he modelled the fluid coupling by using two approaches namely constant velocity approach (mean flow path) and linear velocity approach. Here he assumed the velocities increase linearly with that of the distance from the mean radius. The fluid coupling model discussed in this paper is based on the model by Wallace [7], the model developed by him was extended for a variable fill fluid coupling with continuous filling and emptying.

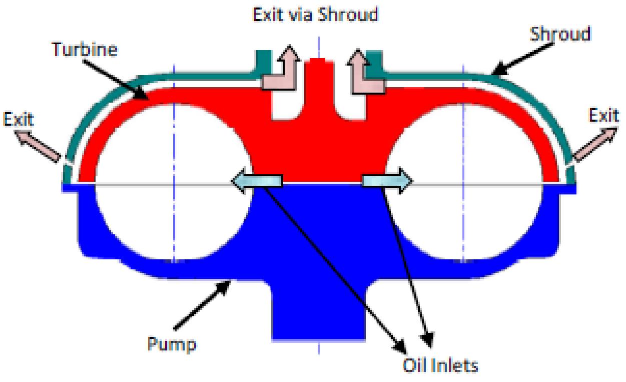


Fig.1 Schematic diagram of a Variable Fill Fluid Coupling

The cross section of the fully filled fluid coupling and its idealized outer flow regime indicating different flow features are shown in the Fig. 2 and Fig. 3. In this model the mass flow rate circulating between the two halves of the coupling are determined by using the empirical energy loss equations [8]. Here the centrifugal head dominates and the flow is towards the outer edge of the fluid coupling and it is considered as outer flow regime. The outer centered flow has an idealized circular vortex path and is used for simplifying the model [8]. On applying the continuity equation to both the halves of the coupling, inner and outer radial location R_1 and R_2 corresponding to the mean flow path Rm can be determined. Coupling size, flow regime and the oil fraction inside the coupling are used as a function to determine the mean flow path.

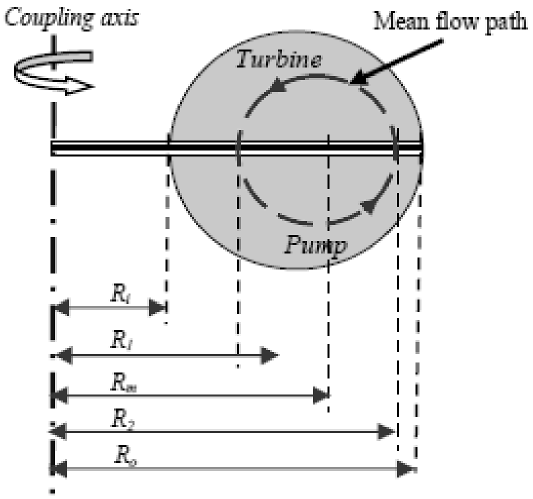


Fig. 2 Cross section of a fully filled fluid coupling Indicating flow features

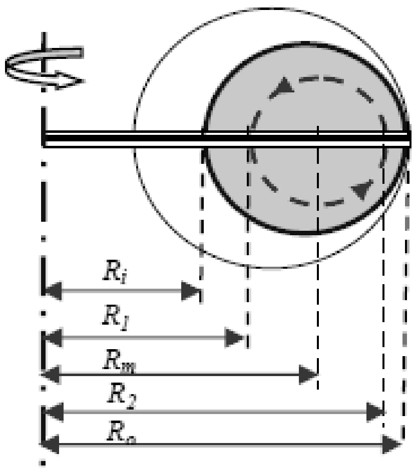


Fig. 3 Idealized Outer flow regime

A. *Outer Centred Flow Regime*

The Centre of the Mean flow path is given by the equation (1)

$$R_m = \sqrt{(R_o^2 + R_i^2)/2} \qquad (1)$$

The Mean flow paths inner and outer radii are given by (2) and (3)

$$R_1 = \sqrt{(R_m^2 + R_i^2)/2} \qquad (2)$$

$$R_2 = \sqrt{(R_m^2 + R_o^2)/2} \qquad (3)$$

where, R_i and R_o are the inner and outer radii of the oil filled inside the coupling. Here the R_i and R_o depends on the centrifugal flow diameter D and the vortex flow diameter d.

