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Optimization of Air Compressor Location and Capacity to Improve Energy Efficiency and Performance in Manufacturing Facilities

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UNIVERSITY OF MIAMI

OPTIMIZATION OF AIR COMPRESSOR LOCATION AND CAPACITY TO IMPROVE ENERGY EFFICIENCY AND PERFORMANCE IN MANUFACTURING FACILITIES

By

Joel Zahlan

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Compressed air is the most expensive source of energy in manufacturing facilities and is responsible for over 10% of industrial energy usage in the U.S. Air compressors are only 11-13% efficient, making correct operation, sizing and location critical to energy conservation and optimization. Correct location and sizing of air compressors is a complex problem that requires consideration of multiple variables and uncertainties. The current method of locating and sizing air compressors is non-scientific and is largely based on unverified assumptions, availability of space and equipment, and convenience of location. This leads to significant increases in energy cost to manufactures. To resolve this, the authors propose a mathematical framework to identify the optimal air compressor location in a dynamic manufacturing facility. The introduced framework will minimize air leaks, pressure drops, and kW demand of the compressor, as well as correctly size the air compressor at each zone by determining the machine load profile and matching air supply and demand in the facility. The model will also consider location inconvenience and compressor noise in its decision process by allowing users to rate their preference level at each zone. Ultimately the goal is to propose a novel framework that finds the optimal air compressor location and size in a manufacturing facility while considering the user's preference for location and actual air losses in the system. The

proposed model will also take into account air demand, air leaks, and air pressure set point variations.

DEDICATION

I would like to dedicate this work to my father Joseph, mother Lena, sister Leona and companion Shireen. I thank them kindly for their unconditional love, support, resolute understanding, and steadfast patience during the coursework, research, and writing of this dissertation.

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CHAPTER 1: INTRODCUTION

Compressed air is often regarded as the fourth utility, and is one of the most critical applications in production and process environments (Air & Rollins, 1961; Foss, 2005). Seventy percent of all manufacturing facilities in the United States (U.S) rely on compressed air systems, and in many cases, failure of these systems leads to the shutdown of their entire manufacturing process (Cerci, Cengel, & Turner, 1995; U. S. D. O. Energy, 2014; XENERGY, 2001).

Compressed air accounts for approximately 10% of total industrial-energy use in the U.S, resulting in a staggering consumption of 90 billion kilowatt hour (kWh) per year (Laboratory, 1999; Senniappan, 2004; XENERGY, 2001). Furthermore, the total installed power capacity of compressed air systems in the U.S is estimated at more than 17 million horsepower (Talbott, 1993), accounting for about 16% of the industrial motor system energy use (XENERGY, 2002). However, a well-designed compressed air system is only about 11% efficient (Foss, 2005) with some estimates stating that poorly designed systems account for up to $3.2 billion in wasted utility payments in the U.S each year (C. A. Institute & Gas, 2012). This inefficiency, combined with the fact that compressed air is the most expensive form of energy to deliver (Fig. 1 (Yuan et al., 2006)), makes it critical for manufacturing facilities to seek to optimize compressed air energy efficiency and reduce cost.

In 2010 the U.S Energy information Administration (EIA) conducted a comprehensive Manufacturing Energy Consumption Survey (MECS) of 15,500 manufacturing facilities in the U.S (Administration, 2010). The data gathered from this survey was analyzed and a Manufacturing Energy and Carbon Footprint document was developed. This data shows that 14,064 TBtu is used in onsite energy at manufacturing facilities in the U.S. Of this 78% is Fuel (Natural Gas, Byproducts, Coal, etc.), 17% is Electricity, and 5% is Steam. Of the total energy use, 2012 TBtu is used by machine drive (compressed air, pumps, fans, etc.) and of this 17% or 341 TBtu accounts for Compressed Air. Furthermore 23% of total electrical consumption to machine drives in manufacturing facilities is consumed by air compressors (-Energetics, 2010). According to the Energy and Carbon Footprint document, of the 341 TBtu consumed by the air compressor systems, 87.7% or 299 TBtu is written off as losses, making compressed air the least efficient machine drive system in the surveyed facilities.

This further illustrates the importance of optimizing air compressor systems in manufacturing facilities. Unfortunately, compressed air systems are one of the least understood processes in most manufacturing facilities (Holdsworth, 1997; Risi, 1995), mainly due to the widespread misconception that compressed air is inexpensive.

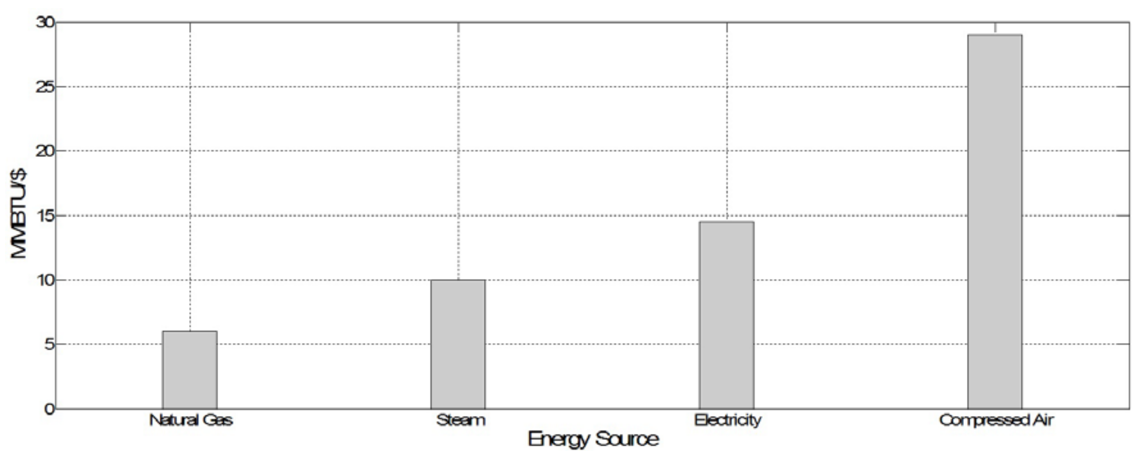

Fig. 1. Cost of Energy Delivery Modes (Yuan et al., 2006)

The energy associated with operating a compressed air system accounts for the largest cost to the user, often exceeding the initial cost of the compressor by up to five times over its lifespan (Cengel YA, 2000; Challenge, 2014; Group, 2001; Koski, 2002; Sweeney, 2002; Talbott, 1993). Sometimes this cost accounts for up to 70% of the total electric bill in manufacturing facilities (Chunha, 2007; Kaya, Phelan, Chau, & Ibrahim Sarac, 2002; Risi, 1995). These figures point out that by implementing energy conservation measures, facilities can experience substantial energy and cost savings.

1.1. Air Compressor System

Compressed air can be thought of as a form of energy used to operate machinery, equipment, and processes. It is less hazardous and more reliable than electrical energy and is used to power many tools and equipment in manufacturing industries. Although the use of air compressors are abundant, there is a vacuum of information on energy efficiency measures and correct operation of compressed air systems (Holdsworth, 1997; Joseph, 2004; Risi, 1995). Air is free, however the electricity used to compress air is not, and air compressor systems are only 11-13% efficient making them one of the most inefficient systems in facilities (Foss, 2005). It is also much more expensive than electrical energy and as Fig. 1 illustrates can be up to twice the cost of electrical energy. This is why it is important to use compressed air only when other alternative energy streams are unavailable or impractical. Fig. 2 below illustrates the conversion from atmospheric air to compressed air.

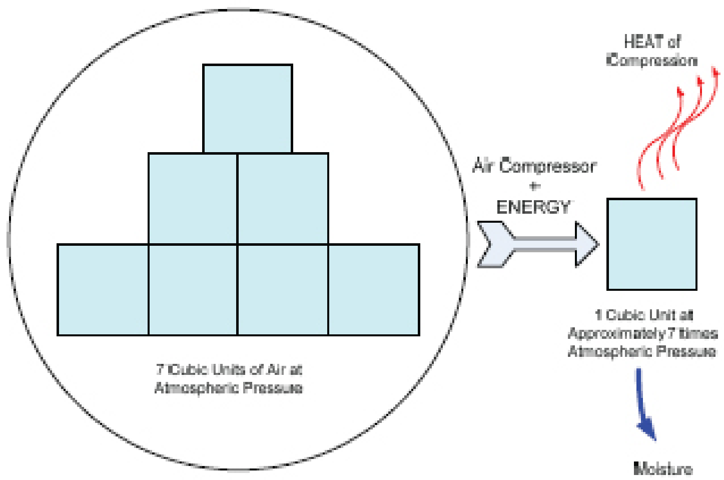

Fig. 2. Conversion of Atmospheric Air to Compressed Air (Chunha, 2007)

Using electricity a compressor takes approximately 7 volumes of air and compresses it into 1 volume of air at approximately 100 psig. This high pressure air is intern used in facilities to power pneumatic equipment and tools.

The DOE suggests that energy accounts for 76% of the total cost of running an air compressor over its life span. 12% accounts for equipment and installation, and the remaining 12% accounts for maintenance. Fig. 3 illustrates this below.

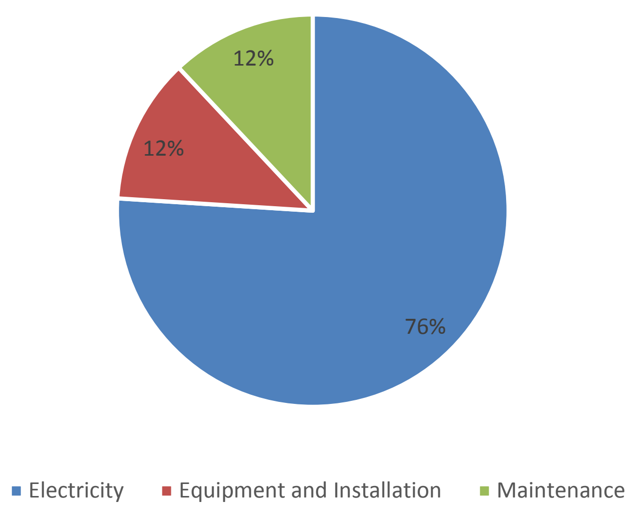

Fig. 3. Typical Lifetime Ownership Cost of Air Compressor (Radgen, 2006)

As Fig. 3 shows, it is very energy intensive to run an air compressor. This further illustrates the importance of optimizing compressed air systems. Savings in energy can have a significant impact on the lifetime ownership costs.

Fig. 4 shows the common components in an air compressor system. These components vary from system to system and are dependent on the quality of air required and working conditions in the compressor location.

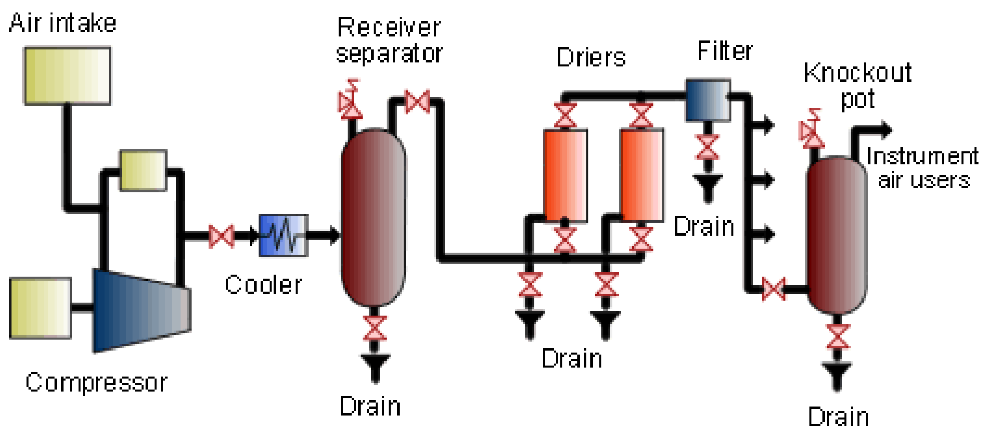

Fig. 4. Air Compressor System Components (Toolbox, 2012)

As it can be seen from Fig. 4 above a compressed air system often has an air compressor, a wet storage tank, dryers, a dry storage tank, pressure and flow valves, piping, and filters. These are all important components that can significantly affect the performance of the air compressor if selected incorrectly.

1.2. Overview of Air Compressor System Losses

The U.S Department of Energy (DOE) states that over 50% of industrial facility's compressed air systems harbor large energy opportunity savings with relatively short payback periods (UD-IAC, 2009; XENERGY, 2001). Energy savings from compressed air system improvements can range from 20-60% of electrical consumption, resulting in thousands, or even hundreds of thousands of dollars in potential annual savings (Efficiency & Energy; XENERGY, 2001). Some of these compressed air efficiency measures include reducing leaks, matching supply with demand, reducing pressure settings, reducing average inlet temperature by using outside air, improving air distribution systems, and optimizing air compressor location. Fig. 5 illustrates the energy savings opportunities of air compressor systems. As it can be seen, air leaks and air compressor system optimization account for the two largest losses in compressed air systems (McKane A, 2005). Sustaining energy savings in a compressed air system requires implementation of a continuous improvement application as further discussed in ISO50001(50001, 2011).