$$R_i = \frac{(D - d\,)}{2} \qquad (4)$$

$$R_o = \frac{(D + d)}{2} \qquad (5)$$

The volume fraction of the oil filled inside the coupling is given by:

$$f_{oil} = {}^{V_{oil}}/_{V_{total}} = {}^{D * d}/_{D_1 + d_1} \qquad (6)$$

Here D1 and d1 are the pitch circle diameter and minor diameter of the coupling respectively. If we look in detail the centrifugal flow diameter D and pitch circle diameter of the coupling D1 is the same. The values of D1 and d1 is taken from the fluid coupling dimensions (D1 = 244mm and d1 =98mm). Therefore for the given volume fraction i.e. percentage fill the vortex flow diameter d can be determined. It is assumed that the relation between the mass flow rate in to the coupling and the percentage fill is parabolic in nature. Therefore for the given mass flow rate corresponding percentage fill can be calculated, from this the value of d and other parameters is calculated.

B. *Constant Velocity Approach*

In turbomachines the torque developed can be calculated by the rate of change of angular momentum as the working fluid passes between the blades of the coupling [8]. The torque equation can be written as (7)

$$T = \dot{m}\left(\omega_p R_2^2 - \omega_t R_1^2\right) \quad (7)$$

$$T = \dot{m}\omega_p \left(R_2^2 - SR_1^2\right) \quad (8)$$

where, $S = \frac{\omega_t}{\omega_p}$ known as speed ratio;

The vortex mass flow rate ṁ is to be calculated for determining the torque transmitted. Here constant velocity approach is considered, where the movement of fluid is at a one single speed which is circulating around the vortex. The input power to the coupling and output power from the coupling which is called as pump power and turbine power is given by (9) and (10)

$$P_p = \dot{m}\omega_p^2\left(R_2^2 - SR_1^2\right) \quad (9)$$

$$P_T = \dot{m}\omega_T\omega_p\left(R_2^2 - SR_1^2\right) \quad (10)$$

The power loss or power dissipated can be calculated by the difference between the pump power and turbine power. The specific energy loss is given by P_L.

$$P_L = P_P - P_T \quad (11)$$

$$P_L = \dot{m}\omega_p^2\left(R_2^2 - SR_1^2\right) - \dot{m}\omega_T\omega_p\left(R_2^2 - SR_1^2\right) \quad (12)$$

$$P_L/\dot{m} = \left(\omega_p^2 - \omega_T\omega_p\right)\left(R_2^2 - SR_1^2\right) \quad (13)$$

$$p_L = \left(\omega_p^2 - \omega_T\omega_p\right)\left(R_2^2 - SR_1^2\right) \quad (14)$$

The total incidence power loss according to Qualman [6] is given by (15)

$$p_i = \frac{1}{2}\left(\omega_p - \omega_T\right)\left(R_2^2 + R_1^2\right) \quad (15)$$

According to Wallace [7], passage power loss which is similar to that of the pipe friction is proportional to the square of the flow velocity and is given by (16)

$$p_f = \frac{1}{2} K * C^2 \quad (16)$$

The total incident power loss and passage power loss are added together and is equated to the power loss which is obtained by subtracting the pump power and turbine power. On equating, the value of the vortex velocity C_{vor} can be calculated and is given by (17)

$$C_{vor} = \frac{\omega_P}{\sqrt{K}}\left[\left(1 - S^2\right)\left(R_2^2 - R_1^2\right)\right]^{0.5} \quad (17)$$