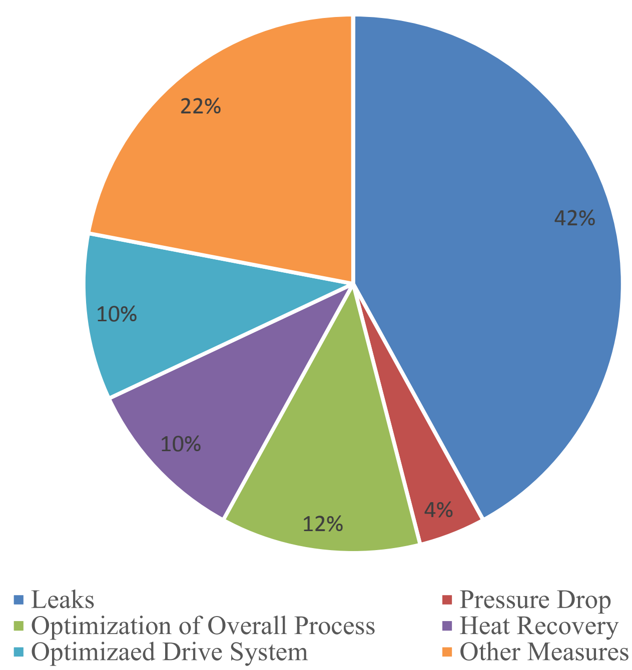

Fig. 5. Energy Savings Opportunities in Air Compressor Systems (McKane A, 2005)

Furthermore, studies suggest waste heat can also account for between 50-90% of compressed air energy losses. With newer and more efficient motors on the market, the motor efficiency of the air compressor can be improved 2-8% over most existing air compressors (Akbaba, 1999; Capehart, Turner, & Kennedy, 2006; Saidur, Rahim, & Hasanuzzaman, 2010). The energy wasted in a poorly designed and maintained compressed air system can account for up to 50% of the energy used by the air compressor and it is believed that half of all these losses can be saved through proper system design and energy conservation measures (Kaya et al., 2002). In this paper the authors propose an optimization model that minimizes the distance compressed air must travel to high demand and high pressure areas, reducing pressure drop and air leaks in the system, and improving the overall performance of the system.

The location of the air compressor in a facility is a critical step in ensuring an energy efficient air compressor system. By optimizing the air compressor location, the distance to the highest demand and pressure zones is minimized, total pressure drop and number of air leaks is reduced, and compressed air distribution system (piping) is improved. According to Scott Foss, President of Plant Air Technology, "The concept design or redesign [of air compressor systems] should be to minimize the highest amount of air mass or volume of air and the distance that the air must flow to support any part of the system from supply to demand (Foss, 2005)". In essence, the goal is to get the compressed air from the supply side to the demand side in the most efficient and cost effective manner to minimize losses in the system (air leaks and pressure drop).

1.3. Contributions of this Research

The current decision making process for air compressor location is non-scientific and based heavily on availability of space and convenience of installation. This has forced users to size their compressors well above the required air compressor capacity so as to account for the losses in leaks and pressure drop due to the inefficient design. Manufacturer's guides on air compressor sizing show that 25-30% more capacity is added as a factor of safety to make up for the unknown losses (K. Compressors, 2000; Q. Compressors). By optimizing air compressor location this factor of safety can be significantly reduced. The air compressor location problem, which thus far is unknown, will be investigated thoroughly in this research, allowing for a better understanding of the system as a whole and more precise and accurate sizing and estimations of the compressed air system.

Optimizing the location of the air compressor will also minimize the air pressure drop in the system. This will allow the compressor to run at lower pressure set-points which intern save in energy costs. In fact for every 2 psi reduction in pressure set point, the air compressor will experience a 1 % decrease in energy consumption. A secondary benefit of controlling pressure drop is a better and more reliable supply of compressed air. This will improve the performance of the pneumatic machines by reducing interruptions and down time.

Air leaks will also be minimized by optimizing the location of the air compressor. This will reduce compressed air on the demand side by reducing the volume of air required to run the pneumatic machines in the facility. Significant energy savings can be realized my minimizing air leaks in compressed air systems.

The model can also be used for correct compressor sizing and correct compressor control's selection in manufacturing facilities. In fact, air compressor sizing is critical to improved energy efficiency. In fact according to Saidur et al, "Two of the most important factors influencing the cost of compressed air are the type of compressor control and the proper compressor sizing. Oversized compressors and compressors operating in inefficient control modes have the highest unit energy and the highest annual operating costs" (Saidur et al., 2010). Over-sized compressors will also lead to an increase in demand (kW) which will lead to significant energy penalties to facilities (Avci, Erkoc, Rahmani, & Asfour, 2013). These all illustrate the importance of correct compressor sizing.

Furthermore piping cost will be minimized. Unnecessary piping will be eliminated during the design phase of the compressed air system, saving on the initial installation of the system. Compressed air will be delivered to machines in the most efficient effective manner.

Lastly the user's preference will be considered during the design phase of the compressed air system. Currently compressors are being placed in available spaces convenient to the user. However a systematic approach to identifying the most appropriate location for the air compressor as shown in this research can result in a more appropriate compressor location. This can result in minimum noise interruptions, more compressor accessibility and hence a better all-round system configuration.

This research can be further expanded to include other important variables such as waste heat, storage, and air intake temperature. Incorporating these variables into the optimization model can result in significant energy savings. The developed model in this research can act as a base for incorporating these variable.

CHAPTER 2: LITERATURE REVIEW

2.1 Air Compressor System Losses (Pressure Drop and Air Leaks)

By optimizing the air compressor location, the distance to the highest demand and pressure zones is minimized, total pressure drop and number of air leaks is reduced, and compressed air distribution system (piping) is improved.

2.1.1 Piping Optimization and Reduction in Pressure Drop

The rate of pressure drop in a system is directly correlated to the distance compressed air must travel to air demand areas. Poorly designed air compressor systems can experience up to a 60% drop in pressure at the point of use of air (Kaya et al., 2002; Saidur et al., 2010).Corrective actions must be taken on the air compressor system to improve pressure drop across the system. The DOE and Compressed Air Challenge state that one of the leading causes of pressure drop is in proper design of the compressed air distribution system. They also state that it is critical to reduce the distance the air travels thought he distribution system (Laboratory). The Canadian Energy Authorities (CEA) suggest that by doubling the size of piping, a compressed air system will experience ¼ the pressure drop. This intern is a 75% savings in pressure drop across the system (Chunha, 2007). Figure 6 & 7 illustrates the pressure drop for different pressure set points across different sized piping per 100m (330ft) of piping.

Pressure Drop	NOTE! Pressure drops above 1 kg/cm2 (14-15 psi) are in general not relevant.										
Volume Flow FAD		NPS Pipe Size / Innside diameter (mm)									
		1/2		3/4		1		1 1/4		1 1/2	
		15.798		20.93		26.645		35.052		40.894	
(m3/min)	(cfm)	(kg/cm2)	(psi)	(kg/cm2)	(psi)	(kg/cm2)	(psi)	(kg/cm2)	(psi)	(kg/cm2)	(psi)
0.1	4	0.02	0.22								
0.2	7	0.06	0.80	0.01	0.20						
0.3	11	0.12	1.69	0.03	0.41	0.01	0.12				
0.4	14	0.20	2.88	0.05	0.71	0.01	0.21				
0.5	18	0.30	4.35	0.07	1.07	0.02	0.32	0.01	0.08		
0.6	21	0.43	6.09	0.10	1.49	0.03	0.45	0.01	0.11		
0.7	25	0.57	8.10	0.14	1.99	0.04	0.59	0.01	0.15		
0.8	28	0.73	10.38	0.18	2.54	0.05	0.76	0.01	0.19	0.01	0.09
0.9	32	0.90	12.90	0.22	3.16	0.07	0.95	0.02	0.24	0.01	0.11
1	35	1.10	15.68	0.27	3.84	0.08	1.15	0.02	0.29	0.01	0.13
1.2	42			0.38	5.38	0.11	1.61	0.03	0.41	0.01	0.19
1.4	49			0.50	7.16	0.15	2.14	0.04	0.54	0.02	0.25
1.6	56			0.64	9.16	0.19	2.74	0.05	0.70	0.02	0.32
1.8	64			0.80	11.40	0.24	3.41	0.06	0.87	0.03	0.40
2	71			0.97	13.85	0.29	4.14	0.07	1.05	0.03	0.49
2.2	78			1.16	16.52	0.35	4.94	0.09	1.25	0.04	0.58
2.4	85			1.36	19.40	0.41	5.80	0.10	1.47	0.05	0.68
2.6	92					0.47	6.73	0.12	1.71	0.06	0.79
2.8	99					0.54	7.72	0.14	1.96	0.06	0.91
3	106					0.61	8.77	0.16	2.23	0.07	1.03
3.5	124					0.82	11.66	0.21	2.96	0.10	1.37
4	141					1.05	14.93	0.27	3.79	0.12	1.75
4.5	159					1.30	18.57	0.33	4.71	0.15	2.18
5	177							0.40	5.73	0.19	2.65
5.5	194							0.48	6.83	0.22	3.16
6	212							0.56	8.02	0.26	3.71
6.5	229							0.65	9.30	0.30	4.30
7	247							0.75	10.67	0.35	4.94
7.5	265							0.85	12.12	0.39	5.61
8	282							0.96	13.66	0.44	6.32
8.5	300							1.07	15.28	0.50	7.07
9	318							1.19	16.99	0.55	7.86
9.5	335							1.32	18.77	0.61	8.69
10	353							1.45	20.64	0.67	9.55
15	530									1.42	20.22

Fig. 6. Pressure Drop Across Piping (0.5” to 1.5”) (McKane A, 2005)

Pressure Drop		NOTE! Pressure drops above 1 kg/cm2 (14-15 psi) are in general not relevant.									
Volume Flow FAD		NPS Pipe Size / Innside diameter (mm)									
		1 1/2		2		2 1/2		3		4	
		40.894		52.501		62.713		77.928		102.26	
(m3/min)	(cfm)	(kg/cm2)	(psi)	(kg/cm2)	(psi)	(kg/cm2)	(psi)	(kg/cm2)	(psi)	(kg/cm2)	(psi)
0.8	28	0.01	0.09								
0.9	32	0.01	0.11								
1	35	0.01	0.13								
1.2	42	0.01	0.19								
1.4	49	0.02	0.25	0.01	0.07						
1.6	56	0.02	0.32	0.01	0.09						
1.8	64	0.03	0.40	0.01	0.11						
2	71	0.03	0.49	0.01	0.14						
2.2	78	0.04	0.58	0.01	0.17						
2.4	85	0.05	0.68	0.01	0.20	0.01	0.08				
2.6	92	0.06	0.79	0.02	0.23	0.01	0.09				
2.8	99	0.06	0.91	0.02	0.26	0.01	0.11				
3	106	0.07	1.03	0.02	0.30	0.01	0.12				
3.5	124	0.10	1.37	0.03	0.39	0.01	0.16				
4	141	0.12	1.75	0.04	0.50	0.01	0.21				
4.5	159	0.15	2.18	0.04	0.63	0.02	0.26	0.01	0.09		
5	177	0.19	2.65	0.05	0.76	0.02	0.31	0.01	0.11		
5.5	194	0.22	3.16	0.06	0.91	0.03	0.37	0.01	0.13		
6	212	0.26	3.71	0.07	1.06	0.03	0.44	0.01	0.15		
6.5	229	0.30	4.30	0.09	1.23	0.04	0.51	0.01	0.17		
7	247	0.35	4.94	0.10	1.42	0.04	0.58	0.01	0.20		
7.5	265	0.39	5.61	0.11	1.61	0.05	0.66	0.02	0.22		
8	282	0.44	6.32	0.13	1.81	0.05	0.75	0.02	0.25		
8.5	300	0.50	7.07	0.14	2.03	0.06	0.83	0.02	0.28	0.01	0.07
9	318	0.55	7.86	0.16	2.25	0.06	0.93	0.02	0.31	0.01	0.08
9.5	335	0.61	8.69	0.17	2.49	0.07	1.02	0.02	0.35	0.01	0.09
10	353	0.67	9.55	0.19	2.74	0.08	1.13	0.03	0.38	0.01	0.10
15	530	1.42	20.22	0.41	5.80	0.17	2.38	0.06	0.80	0.01	0.21
20	706			0.69	9.87	0.28	4.06	0.10	1.37	0.02	0.35
25	883			1.05	14.92	0.43	6.13	0.15	2.07	0.04	0.53
30	1059			1.46	20.90	0.60	8.59	0.20	2.90	0.05	0.75
35	1236					0.80	11.43	0.27	3.86	0.07	0.99
40	1412					1.03	14.63	0.35	4.94	0.09	1.27
45	1589					1.28	18.20	0.43	6.14	0.11	1.58
50	1765							0.52	7.46	0.13	1.92
55	1942							0.62	8.90	0.16	2.29
60	2118							0.73	10.46	0.19	2.69
65	2295							0.85	12.13	0.22	3.12
70	2471							0.97	13.91	0.25	3.57
75	2648							1.11	15.80	0.28	4.06
80	2824							1.25	17.81	0.32	4.58
85	3001							1.40	19.92	0.36	5.12
90	3177									0.40	5.69
95	3354									0.44	6.29
100	3530									0.48	6.92
150	5295									1.03	14.64

Fig. 7. Pressure Drop Across Piping (1.5” to 4”) (McKane A, 2005)

As it can be seen from both Figures 6 & 7 significant savings in pressure drop can be realized by minimizing the distance compressed air must travel to the desired locations. Journal publications by D. Kaya et al., Mark D’Antonio et al, R. Saidur et al all agree that one of the largest opportunities for saving energy in a compressed air system is the reduction of pressure drop across piping. This can be done by both correctly sizing the compressed air piping and by minimizing distance to the various demand locations. Table 1 below shows the suggested piping size for various flow rates.

Table 1 (Chunha, 2007)

CAE recommended piping size for compressed air systems. As the flow rate (scfm) increases, the size of the recommended piping increases as well. Pressure drop is proportional to distance and flow rate (scfm).