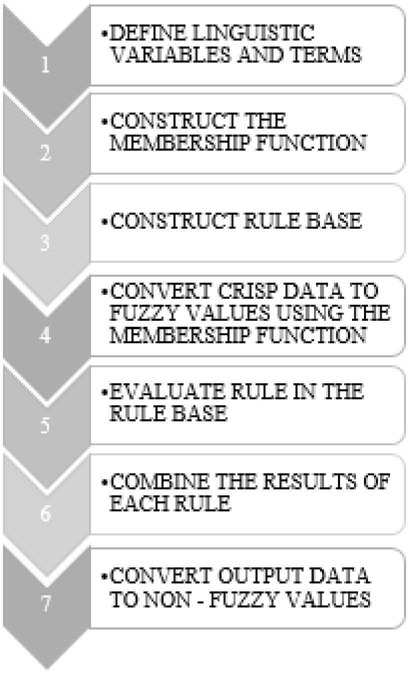
On knowing the vortex velocity the mass flow rate for the outer centered flow can be calculated by using (18)

$$\dot{m} = \rho\pi C_{vor}\left(R_m^2 - R_i^2\right) \quad (18)$$

III. FUZZY LOGIC CONTROLLER

The flow rate of the fluid in to the VFFC is controlled by means of a fuzzy logic controller. The input to the controller is the engine coolant temperature. The flow rate is varied in accordance to the engine coolant temperature. Precise control can be achieved by using this fuzzy logic controller. The fuzzy logic toolbox in MATLAB is used for this purpose. The general fuzzy logic algorithm [9] gives a better understanding about the fuzzy system working and is shown in the Table 1.

TABLE I. Fuzzy Logic Algorithm



The input and output membership functions are shown in the Fig. 4 and Fig. 5. . “If Then” Rule is used to give a relationship between the input and output variables. Mamdani type fuzzy logic controller is chosen with a min max composition. Centroid method of defuzzification is used. The ruler view of the fuzzy logic controller is shown in the Fig. 6.

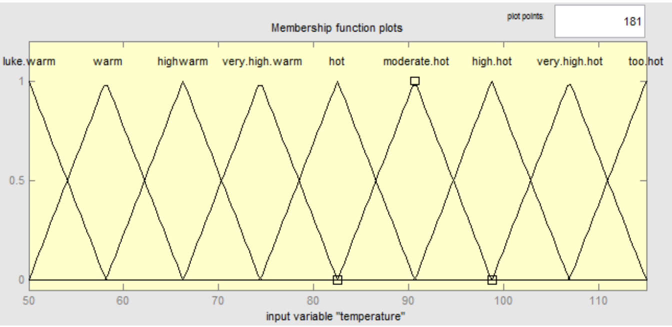


Fig. 5 Input Membership Function

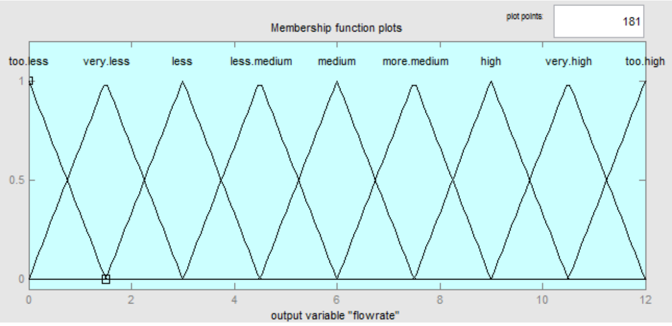
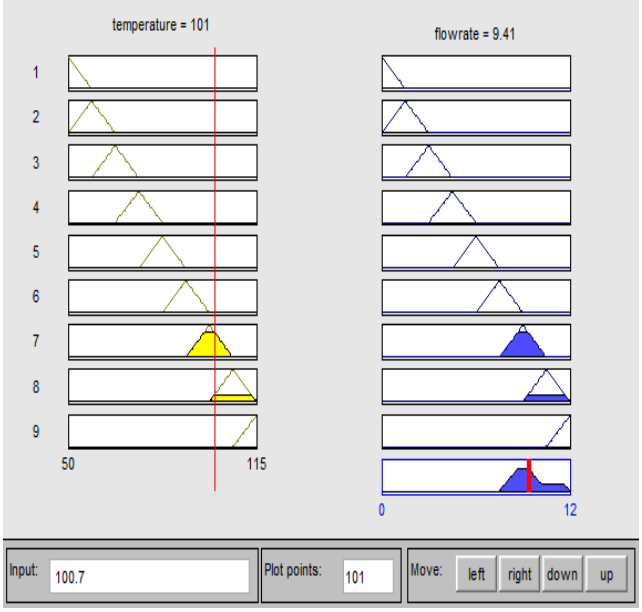