Pipe Diameter (inches)	Equivalent Flow (scfm)
1	36
2	263
3	431
4	909
6	2,679
8	6,757
10	14,286

The relationship between pressure drop and energy consumption is quite clear. The more pressure drop in the Air compressor system, the higher the pressure set-point at the air compressor. In fact, for every 2 psi increase in the air pressure set point, the air compressor experiences a 1 % increase in energy consumption (Christina & Worrell, 2008; K. Compressors, 2000; D'Antonio, Epstein, Moray, & Schmidt, 2005). In order to ensure that machines run uninterrupted, the air compressor must match the pressure requirements of each machine in the facility.

Pressure drop can be calculated theoretically using equations 1 and 2 (Air & Rollins, 1961) below:

$$f = \frac{0.1025LQ^2}{rd^{5.31}} \tag{1}$$

$$r = \frac{Pressure\ (psi) + Atmospheric\ Pressure\ at\ Location\ (psi)}{Atmospheric\ Pressure\ at\ Location\ (psi)} \tag{2}$$

where f = pressure drop (psi), L = total length of pipe (ft) to demand zones, Q = cubic feet of free air per second plus total air leaks per zone location calculated in the previous section, r = ratio of compression (from free air) at entrance of pipe, d = actual internal pipe diameter.

2.1.2 *Reduction in Air Leaks*

Air leaks are the single greatest cause of energy loss in manufacturing facilities with compressed air systems and account for 20-50% of compressed air losses (McKane, 2008; Risi, 1995; Saidur et al., 2010). The majority of air leaks occur in Air leaks, occur at the joints, flange connections, elbows, reducing bushes, sudden expansions, valve systems, filters, hoses, check valves, relief valves, extensions, and the equipment connected to the compressed-air lines (Saidur et al., 2010). The amount of air lost to leaks is dependent on the line pressure of the pipe and the temperature of the air at the point of the leak (Saidur et al., 2010).By minimizing distance to the highest pressure and volume locations, the goal is to minimize the number of joints in piping and the distance air must travel to the highest pressure and volume equipment.

Table 2 below illustrates the cost of having different sized compressor air leaks in a system. As it can be seen, there is a significant cost associated with having air leaks in a system. It is important to minimize air leaks in a system by both fixing existing leaks, and minimizing the piping required to supply compressed air to the demand locations. This will intern result in less potential for air leaks in a system.

Table 2

Annual Cost of Compressed Air Leaks Based on $0.10/kWh Electricity Cost (Challenge, 2014).

	1 Shift (2250 hrs)	**2 Shifts (4250 hrs)**	**3 Shifts (8400 hrs)**
1/16" leak	$200	$380	$750
¼" leak	$3,210	$6,070	$11,990
3/8" leak	$7,230	$13,650	$26,980
½" leak	$12,820	$24,210	$47,850

The resulting air leaks in a system can be calculated theoretically using equation (3) (Air & Rollins, 1961):

$$V_f = \frac{NL * (T_i + 460) * \frac{P_l}{P_i} * C_1 * C_2 * C_d * \frac{\pi D^2}{4}}{C_3 * \sqrt{T_i + 460}} \tag{3}$$

where Vf = volumetric flow rate of free air, cubic feet per minute, T_i = temperature of the air at the compressor inlet (yearly average), 76.5 °F, P_l= line pressure at leak in question, psia (100 psia), P_i = inlet (atmospheric) pressure, psia, $C1$= isentropic sonic volumetric flow constant, 28.37 ft/sec-°R0.5, $C2$= conversion constant, 60 sec/min, Cd= coefficient of discharge for square edged orifice1, 0.8 no units, π = Pythagorean constant, 3.1416, $C3$=

conversion constant, 144 in2/ft2, T_l= average line temperature, 76.5°F (assumed to be the same as temp at the leak), D = leak diameter and $NL =$ the number of leaks in the system.

Other less complex formulas for calculating leaks can be seen in equations 4 and 5 below. Equation xx is used to estimate leaks in a load/unload or start/stop air compressor. To use this formula, the air compressor is run with no demand in the facility. The compressor is timed for how long it takes to load and unload. The compressor will load and unload because the existing ar leaks in the facility will cause the compressor to cycle when the pressure in the system drops below the designated pressure set point. The formula used to calculate leaks in the scenario described is shown as follows(Laboratory):

$$Leakage\ (\%) = \left[\frac{T*100}{T+t}\right] \tag{4}$$

Where T is the on-load time of the air compressor expressed in minutes and t is the off-load time of the air compressor expressed in minutes. In this case leakage is expressed as a percentage.

Equation 5 can be used in systems with other control strategies. In this case the air compressor is run at normal operating pressure (P_1). The amount of time T is then measured for the pressure to drop to a lower pressure point (P_2). The leakage rate can be calculated as follows(Laboratory):

$$Leakage\ (cfm\ free\ air) = \frac{V*(P_1 - P_2)}{T*14.7} * 1.25 \tag{5}$$

2.1.3 *Air Compressor Capacity and Sizing*

With respect to compressor sizing, Kaeser Compressor's *Designing your Compressed Air System* suggests using air demand and load factor of machines to calculate appropriate compressor size, however, losses, errors and reserves are accounted for by adding an additional 45% cfm to the system (K. Compressors, 2000). Similarly, Quincy Compressors and Champion both suggest over sizing air compressors by an additional 25% (Q. Compressors; Sill, 2008). Altlas Copco is more conservative, adding an additional 10% - 20% to account for leaks (Copco, 1998). As the DOE explains “Oversized air compressors are extremely inefficient because most compressors use more energy per unit volume of air produced when operating at part-load”(Efficiency & Energy) . Furthermore, D.M. McCulloch and B. Scales recommend not adding an additional “fudge factor” as it leads compressors operating less efficient and at less than full load (McCulloch & Scales, 2005).

2.2 Air Compressor Location Problem

There are many research articles and journal publications that describe the energy savings potential and recommend energy saving strategies for compressed air systems. Most of these publications discuss the importance of air compressor location and placement, however none of these actually measure the savings potential of optimizing air compressor location, and the resulting efficiency gain from doing so. Journal publications by D. Kaya et al., Mark D’Antonio et al, R. Saidur et al., give an overview of compressed air conservation measures including air leak and pressure drop reduction, complete with analysis, potential energy and cost savings of minimizing these losses. However none quantify the effect air compressor location has on energy cost and losses. (D’Antonio et al., 2005; Kaya et al., 2002; Saidur et al., 2010). Compressed Air and Gas Handbook (Air & Rollins, 1961), The Compressed Air Industry Sourcebook, published by the DOE (C. A. C.-U. S. D. o. Energy, 1998)and the Compressed Air Challenge Group (CAC) all suggest that pressure drop and air leaks are directly correlated to system design. They indicate that distance to air compressor demand is important, and introduce equations for calculating air leaks and pressure drop that suggest loses have a linear correlation to distance.

With respect to air distribution and air compressor optimization, Kaeser Compressor's *Designing your Compressed Air System* states that it is essential to reduce the distance that air must travel, and discusses the importance in minimizing pressure drop and air leaks in the system, however there is no suggestion on potential (K. Compressors, 2000). Furthermore, R.S. Foss, The Compressed Air and Gas Institute (CAGI), CAC and the DOE all explain that minimizing the distance of compresse1/4" leak d air to the highest demand locations is the key in creating an optimal distribution system and reducing energy costs (C. A. C.-U. S. D. o. Energy, 1998; Foss, 2005; C. A. a. G. Institute, 2002). Through information provided by the manuscripts and texts listed above, it is clear that the location of air compressors in manufacturing facilities plays a critical role in the overall optimization of the air compressor system. The authors' experience visiting over 270 manufacturing facilities in Florida and conducting dozens of comprehensive air compressor audits has also revealed the importance of air compressor location in manufacturing facilities.

To best of our knowledge, the air compressor location optimization problem is first addressed by Zahlan and Asfour (Joel Zahlan & Asfour, 2015) and Zahlan et. al (J Zahlan, Avci, & Asfour, 2014). These papers introduce two mathematical models for finding the optimal air compressor location in a manufacturing facility, using distance, pressure, and air demand. The models differ in that Zahlan et. al does not consider the user's preference, the load factor of machines, or fluctuations in air demand. Zahlan and Asfour introduce a more sophisticated mathematical model capable of finding the optimal location while considering the user's preference, however energy losses in both models are measured as a function of distance to air demand locations. Moreover, uncertainties in key parameters, such as the number and the size of air leaks and fluctuations of pressure are not considered. These parameters are assumed to be constant and unchanged in the analysis, which reduces the precision of the model as leaks and pressure fluctuations are unpredictable in the system. Finally, while the models in these papers tackle the optimal location problem, they do not explicitly address the issue of correct air compressor sizing, which is critical for maximizing energy savings.

In this paper, a simulation optimization framework is proposed for joint optimization of air compressor location selection and capacity sizing in a dynamic manufacturing facility. The introduced framework minimizes air leaks and pressure drops, by reducing the distance compressed air must travel to demand locations. It will also correctly size the air compressor by matching air supply with the air demand in the facility. The model will identify the kW at each zone required to meet air demand at any given point. Uncertainty is incorporated into the model for air pressure fluctuations, load variation of machines, and the number and size of air leaks. This allows the model to consider most realistic scenarios in the facility setup. Machines and their associated cfm demand are taken from literature (Rand, 2013), and are representative of typical machines in the U.S. manufacturing facilities.

The proposed model allows the user to assign a level of preference to each location, enabling them to incorporate noise and inconvenience of location into the decision making process. Finally a sensitivity analysis is conducted to understand the relationship between air compressor energy consumption and the user's location preference. Ultimately the goal is to propose a novel mathematical formulation that finds the optimal air compressor location and size in a manufacturing facility, while considering the users preference, air losses and several important uncertainties discussed further in the sections below.

CHAPTER 3: SIMULATION-OPTIMIZATION MODEL (FIRST MODEL)

The key to finding the optimal air compressor location in a facility is to first identify and prioritize the determinants for air compressor energy consumption and the user's location preference. Consequently, two mathematical models are developed to determine the optimal air compressor location in a respective facility that minimizes total energy consumption while considering the user's location preference. The following sections discuss the development of the two proposed multi-objective models and their derivatives.

3.1 Mathematical Formulation of the Proposed Model (First Model)

The proposed model has two goals: (1) minimizing the total energy consumption of the air compressor and hence the energy cost for the facility, (2) maximizing the user's preference for the air compressor location in the facility. The two goals are incorporated into a single objective function via a weighted sum of:

1) the product sum of i) distance between the air compressor location and each defined facility zone, ii) the compressed air demand (air volume * load factor), and iii) pressure at each respective zone.

2) a predetermined parameter that defines the level of user preference for air compressor location, while incorporating zone availability with respect to zone temperature and existence using a binary constraint.

The resulting mathematical formulation of the proposed model can be written as follows:

$$\hat{z} = \underset{z \in \{1,2,\ldots,N\}}{\operatorname{argmin}} \left([\alpha I_z + 1] \times \sum_{i=1}^{N} \sum_{j=1}^{R_i} \left[\left(P_{ij} * \dot{V}_{ij} * l_{ij} \right) * D_{iz} \right] \right) + B_z \tag{6}$$

Subject to:

$$B_z = \begin{cases} 0 & \text{when zone is feasible} \\ \infty & \text{otherwise} \end{cases}$$

$$175\ cfm \le \dot{V}_{ij} \le 250\ cfm$$

$$0 \le D_{iz} \le d$$

$$0 \le P_{ij} \le 125\ psi$$

$$z \in \{1,2, \ldots, N\}$$

$$0 \le l_{ij} \le 1$$

where:

$$E = \sum_{i=1}^{N} \sum_{j=1}^{R_i} \left[\left(P_{ij} * \dot{V}_{ij} * l_{ij} \right) * D_{iz} \right] \tag{7}$$

$$W = \alpha E \tag{8}$$

$$D_{iz} = |X_i - X_z| + |Y_i - Y_z| \tag{9}$$

$$I_z = 1, 2, 3, 4, 5$$

$$0 \le \alpha \le 1$$

where N and R_i represents the number of zones in the facility and the number of machines in each zone respectively. $\underset{z \in \{1,N\}}{\operatorname{Argmin}}$ is the value of z for which the objective function attains the minimum output. For machine j in zone i, P_{ij} is the pressure (psig) requirement, V_{ij} is the volume (cfm) requirement, and l_{ij} is the load factor (%). D_{iz} is the distance between zone i, and the potential air compressor zone z. The user preference for air compressor location of zone z is represented by a multi-level user preference index, I_z, which is captured by the weight, W, in the objective function. W is a linear function of the energy consumption term E. $\alpha \in [0, 1]$ designates the fraction of E used as W. When $\alpha = 0$ the model will only minimize the energy consumption of the air compressor. When $\alpha = 1$ the model will minimize both energy consumption and user preference equally. Clearly higher values of W indicate cases where the user preference for air compressor location is more important to the user than energy efficiency. A higher W normally leads to greater air compressor energy consumption within a given facility (See Section 3.2). Additionally, the feasibility of the air compressor location at zone z is represented by a binary constraint, B_z, which considers zone temperature, zone existence, and zone feasibility. d is the maximum possible rectilinear distance within the modeled facility. The optimal zone for air compressor location is the zone z that minimizes the above function (6).

Figure 8 below illustrates the modeled problem.

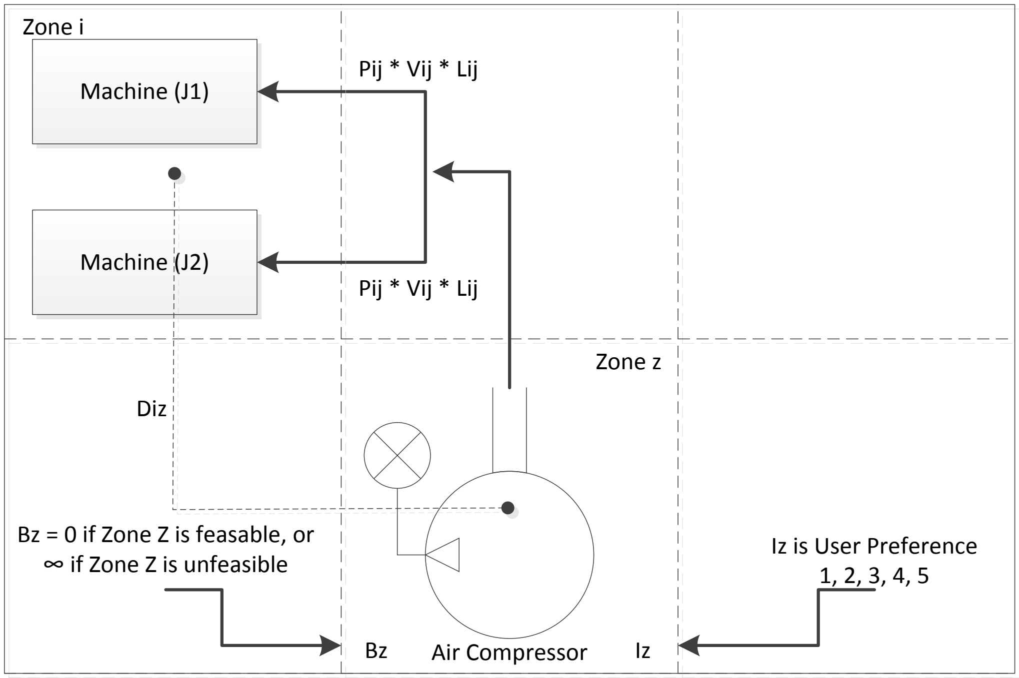

Fig. 8. Air Compressor Relationship of Variables in Optimization Model

The air compressor positioned at zone z feeds air to machine $J1$ & $J2$ in zone i. As seen, D_{iz} is the rectilinear distance from zone z to zone i, $I_z$is the user preference level at zone z, and B_z is the binary constraint at zone z.

The following sections detail the various terms in the mathematical model.

3.1.1 Distance Term in Mathematical Model

Minimizing distance (D_{iz}) from the air compressor location (z) to each air demand location (i) is one of the most significant determinants in minimizing air compressor energy consumption.

In order to understand the distance term (D_{iz}) in the function, it is important to first describe how the model addresses the facility to be optimized. Facility size is given as an input of length and width in feet. The facility is then divided into zones for further analysis. Zone size selection will be discussed in section 2.3 of this paper.

A rectilinear zone-to-zone distance matrix is established to determine the distance compressed air must travel to each demand zone i. The rectilinear distance between zones is used because it is the most representative of a compressed air piping distribution layout. For facilities that are not rectangular in shape, zones that do not fall within the established rectangular shape can be eliminated from analysis by using a binary constraint (B_z) which will be discussed later in this section. Rectilinear distance between each zone is calculated using Equation (10) (Chalmet, Francis, & Kolen, 1981; Hanan, 1966).

$$D_{iz} = |X_i - X_z| + |Y_i - Y_z| \tag{10}$$

By minimizing distance to air demand locations, the compressed air system will experience a reduction in pressure drops and air leaks. Both pressure drop and air leaks have a significant effect on energy consumption in a compressed air system.

Pressure drop is linearly correlated to distance and can be expressed in equation (11) below:

$$dp = \frac{7.57 * q^{1.85} * L * 10^4}{d^5 * p} \tag{11}$$

where *dp* is the pressure drop in the pipe (psi), q is the volume flow rate (m3/min), L is the length of the pipe (m), d is the diameter of the pipe (mm), and p is the pressure in the pipe (kg/cm2). If all variables in equation 11 are kept constant and the length of the pipe is increased, it can be noticed that the pressure drop *dp* in the pipe will increase linearly.

Air leaks are the second significant loss in compressed air systems. It is suggested in many areas that on average, compressed air systems lose up to 40% of the air produced to air leaks and are the single greatest loss of energy in compressed air systems at manufacturing facilities (Saidur et al., 2010). In general air leaks, occur at the joints, flange connections, elbows, reducing bushes, sudden expansions, valve systems, filters, hoses, check valves, relief valves, extensions, and the equipment connected to the compressed-air lines (Saidur et al., 2010). In this paper we focus mainly on the air leaks occurring at the joints of pipes. Since the number of machines in the modeled facility does not change, the assumption is made that hoses, check valves, equipment connected to compressed-air lines, etc. remain the same. The only significant variable that changes is the length of piping required to supply compressed air to air demand areas. The further the distance the more piping is required, which in turn means the more joints and the higher chance of leaks. The number of leaks is conservatively estimated at 1/64 inch leak per 100 ft of piping. The assumption is made that air leaks in the system are linearly proportional to distance.

Therefore it is suggested that reducing the distance compressed air must travel to high demand and high-pressure zones will result in a more efficient compressed air system with fewer pressure and leak losses.

3.1.2 Energy Consumption Term in Mathematical Model

The energy consumption component of the proposed optimization model is derived in part by calculating effective demand and pressure requirements of the machines within the system. Effective air demand is the product of Volume (V_{ij}), and Load Factor (l_{ij}) of a machine (K. Compressors, 2000). Effective demand is used in industry to establish flow demand for compressed air systems (Brown, 1997; K. Compressors, 2000). Using the effective demand at each zone, the objective function establishes a demand profile for the entire facility. The function is then able to position the air compressor closest to the zones with the highest effective demand rate.

The load factor (l_ij) of the machines at each zone are in part determined by data from Ingersoll Rand (See Table 1) [49]. The percent load of a machine in the facility determines the length of time the machine is on, and directly affects demand (cfm) in the facility. Obviously higher machine loads will result in higher compressed air demand making them both directly correlated.

The load of each machine in a facility can be very dynamic and is dependent on various factors such as production schedules, product demand, availability of raw materials, etc. Although Ingersoll Rand suggests loads for various types of machines, there can still be some unexpected load change or uncertainty that can affect air demand in a facility. In order to accurately model the load of each machine and to account for uncertainty in the facility, the model varies the load factor of each machine, using the suggested loads provided by Ingersoll Rand as guidance. This will be discussed further in sections 2.4 and 3.3 of the paper.

Air pressure (P_{ij}) is the second significant determinant of air compressor energy consumption. Air pressure is linearly correlated to energy consumption; the higher the air compressor pressure set point or machine pressure requirement, the more energy required. In fact, for every 2 psi increase in the air pressure set point, the air compressor experiences a 1 % increase in energy consumption (Christina & Worrell, 2008; K. Compressors, 2000;

D'Antonio et al., 2005). In order to ensure that machines run uninterrupted, the air compressor must match the pressure requirements of each machine in the facility.

The proposed energy consumption term in the optimization model is:

$$P_{Watts} = P_{ij} * \dot{V}_{ij} * L_{ij} \tag{12}$$

To develop this term it is important to first derive power (P_{watts}). To move compressed air from point A to point B work (W) must be performed. To move an object, the linear path integral of the force component must be taken in the direction of motion, multiplied by the distance moved as shown in equation 13.

$$W = \int_A^B \overrightarrow{F} * \overrightarrow{ds} \tag{13}$$

where W is work, F is force, and d is distance.

We then replace $\overrightarrow{F}$ in equation 13 with the force per unit charge in an electric field (E).

$$\vec{E} = \frac{\vec{F}}{Q} \tag{14}$$

where E is the resulting electric field, and Q is the unit charge.

Combining Equation 13 and 14 gives us the expression of work in an electric field to move a charge Q from point A to point B (15):

$$W = \int_A^B \overrightarrow{E} * Q * d\vec{s} \tag{15}$$

From equation 15 we can develop the expression of *Power* to move Q in the electric field from one point to the next.

$$Power = \frac{dW}{dt} = \int_A^B \frac{dQ}{dt} * \overrightarrow{E} * d\vec{s} = i * \int_A^B \vec{E} * d\vec{s} \qquad (16)$$

where i is current and $i = dQ/dt$. The linear integral displayed in equation 16 is the Voltage term(V).

$$V = \int_A^B \vec{E} * d\vec{s} \qquad (17)$$

By substituting voltage into 17 we get the expression for instantaneous power.

$$Power = V * i \qquad (18)$$

Through some dimensional analysis the expression of Power is then converted into watts as shown in equation 19.

$$Power = \left[\frac{\left(\frac{\frac{Kg * m^2}{s^2}}{s}\right)}{C}\right] * \left[\frac{C}{s}\right] = \frac{\frac{kg * m^2}{s^2}}{s} = \frac{N * m}{s} = Watt \qquad (19)$$

where C is coulombs, N is Newton's and is N = $kg * m/s\text{^}2$. *m* is meters and *s* is seconds. Lastly P_{Watts} is expanded to result in the Pressure (P_{ij}) times Volume ($\dot{V}_{ij}$) function in the optimization model.

$$P_{Watts} = \frac{N * m}{s} = \left[\frac{N}{m^2}\right] * \left[\frac{m^3}{s}\right] \qquad (20)$$

$$P_{Watts} = P_{ij} * \dot{V}_{ij} \qquad (21)$$

3.1.3 User Preference Index and Binary Constraint in the Model

Although the most energy efficient location results in the least cost to the facility, our goal in the mathematical model is to maximize as well the user satisfaction. By incorporating user preference, the user can identify preferred areas for air compressor location. The introduced user preference index and its associated weight add an additional degree of freedom to effectively manage the trade-off between the energy cost and the user's location preference.

To model the user's preference for air compressor location in the facility, a five level user *preference index* I_z combined with a binary constraint B_z is introduced (See Equation 1). The model uses an *a priori* preference (Ngatchou, Zarei, & El-Sharkawi, 2005) approach to represent the preference level of I_z at each zone.

The user preference index has a range from 1 to 5 where 1 is the most desired (least inconvenient) location and 5 is the least desired (most inconvenient) location for the user. For example, zones that fall next to production lines or office space will normally be expected to have a preference index value of 5 (meaning most inconvenient), while zones that fall directly on the outside of the facility wall, or in an available space in the facility, will be assigned a preference index value closer to *1*.

The binary constraint B_z is introduced to determine zone feasibility in the objective function. For this model, the proposed function assigns 0 to B_z if a zone that exists within the true facility layout is feasible, and has a temperature that always falls within the defined temperature range $15°F \leq Tz \leq 125°F$ (Company, 2005), or ∞ otherwise. If the compressor is located inside the facility, but has an outside air intake, the outside temperature of the location is used. If the maximum or minimum outside air temperature

(OAT) of the facility location does not fall within the prescribed temperature range for effective air compressor performance, the model will neither assign the compressor to an outdoor zone nor any prospective zone using an outside air intake. B_z also indicates if a zone is feasible or not. Once a facility is initially defined as a rectangular shape, if the facility is actually an "L" shaped facility, the zones located outside the "L" shape will be designated as non-applicable zones by the function. Additionally, if a zone falls on a production line it is considered infeasible for air compressor location.

3.1.4 Assumptions in Mathematical Model

In this section the assumptions used in the optimization-simulation model are discussed. Some assumptions are mentioned and discussed further in their respective sections. This is done to give more background to the assumption.

It is assumed that the compressed air flow in the pipe will only be laminar and not turbulent. This assumption is made since the model is only designed for one compressor in one location and therefore there should not be any friction causing turbulence in the pipes. Furthermore the model considers a well-designed piping system that minimizes T-joints and dead ends hence reducing the turbulence in the piping structure significantly.

It is also assumed that for every 2 psi increase in the air pressure set point, the air compressor experiences a 1 % increase in energy consumption (Christina & Worrell, 2008; K. Compressors, 2000; D'Antonio et al., 2005). This is discussed further in section 2.1.1. Since the number of demand locations and number of machines do not change in the model, piping drops are assumed to be uniform for each compressor location and do not affect the pressure drop analysis.

The assumption is made that the relevant air leaks only occur at joints in piping and occur at a rate of 1/64 inch leak per 100ft of piping. Hence the assumption that air leaks in the system is linearly proportional to distance. This is discussed further in section 2.1.1.

Lastly we assume that the compressed air system is set-up in the same way at each selected zone. Therefore the performance of the air compressor and the cost of the set-up are not affected the selected zone.

3.1.5 *Zone Size and Characteristics*

For the purpose of this research, a zone size is defined as a 20ft x 20ft square. This zone size was selected based on three considerations: 1) The selected zone size must be representative of an air compressor room size at a typical manufacturing facility in the United States, 2) the selected zone size must representative of a typical area of air demand (machine area size) in a manufacturing facility, and 3) the selected zone size must not be too small as this will lead to unnecessary complications in the optimization process, making it inefficient and sometimes not feasible. On the other hand, if a zone size is too large, it will not be representative of the air compressor demand and pressure attributes for all parts of the zone area.

3.1.5.1 *Compressor Room Size and Layout*

It is recommended by OSHA that there be 4ft of space around each piece of electrical equipment in the compressor room for maintenance purposes and accessibility. The exact layout of the compressed air system will depend on several elements such as air quality, air supply, storage requirements, etc. Fig. 9 below shows the typical setup in an air compressor room. Compressor Rooms typically consist of the following equipment: Air

Compressor, Wet Storage Receiver, Air Dryer, Dry Storage Receiver, and filtration equipment (K. Compressors, 2000).