Fig. 5 Output Membership Function



Input: 100.7
Plot points: 101

Fig. 6 Ruler View

IV. GENERALISED PLANT MODEL

The overall model of the proposed system is shown in the Fig. 7. For the given engine coolant temperature a suitable flow rate is obtained from the fuzzy logic controller. The fluid flow rate in to the Variable fill Fluid coupling is responsible for maintaining the percentage fill of fluid inside the coupling. Thus the Percentage fill is determined by the fluid flow rate in to the coupling, which is in turn decided by the Engine coolant temperature. In simple words, if the engine coolant temperature is high, percentage fill is high and vice versa. This Percentage fill (i.e. Volume Fraction) has an effect on the torque transmitted by the variable fill fluid coupling. As the radiator fan is driven by this fluid coupling, the fan torque is controlled by the percentage fill. The SIMULINK model for the system with a fuzzy logic controller is shown in the Fig. 8.

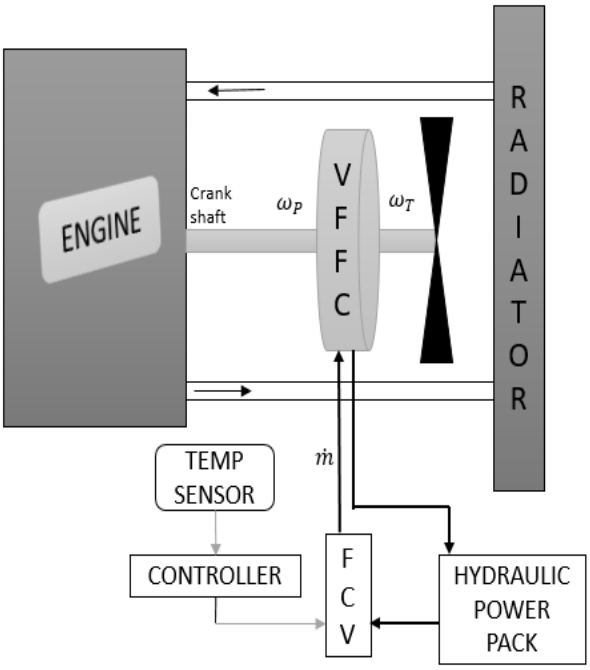


Fig. 7 over all model of the system

V. RESULTS AND DISCUSIIONS

Simulations were done by varying percentage fill values and the slip values at a constant input speed of (pump speed) of 6700rpm. Corresponding Torque values, Pump power P_p, Turbine power T_p are also obtained from the model. Engine coolant temperature values are related with percentage fill values. The obtained results from the model are shown in the Table 2 and Table 3. A graph is also plotted between the turbine speed and the torque for various fill percentage and is shown in the Fig. 9.

It is seen from the graph that by varying the percentage fill values of the VFFC, the output torque values are controlled. Thus the torque and power supplied to the engine radiator fan is reduced accordingly by varying the percentage fill values.