As it can be seen in Fig. 9, there is 4ft of distance between each piece of equipment, and each piece of equipment and the walls of the room. Following this criterion illustrates that the zone size of 20ft x 20 ft is representative of a compressor room in a manufacturing facility. For the purpose of this research the equipment is set-up in a square footprint, however this equipment can be set-up whichever way the facility chooses to as long as it meets the OSHA suggested 4ft spacing.

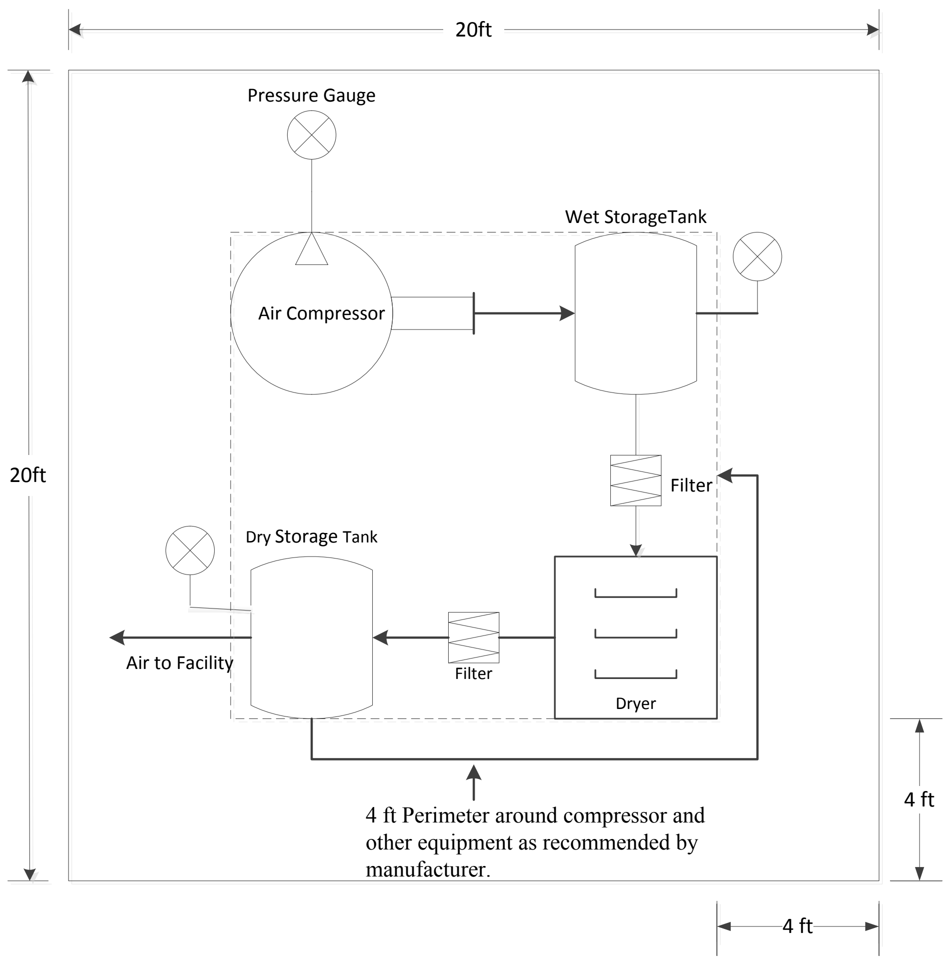

Fig. 9. Typical Air Compressor Room Set-up and Foot Print

3.1.5.2 *Typical Area of Air Demand (Machine Sizes)*

Typical area of zone i for air demand also plays an important role in the selection of zone size. According to the “Facilities Design” book by Sunderesh S. Heragu we see that the average foot print of a machine in a production area typical facility is 346 sq/ft (Heragu, 2006). This is a bit less than the 400 sq/ft per zone size suggested in this paper.

3.1.5.3 *Zone Size Representation*

Finally we test the selected zone size to ensure that the placement of the air compressor at any part of the zone will have negligible pressure drop and air leaks from one point to the next. Using Pythagoreans Theorem we calculate the longest point air must travel from one corner of the zone to the next is 28ft. Based on the air leak assumption stated above, air leaks on the joints of pipes can be assumed negligible. Furthermore the calculated pressure drop using equation xx above is 0.054 psi, which can also be considered negligible.

Once the facility is divided into zones, additional zones are assigned to the outer perimeter of the facility. This allows for the optimization model to consider assigning the air compressor to an area directly outside of the facility wall (See Fig 11). Positioning compressors on the outside of facilities is a common practice in warmer environments where freezing temperature is not an issue.

The Load Factor (L_{ij}) and Volume (V_{ij}) is calculated for each individual machine at each respective zone. To account for the uncertainty of demand in the facility the model randomly shifts the load factor of each machine within a prescribed range. In this paper, air pressure (P_{ij}) requirement is calculated as the maximum pressure required at each zone. For example, if the user defines two machines for Zone 1 and machine 1 has a minimum pressure requirement of 90 psi, and machine 2 has a minimum pressure requirement of 110 psi then the pressure requirement for the respective zone will be 110 psi.

3.1.6 *Simulation-Optimization Method for Identifying Optimal Location*

In this paper, a simulation-optimization technique is used to identify the optimal location that satisfies the optimization model and its constraints. Simulation-optimization provides the capability to determine the optimal zones in the facility by evaluating all possible facility designs and determining the optimal solution for the specific set of constraints and parameters [43-46]. Simulation-optimization is used because it can effectively find the optimal or near optimal solution with uncertainties present. The proposed model has uncertainty in machine load factor. This is done to accounts for variations in machine loads and is discussed further in section 3.3. 1,500 iterations of the facility were simulated to test the effect of variation on the optimal air compressor location. The model also conducts a sensitivity analysis on all possible weights of the user preference, which can be done directly using the simulation-optimization technique.

Moreover in simulation-optimization the change in complexity of the system does not significantly affect the performance of the model, and the objective function and constraints can be changed from one run to the next for all alternative designs in the system [47]. It also used since it offers a more visual and comprehensive approach to finding the optimal solution. This allows for the model to be generalized further and be used by practitioners with limited knowledge of the mathematical formulations embedded in the model. All of this allows for increased complexity and flexibility in future model designs.

The proposed model evaluates all iterations of the system before finding the optimal solution. By defining a finite number of zones and defining their boundaries and characteristics, the performance output for all zones in the facility model is calculated for each iteration. Minimizing the performance output will result in the optimal location with high probability that satisfies the objective function. Figure 5 illustrates a graphical representation of the proposed simulation-optimization approach. As it can be seen, the model will move from zone to zone calculating a performance output for each location of the air compressor. All zones of the facility are evaluated.

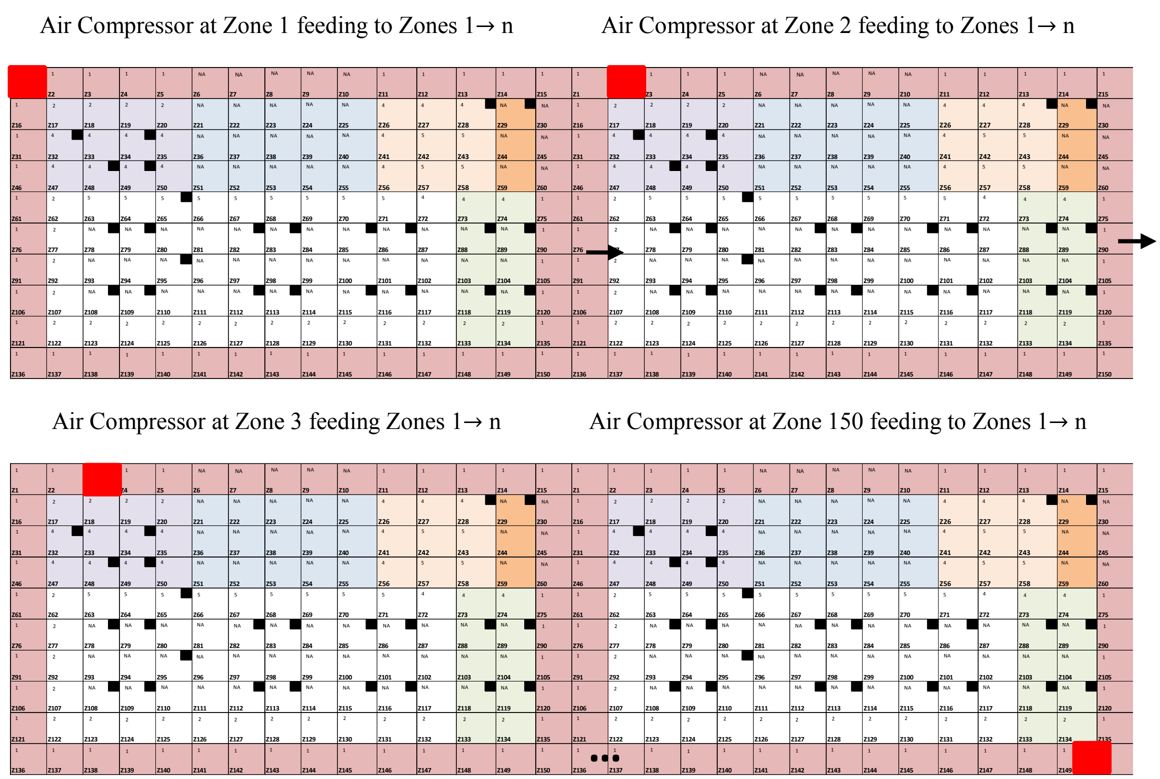

Fig. 10. Graphical Representation of Simulation-Optimization Approach.

For the modeled facility the proposed simulation model will run n number of times, where n is the number of total zones in the facility. This will calculate the compressor capacity needed to supply air demand at all zones in the facility. This is mathematically represented in equation 22 below.

$$.\hat{z} = \min\left[\left([\alpha I_1 + 1] \times \sum_{i=1}^{N} \sum_{j=1}^{R_1} \left[\left(P_{i,j} * \dot{V}_{i,j} * l_{i,j}\right) * D_{i,1}\right]\right) + B_1\right]$$

$$+B_2 + \left[\left([\alpha I_2 + 1] \times \sum_{i=3}^{N} \sum_{j=1}^{R_1} \left[\left(P_{i,j} * \dot{V}_{i,j} * l_{i,j}\right) * D_{i,2}\right]\right) + B_2\right] + \cdots \left[\left([\alpha I_n + 1] \times \sum_{i=1}^{N} \sum_{j=1}^{R_1} \left[\left(P_{i,j} * \dot{V}_{i,j} * l_{i,j}\right) * D_{i,n}\right]\right) + B_n\right] \quad (22)$$

Once the parameters and attributes of each zone have been defined, a simulation-driven analysis is conducted for the proposed optimization model.

3.2 Results of Proposed Simulation-Optimization Model (First Model)

A simulation driven analysis is conducted to investigate the feasibility of using the proposed objective function for finding the optimal air compressor location in a facility. The experimental analysis is conducted in three phases: The facility model is first developed, zone characteristics are defined, and a simulation analysis is then conducted to find the optimal air compressor location.

3.2.1 Phase 1: Facility Model Development

The facility size and layout are defined and incorporated in the model, and the facility is divided into zones. Special attention is taken to generate a simulation environment that realistically represents the characteristics of a typical manufacturing facility in the United States. The facility model is developed based on the authors’ experience visiting over seventy manufacturing facilities in the United States. A rectangular shaped facility layout is selected for this simulation analysis (See Fig. 11.).

The considered facility has a length of 260 feet and a width of 160 feet resulting in a 41,600 sq/ft area (Fig. 11.). The figure legend shows the seven main areas in the facility. These are: Outside Space (perimeter of facility wall), Office Space, Production Area, Machine Shop, Paint and Sandblasting Area, Shipping and Receiving, and Warehouse Space. As discussed in section 2.2.1, a square zone size of 20ft x 20ft was selected as it is representative of an ideal compressed air demand area in a facility. This zone size resulted in the facility being divided into 150 zones. 104 zones represent the inner facility area, while 46 zones represent the perimeter of the facility. Once the facility layout and zones are established, the characteristics of each zone are defined.

3.2.2 *Phase 2: Zone Characteristics Identification*

For this facility model, fifty-nine machines were defined (Table 3) at twenty-four zones in the facility (Fig. 11). Fig. 11 displays the demand zones for the established facility and the associated user preference for each zone in the facility. The user preference levels are assigned using preferential locations as suggested by twenty-two experienced facility managers. The value of I_z, varies across individual users. As it can be see, the facility is divided into 150 20ft x 20ft zones. NA represents zones that are not feasible for air compressor location (Fig.11).

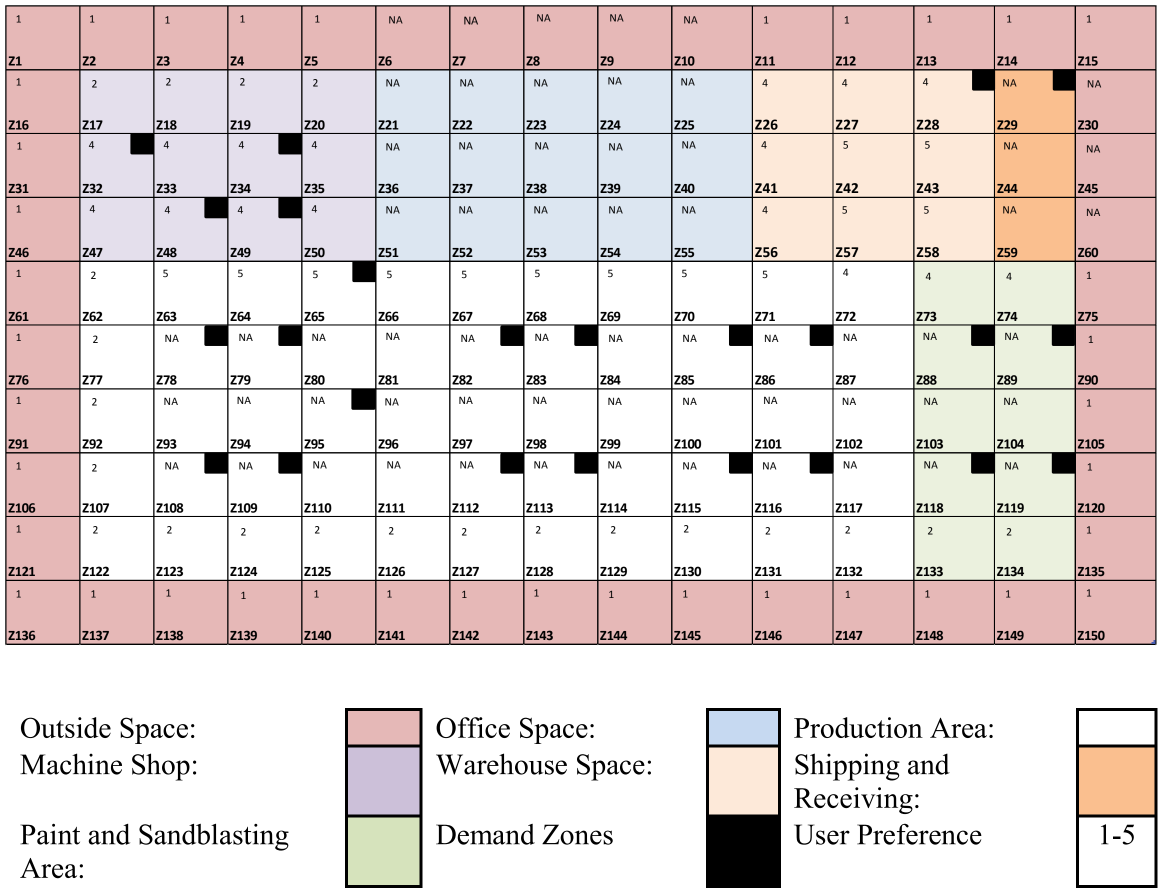

Fig. 11. Modeled Facility, Identified Demand Zones and User Preference per Zone

The air temperatures for all zones in the facility satisfy the temperature constraint in the objective function. This assumption is based on two important factors: 1) The facility location was set in Miami, Florida where the outside temperature always falls within the appropriate temperature range for air compressor operation [48], and 2) The outside air temperature will be used as the base temperature throughout the interior of the facility since it is assumed that the air compressor will have an outside air intake at any selected zone.

Machines are picked to correspond with appropriate areas in the facility. For example, the machine shop area has pneumatic tools (sanders, grinders, screwdrivers) that are usually found in a typical machine shop in a manufacturing facility. Furthermore, the air volume, pressure, and load factor of each tool is also representative of actual tool settings and usage (See Table 3) [49]. It is important to note that the model varies the load factors shown in Table 1 to account for the uncertainty of changing demand.

Table 3

Displays pressure (psi), load factor, and volume (cfm) for each individual machine, at each zone in the modeled facility.

Zone	Machine	PSI	CFM	LF(%)
Z28	Nailer #1	80	6	15%
	Nailer #2	80	6	15%
Z29	Stapler #1	80	5	15%
	Stapler #2	80	5	15%
Z32	Drill #1	70	60	20%
	Drill #2	85	50	20%
	Dusting Blow Gun	80	5	30%
Z34	Dusting Blow Gun	80	5	30%
	Grinder	75	20	30%
	Vacuum Cleaner Shop	100	6	40%
Z48	CNC	80	8	20%
	Dusting Blow Gun	80	5	30%
Z49	Riveters	90	12	15%
	Screw Drivers	90	5	15%
Z65	Drill	80	60	20%
	Dusting Blow Gun	80	5	30%
Z78	Impact Wrench	80	10	30%
	Screw Drivers	90	5	15%
	Air Hoist	80	5	40%
Z79	Impact Wrench	80	10	30%
	Screw Drivers	90	5	15%
Z82	Grease Gun	70	4	20%
	Spring oiler	50	4	20%

Zone	Machine	PSI	CFM	LF(%)
Z88	Sandblaster	90	200	10%
Z89	Air Hoist	80	5	20%
	Paint Sprayer	50	20	40%
	Touch Up paint Gun	90	4	15%
	Undercoat paint Gun	90	19	40%
Z95	Drill	80	60	20%
	Dusting Blow Gun	80	5	30%
Z108	Impact Wrench	80	10	30%
	Screw Drivers	90	5	15%
	Air Hoist	80	5	40%
Z109	Impact Wrench	80	10	30%
	Screw Drivers	90	5	15%
Z112	Grease Gun	70	4	20%
	Spring oiler	50	4	20%
Z113	Air Hoist	80	5	40%
	Dusting Blow Gun	80	5	30%
Z115	Drill	80	60	20%
	Impact Wrench	80	10	30%
	Screw Drivers	90	5	15%
	Vacuum Cleaner Shop	100	6	40%
Z116	Spring Oiler	50	4	20%
	Dusting Blow Gun	80	5	30%

Z83	Air Hoist	80	5	40%		Screw Drivers	90	5	15%
	Dusting Blow Gun	80	5	30%	Z118	Sandblaster	90	200	10%
Z85	Drill	80	60	20%	Z119	Air Hoist	80	5	20%
	Impact Wrench	80	10	30%		Paint Sprayer	50	20	40%
	Screw Drivers	90	5	15%		Touch Up paint Gun	90	4	15%
	Vacuum Cleaner Shop	100	6	40%		Undercoat paint Gun	90	19	40%
Z86	Spring Oiler	50	4	20%					
	Dusting Blow Gun	80	5	30%					

Once the facility model is established, an air demand profile is generated for the facility (Fig.7). The air demand is dependent on the Volume Vij and Load Factor lij of each machine present at each zone i. The resulting range in total facility cfm demand across all iterations was 203.8cfm to 225.2 cfm. A sample of the air demand profile in the facility is shown in Fig. 7.

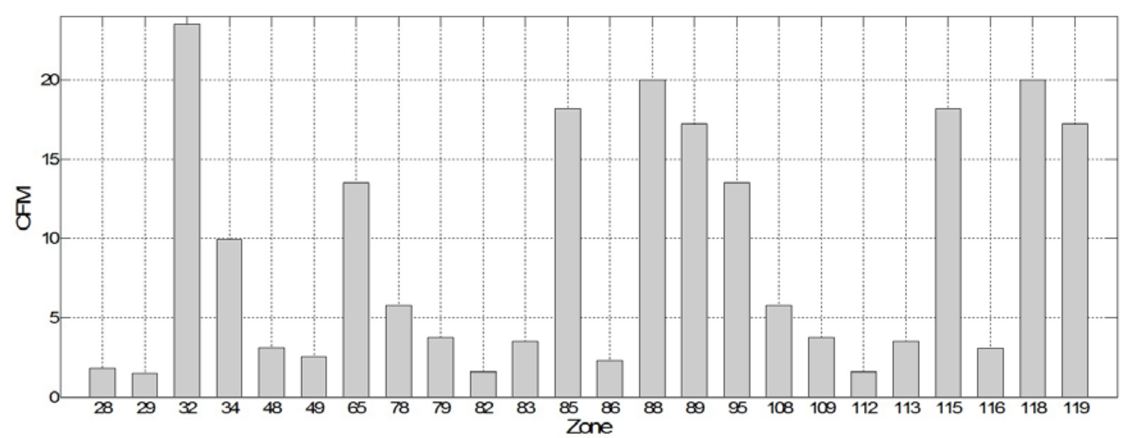

Fig. 12. Air Demand (cfm) at Twenty-Four Zones in Modeled Facility.

3.2.3 Phase 3: Simulation Analysis

A simulation-driven analysis is conducted to identify the optimal air compressor location for the modeled facility. The analysis is developed and implemented in MATLAB due to its effectiveness at solving deterministic simulations. Other software such as LabVIEW, and ARENA Simulation Software can also be used.

The simulation analysis is performed for the following set-ups of the model:

1) To find the most energy efficient air compressor location with no articulation of location preference by the user or consideration of zone feasibility (Energy Baseline).
2) To find the most optimal compressor location by conducting a sensitivity analysis on all weights ($\alpha \in [0, 1]$) of user preference in the function and with consideration of zone feasibility.

1,500 iterations of the facility model are simulated. For each iteration, the model randomly varies the suggested load factor of each machine by ±20%. This allows for the model to accurately incorporate uncertainty of load variations in the facility. With 95.2% probability of correct selection, Zone 85 (Z85) is the optimal location with no articulation of user preference. Furthermore, Zone 83 (Z83) in 4.8% of the runs was identified as the alternative optimal location. An additional 500 iterations were run to assess the difference in savings of Z85 in comparison to Z83, which was found to be 0.065% on average. The range of savings at zone will be discussed later in this section.

Zone 85 is the optimal location with no articulation of user preference. Because the proposed model also considers user preference, the energy optimal location (Zone 85) is not always the best location for the air compressor. By incorporating user preference into the model, the identified optimal location can shift depending on the user preference level at each zone. In this case, Zone 85 is not feasible for an air compressor location since it falls on a production line. Considering user preference will result in a shift of the optimal location which is further discussed below.

To understand the relationship between user preference and energy consumption, the authors conduct a sensitivity analysis on the model by assigning different weights to the user preference in order to investigate the effect it has on the resulting solution. Several values of α from 0-1 were tested. More specifically each simulated iteration of the facility was run 100 times for 0.01 intervals and the resulting output was noted. Figure 10, 11, and 12 show samples of three tested α outputs. The sensitivity analysis resulted in three distinct optimal zones with high probability. For a high energy priority, Zone 70 (Z70), with a probability of 97.3%, is found optimal for $\alpha \in [0, 0.05]$. For a balance between energy and user preference, Zone 130 (Z130), with a probability of 96.7%, is found optimal for $\alpha \in (0.05, 0.13]$. For a high user preference priority Zone 145 (Z145), with a probability of 99.1%, is found optimal for $\alpha \in (0.13, 1]$. Fig. 8 & 9 display the locations of these zones in the facility (See Table 4).

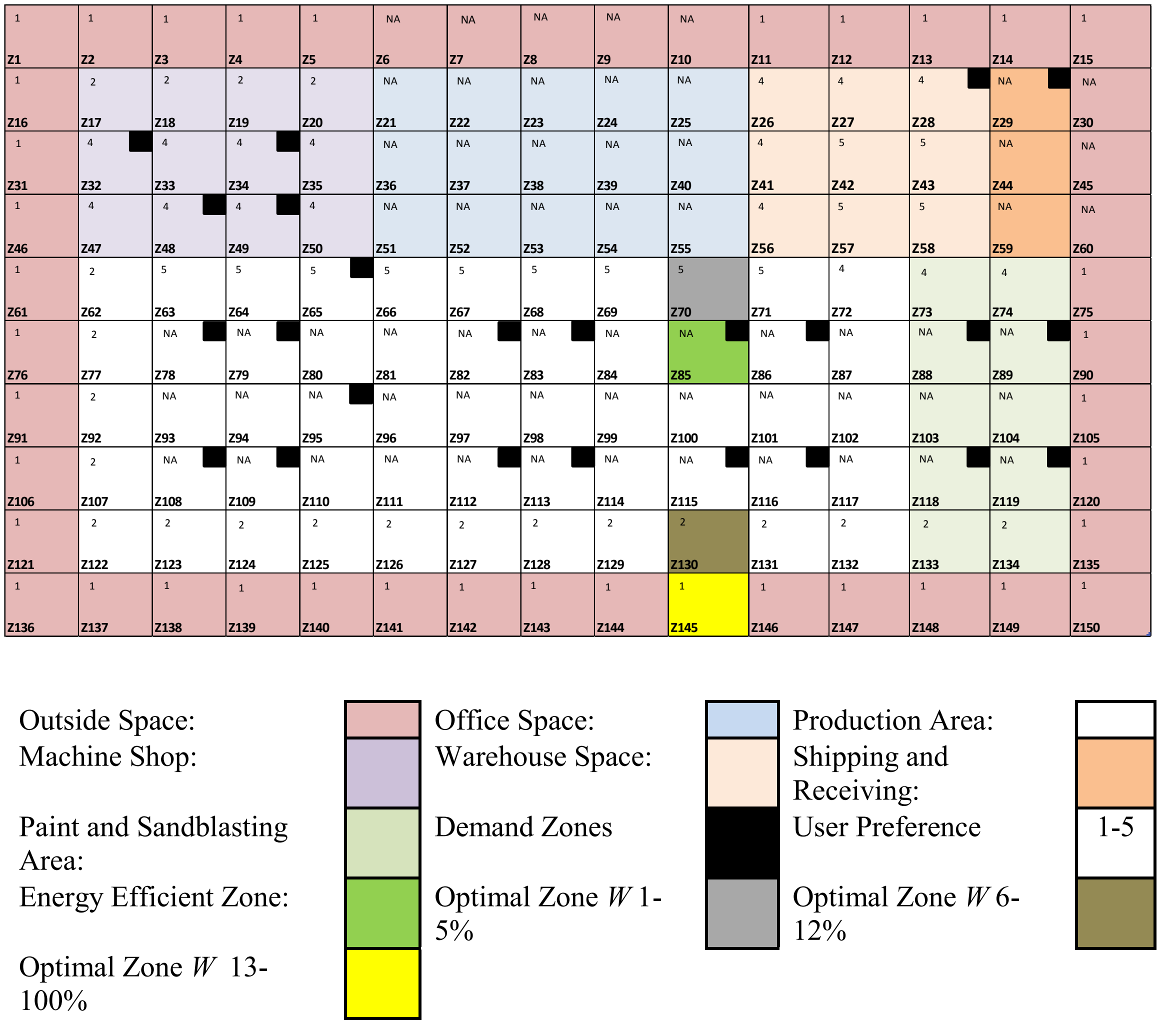

Fig. 13. Facility Layout with Energy Efficient (Z85) and Identified Optimal Locations (Z70, Z130, and Z145)

A 3-D representation of the energy/performance output that is required by the air compressor to meet air demand at each zone in the facility is presented in Fig. 14. Since the model is set to minimize the utility function, the lowest calculated energy output results in the most energy efficient zone. It can also be seen that by increasing the weight of α in the model, the optimal zone shifts to a zone with a higher, more acceptable user preference index.