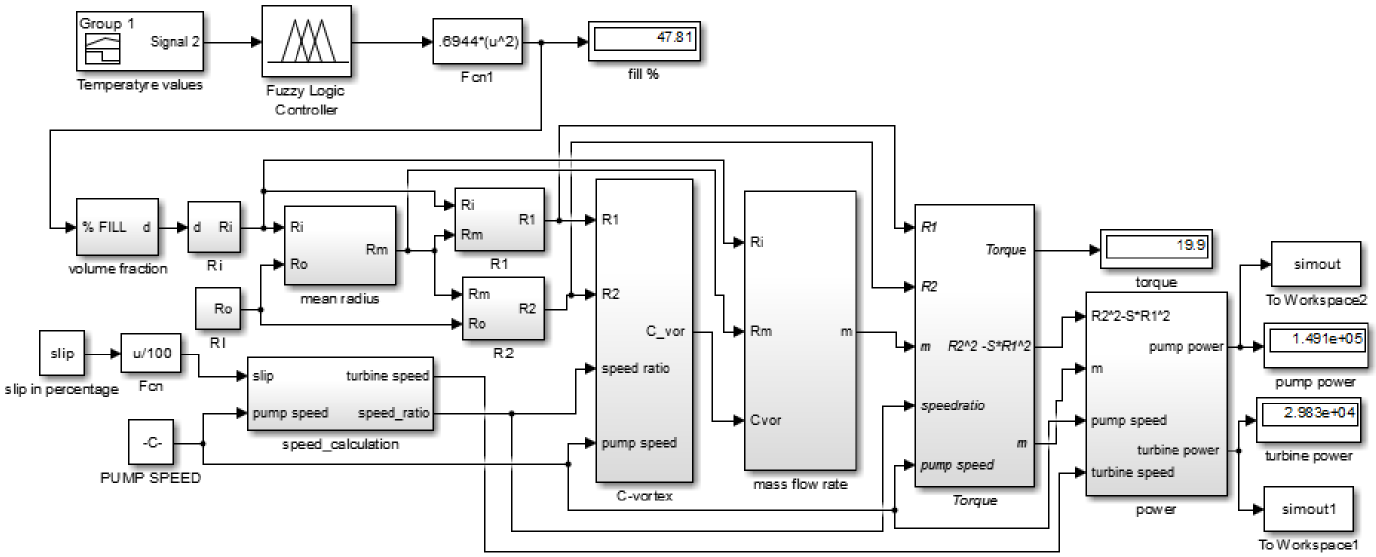


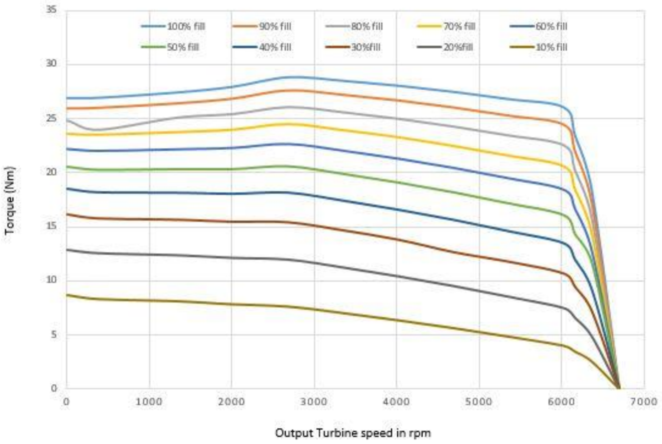
Fig. 8 SIMULINK model of the system

TABLE 2 Output torque, pump and turbine power at different percentage fill at input speed 6700 rpm