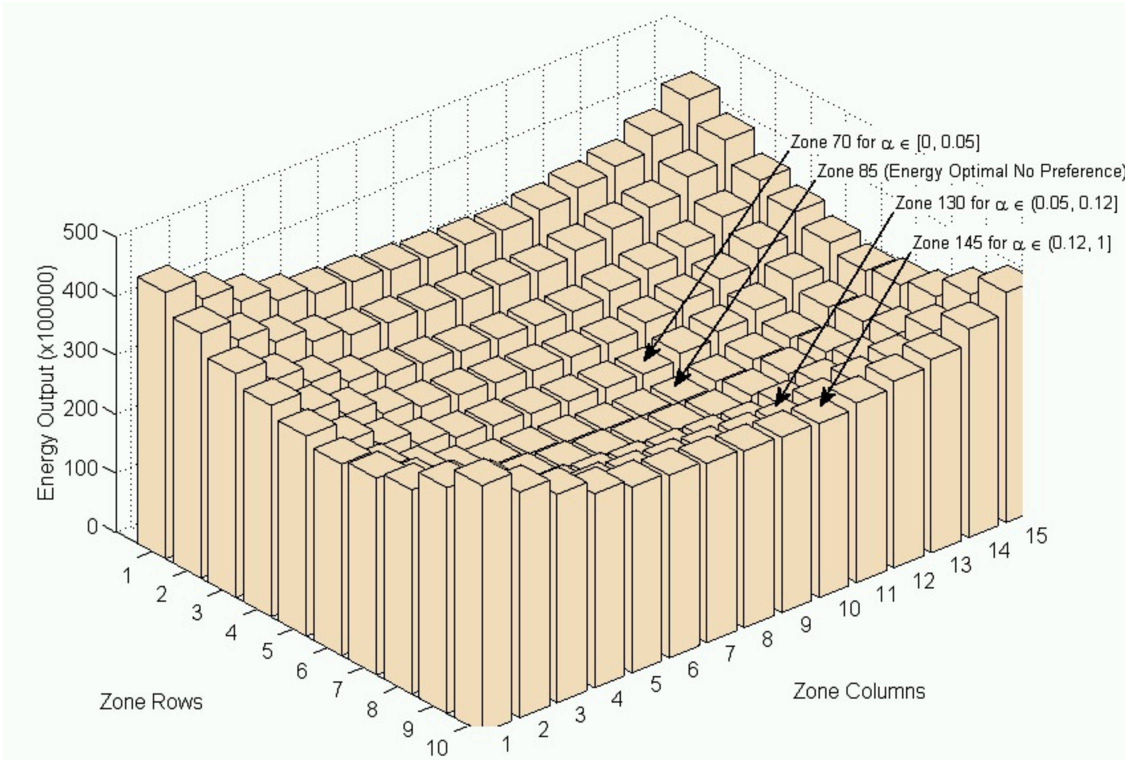

Fig. 14. 3-D Representation of Energy Output at Each Zone

Additionally Fig. 14 displays that the further away the air compressor from Z85, the larger the required energy/performance output.

Since the model shifts the load factor of each machine in the facility, the demand for compressed air will vary for each iteration. The resulting range in total facility cfm demand across all iterations was 203.8 cfm to 225.2 cfm. This will in turn cause variation in the potential energy savings. The range of possible energy savings resulting by placing the air compressor at Zone 85 is 8.59% to 13.94% with an average energy savings of 10.97%. Zone 70, the optimal zone for α∈ [0, 0.05] has a possible energy savings range of 8.03% to 13.06% with an average savings of up to 10.31%. Zone 130, the optimal zone for α∈ (0.05, 0.13]) resulted in a possible energy savings range of 6.27% to 11.38% with an average savings of up to 8.34% and finally, Zone 145, the optimal zone for α∈ (0.13, 0.1]) resulted in a possible energy savings range of 4.91% to 9.64% with an average savings of up to 6.28% (See Table 4).

Table 4 displays the results of the simulation-driven analysis.

Table 4

Results of simulation-driven analysis. Table 4 displays the probability of correct selection of the optimal zones, the range of savings and average savings for all iterations of the model. Note that Zone 85 and 83 are optimal with no articulation of user preference.

	Optimal				**Alternative**				
	Zone	Probability (%)	Range (%)	Average (%)	Zone	Probability (%)	Range (%)	Average (%)	Difference in Avg. Savings (%)
Energy Optimal No Preference	85	95.2	8.59-13.94	10.97	83	4.8	8.59-13.25	10.905	0.065
Optimal for $\alpha \in [0, 0.05]$	70	97.3	8.03-13.06	10.31	71	2.7	7.96-12.65	10.43	0.12
Optimal for $\alpha \in (0.05, 0.13]$	130	96.7	6.27-11.38	8.34	129	3.3	6.03-11.15	8.39	0.05
Optimal for $\alpha \in (0.13, 1]$	145	99.1	4.91-9.64	6.28	146	0.9	4.89-9.47	6.41	0.13

As shown in section 3.3, as the user preference priority increases the potential energy savings will decrease. In fact there is a 4.62% reduction in potential savings between Z145 and Z85; however, Z145 can still potentially save an average of 6.2% even with the highest user preference priority.

Fig.15 illustrates the potential energy savings at each optimal zone for each iteration of the facility.

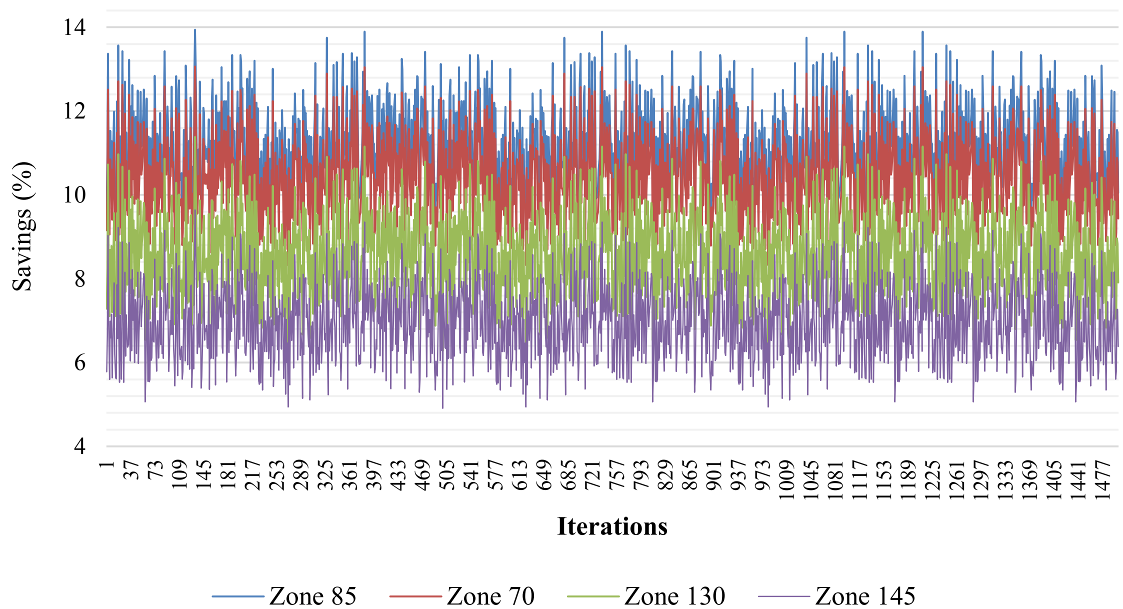

Fig. 15. Resulting Percent Savings for Each Model Iteration

As shown in Fig. 15, Zone 85 results in the highest potential savings. As the weight of user preference increases the optimal location shifts to Zone 70, Zone 130, and Zone 145 respectively. This shift reduces the potential energy savings achievable by the model while increasing the user's preference for location. Even with a high user preference the model can still achieve significant savings.

To understand the effect of the user's preference α on the optimization model, the energy baseline($\alpha = 0$) is plotted against samples of the energy/performance output for α = 0.05, 0.12, and 0.2. The selected values of α fall within the range of that identifies optimality at zone Z70, Z130, and Z145 respectively.

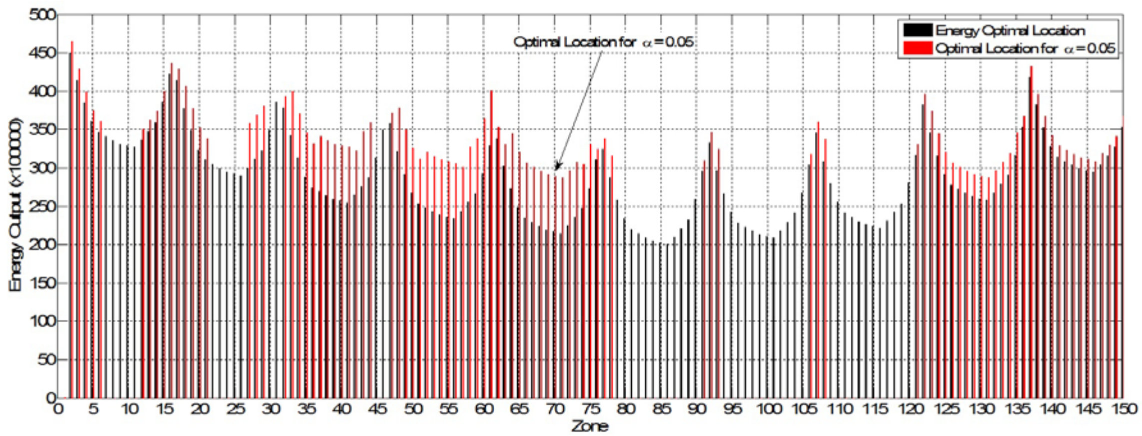

Fig. 16. Baseline Energy vs. Energy with User Preference α = 0.05.

As it can be seen in Fig. 16, a selected weight of α = 0.05 results in the model reaching optimality at Z70 (Fig. 16). Zones assigned *NA* (not feasible) are not considered in the analysis, hence the missing red bars in the chart. Z70 has a user preference level of *5* in the model indicating a zone with a low user preference index.

Next the model is run for α = 0.12 (Fig. 17). Running the model for α = 0.12, the optimal zone shifts from Z70 to Z130 (Fig. 17). The shift results in an optimal zone with a user preference level index of 2. This indicates that the zone is more favorable to the user. Z130 balances both the energy consumption and the user's preference

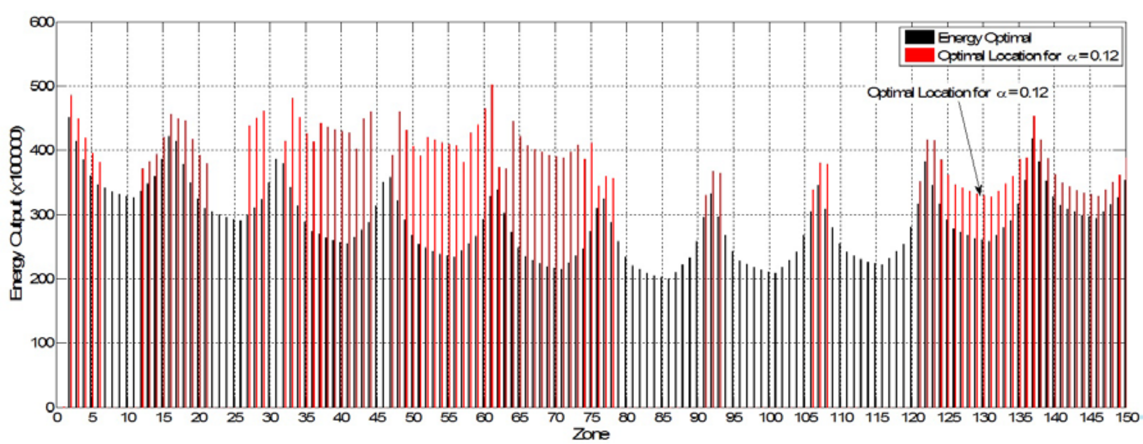

Fig. 17. Baseline Energy vs. Energy with User Preference α = 0.12.

Last the model is run for α = 0.2 (Fig. 18). Plotting the effect of α = 0.2, results in the optimal zone shifting from Z130 to Z145 (Fig. 17). Z145 has a user preference index of *1*.For this particular model, $\alpha \geq$ 0.13 will always result in Z145 as the optimal zone. Z145 is on the outside perimeter of the facility, and is most favorable to the user and has a higher energy/performance output than Zone 85, 70, and 130.