SLIP	O/P Speed	50% fill			40% fill			30% fill			20% fill			10% fill		
		O/P t	Pp	Pt	O/P t	Pp	Pt	O/P t	Pp	Pt	O/P t	Pp	Pt	O/P t	Pp	Pt
0	6700	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
5	6365	11.73	148	140.6	9.26	133.6	126.9	7.23	116	110.2	4.9	93.5	88.3	2.56	62.47	59.35
8	6164	14.32	184.7	169.9	11.98	166.7	153.4	9.42	144.7	113.1	6.59	116.7	107.4	3.46	77.97	71.73
10	6030	16.07	203.1	182.8	13.5	183.3	165	10.65	159.2	143.2	7.5	128.3	115.5	4	85.75	77.17
20	5360	17.18	197.7	158.1	14.63	178.4	142.8	11.75	154.9	123.9	8.56	124.3	99.93	4.85	83.46	66.67
30	4690	18.2	192.1	134.5	15.68	173.4	121.4	12.63	150.6	105.4	9.55	121.4	84.97	5.64	81.11	56.77
40	4020	19.1	186.4	111.8	16.62	168.2	100.9	13.78	146.1	87.63	10.45	117.8	70.66	6.36	78.69	47.21
50	3350	19.9	180.4	90.22	17.46	162.9	81.45	14.64	141.1	70.71	11.27	114	57.01	7.03	76.19	38.09
60	2680	20.6	174.3	63.73	18.2	157.4	62.95	15.39	136.6	54.65	12	110.2	44.06	7.63	73.6	29.44
70	2010	20.35	161.7	48.51	18.1	146	43.79	15.44	126.7	38.02	12.16	102.2	30.65	7.84	68.27	20.48
80	1340	20.34	152.2	30.43	18.19	137.4	27.47	15.62	119.3	23.83	12.4	96.16	19.23	8.12	64.25	12.65
90	370	20.29	153.4	14.34	18.24	129.5	12.95	15.75	122.4	11.24	12.6	90.64	9.06	8.33	60.5	6.056
100	0	20.56	137.7	0	18.56	124.3	0	16.11	107.9	0	12.9	87.03	0	8.69	58.15	0

TABLE 3 Output torque, pump and turbine power at different percentage fill at input speed 6700 rpm

SLIP	O/P Speed	100% fill			90% fill			80% fill			70% fill			60% fill		
		O/P t	Pp	Pt	O/P t	Pp	Pt	O/P t	Pp	Pt	O/P t	Pp	Pt	O/P t	Pp	Pt
0	6700	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
5	6365	18.43	193.7	184	17.2	186.9	177.6	15.86	179.2	170.2	14.42	170.3	161.8	12.84	160	152
8	6164	23.42	241.7	224	21.9	233.3	214.6	20.23	223.6	205.7	18.42	212.3	195.5	16.64	199.7	183.7
10	6030	26.06	266	240	24.38	256.5	230.9	22.56	245.9	221.3	20.58	233.7	210.4	18.42	219.6	197.7
20	5360	26.84	258.8	207	25.23	249.7	199.7	23.49	239.3	191.5	21.58	227.5	182	19.49	213.8	171
30	4690	27.51	251.5	176	25.98	242.7	169.9	24.31	232.6	162.8	22.47	221.1	154.8	20.45	207.7	145.4
40	4020	28.08	244	146.4	26.63	235.4	141.2	25.03	225.7	135.4	23.28	214.5	128.7	21.3	201.5	120.9
50	3350	28.53	236.2	118.1	27.15	227.9	114	25.63	218.5	109.2	23.93	207.7	103.8	22.04	195.1	97.57
60	2680	28.86	228.2	91.3	27.56	220.2	88.08	26.11	211.1	84.43	24.5	200.6	80.25	22.66	188.3	75.41
70	2010	27.98	211.7	63.5	26.8	204.2	61.27	25.47	195.8	58.74	23.98	186.1	55.83	22.3	174.9	52.46
80	1340	27.46	199.2	39.84	26.37	192.2	38.44	25.14	184.2	36.85	23.76	175.1	33.02	22.17	164.6	32.91
90	370	26.96	187.8	18.78	25.95	198.2	18.12	24.01	173.7	17.37	23.52	165.1	16.51	22.03	155.1	15.51
100	0	26.94	180.3	0	25.9	174	0	24.89	166.8	0	23.6	158.5	0	22.2	148.9	0



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