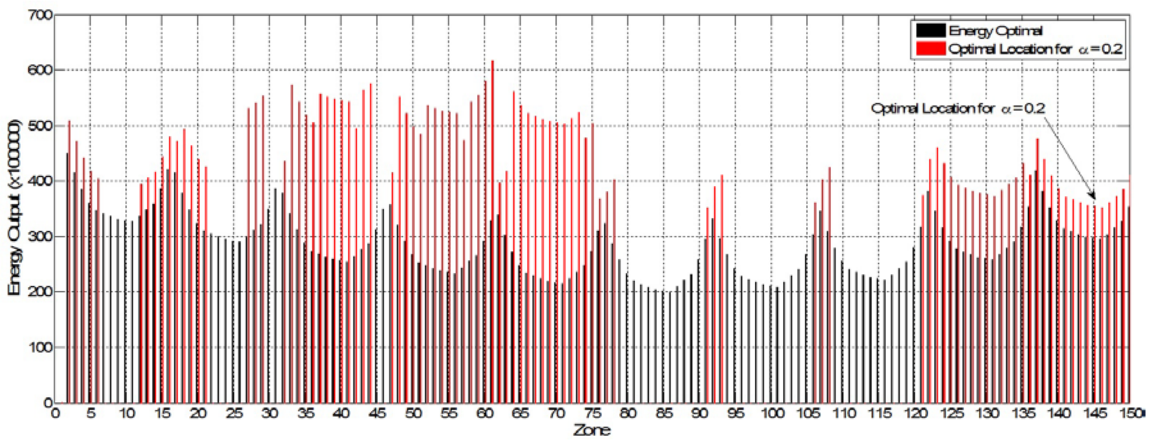

Fig. 18. Baseline Energy vs. Energy with User Preference α = 0.2.

Running a sensitivity analysis shows effect of increasing α, on the resulting optimal locations (Fig. 19). In this case, only feasible zones (Zones that have a user preference index from 1-5) are plotted in ascending order of energy/performance output.

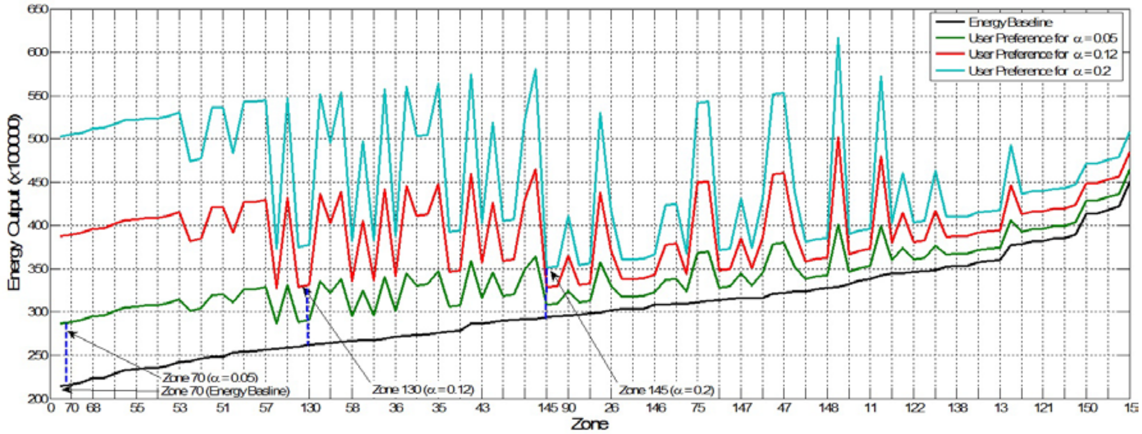

Fig. 19. Energy Baseline vs. Energy with User Preference α = 0.05, 0.12, and 0.2.

As α increases, so does the air compressor energy/performance output at the resulting optimal zone. The energy cost vs. user preference trade-off is clearly displayed by the resulting optimal zone shifting to the right as the weight (αE) of the user preference increases.

3.2.4 Results of First Model

The effect of the air compressor location on energy consumption is displayed by the resulting energy consumption (kW) required to meet demand at each of the eight selected locations (See Table 5).

An energy savings of up to 2.98 kW or 10.8% can be achieved by placing the air compressor in the most energy efficient location Z85. Z70, Z130, and Z145 can save up to 10.11%, 8.1%, and 6.2% in energy consumption respectively. As expected, there is an increase in air compressor energy consumption as the weight of the user's preference increases in the objective function. The increase in energy consumption between Z85 (Energy Baseline) and Z70, Z130, and Z145 is 0.69%, 2.7%, and 4.6% respectively.

Table 5

Summary of Validation Results for the Eight Selected Zones. Table 5 displays the resulting air leaks, pressure drop (psi and %), horsepower (theoretical and effective), and energy consumption (kW) at each zone. Horsepower (hp) is converted to kilowatt (kW) using the conversion ratio of 1hp to 0.746 kW.

	Zone 1	**Zone 70**	**Zone 76**	**Zone 85**	**Zone 96**	**Zone 104**	**Zone 130**	**Zone 145**
Leak (cfm)	16.70	8.3	11.65	7.60	7.78	10.80	9.77	11.2
Pressure Drop (psi)	28.44	13.16	19.02	11.99	12.19	17.57	15.74	18.3
Pressure Drop (%)	14	6.58	10	6	6	9	8	9.14
Theoretical Horsepower	41.70	40.2	40.79	40.06	40.09	40.64	40.45	40.71
Effective Horsepower	47.63	42.8	44.67	42.46	42.54	44.21	43.63	44.43
Energy Consumption (kW)	35.53	31.9	33.32	31.68	31.73	32.98	32.55	33.14

These results suggest that a significant energy and cost savings can be attained by optimizing the air compressor location using the proposed objective function (1). The results also suggest that incorporating user preference can also lead to significant energy savings while maintaining a desired air compressor location. There is a noticeable trade-off between energy consumption and user preference. As the user's preference weight α increases in the function, so does the energy consumption of the newly identified optimal zone.

As the air demand at the facility increases, so will the potential kW and percent savings. It is also important to understand that in order to sustain energy savings in a compressed air system, there must be a continuous improvement application as further discussed in ISO50001 [26].

CHAPTER 4: SIMULATION-OPTIMIZATION MODEL (NEW MODEL)

4.1 Mathematical Model for Simulation-Optimization (New Model):

After further research and more in-depth analysis, a new and potentially more accurate model is developed. This mathematical model is based on a number of equations and uses the direct equations resulting in the various savings opportunities to identify the optimal location. The new model will be able to accurately calculate the pressure drop, air leaks and kW required to supply compressed air to all the demand locations in the facility.

To find the optimal air compressor location in a manufacturing facility it is important to first identify and prioritize the determinants that affect the selection of location and capacity of the air compressor. Energy Consumption (compressor power and system losses) and user location preference are identified as the two main contributing factors to determining the air compressor location and capacity. A mathematical model is developed to determine the optimal air compressor location in a facility that minimizes total energy consumption while considering the user's location preference to maximize user satisfaction (See Section 2.3). The following sections discuss the development and derivation of the energy consumption term, distance term, system loss terms, and user preference term, resulting in the proposed model.

4.1.1 Distance Term in Mathematical Model

As a first step a rectilinear zone-to-zone distance matrix must be established in order to determine the distance the compressed air must travel to each demand zone i. Rectilinear distances between the compressor and demand zones are used as they are the more compatible to a compressed air piping layout. Rectilinear distance between each zone can be calculated using Equation 1 (Chalmet et al., 1981; Hanan, 1966).

$$d_{iz} = |X_i - X_z| + |Y_i - Y_z| \tag{1}$$

where d_{iz} is the distance between zone i, and the potential air compressor zone z. (X_i, Y_i) is the location coordinate of machines at Zone i whereas, (X_z, Y_z) is the location coordinate of the air compressor at Zone z.

Clearly, minimizing total distance to air demand locations will lead to a significant reduction in pressure drops and air leaks. This results in less air losses in the system hence improving the air compressor cfm per kilowatt output (cfm/kW) since the power of the air compressor depends on the air leaks, pressure drop and distances to the demand zones. Hence, the mathematical model must incorporate both pressure drops and air leak losses when calculating the required capacity (kW).

4.1.2 Derivation of the Power Term in the Mathematical Model

For the proposed model, a novel algorithm is developed that captures the necessary power to supply compressed air to all demand locations in part using a combination of standardized fluid mechanics and thermodynamics equations. The model calculates the optimal location by evaluating the *true* compressed air power required to supply air demand in a facility. The mathematical framework that links the air losses to compressor power requirements is presented in this subsection.

4.1.3 Theoretical Kilowatt Derivation

To calculate the required compressor kilowatt at each zone, an equation for calculating work generated by the air compressor is derived. The work generated by the air compressor can be calculated using the suction pressure, discharge pressure, and the exponent of the compression curve as given in Fig. 1. The curve in Fig. 1 is representative of any type of air compressor in which clearance is not considered. In the figure, line $a - d$ represents the absolute initial atmospheric pressure (P_1) in pounds per square inch (psi). $a - b$ represents the compression of air from P_1 to P_2 which is the absolute final pressure after compression in psi. This follows the compression curve $\mathrm{P_1V_1^k = P_2V_2^k}$, where k represents the expansion coefficient and V_1 and V_2 are the volume of air before and after compression. Lastly $b - c$ represents the compressor discharge pressure P_2 in psi.

Work (W) is represented by the area enclosed by lines a, b, c, d, a. Volume (V) is in cubic feet, and pressure (P) is multiplied by the constant 144 to give all pressures in pounds per square foot (psf)(Thorkelson, 1913).

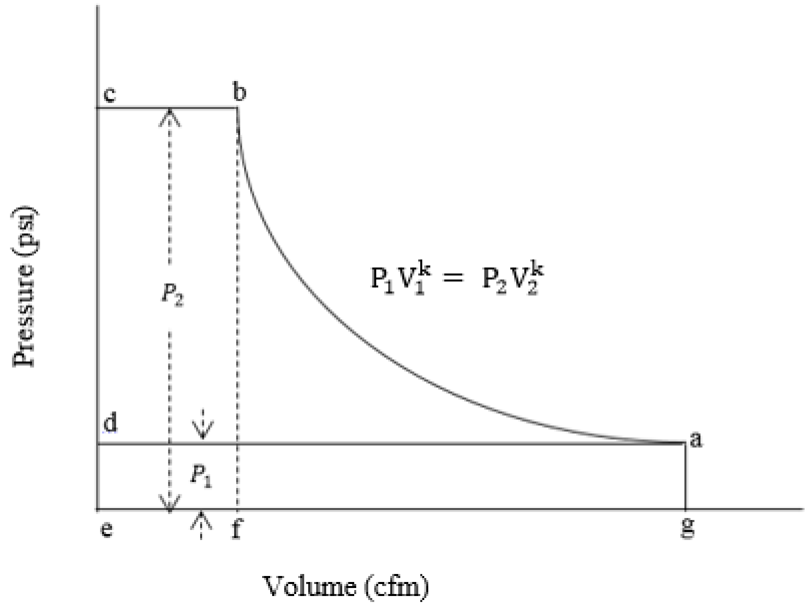

Fig. 20. Air Compressor Work Diagram (Thorkelson, 1913)

To calculate the work needed to compress air from a, d to b, c the required work area can be calculated using Fig. 20. This area is a, b, f, g plus b, c, e, f minus a, d, e, g, which leads to the following equation:

$$W = \frac{144(P_1V_1 - P_2V_2)}{k-1} + 144P_1V_1 - 144P_2V_2 = \left(\frac{k}{k-1}\right) \times 144(P_1V_1 - P_2V_2) \tag{2}$$

The expression in equation (2) can therefore be written as follows:

$$W_{ft.lb} = \left(\frac{k}{k-1}\right) \times 144(P_1V_1)\left[\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} - 1\right] \tag{3}$$

Adding a constant N for the number of strokes per minute and converting the resulting foot pound (ft.lb) to horsepower (hp), by dividing with a constant of 33,000 gives the following equation (Box, 2014; McCulloch, 2003):

$$W_{hp} = \left[\left[\frac{144* P_1*V_1*k}{33{,}000*(k-1)}\right] \times \left[\left(\frac{P_2}{P_1}\right)^{\frac{(k-1)}{N*k}} - 1\right]\right] \tag{4}$$

The resulting horsepower in equation 4 is then converted to kilowatt by multiplying with the constant 0.746

$$W_{kW} = 0.746 * \left[\left[\frac{144* P_1*V_1*k}{33{,}000*(k-1)}\right] \times \left[\left(\frac{P_2}{P_1}\right)^{\frac{(k-1)}{N*k}} - 1\right]\right] \tag{5}$$

where W_{kW} is the work done by the air compressor in kilowatt.

Next the equations to calculate losses in the system are derived. As discussed above, the two main losses in the compressed air system are the pressure drop and the air leaks. Both these losses have a significant impact on the air compressor's energy performance. As the distance to air demand zones increases, so does the potential for the air leaks and the pressure drop in the system.

4.1.4 Pressure Drop in System

Pressure drop accounts for part of the air loss in the system. The pressure drop equation is derived in part from the general equation for calculating adiabatic compressible flow as follows:

$$Q^2 = \left[\frac{A^2}{V_1\left(\frac{f \times d_{iz}}{D_{pipe}} + 2\log_e\frac{P_3}{P_4}\right)}\right] \times \left[\frac{(P_3)^2 - (P_4)^2}{P_3}\right] \tag{6}$$

where P_3 is the pressure at the beginning of the pipe line, P_4 is the pressure at the end of the pipe line, Q is the mass flow rate, f is the friction, d_{iz} is the length of pipe, D_{pipe} is the diameter of the pipe, and A is the cross sectional area of the pipe.

Since using the general flow equation is tedious, a simplified version of the equation is introduced to calculate pressure drops in a pipe. This equation is used by both CAGI and the Compressed Air and Gas Handbook to generate pressure drop tables (McCulloch, 2003; Peurifoy, Ledbetter, & Schexnayder, 1970; Scales, McCulloch, & Challenge, 2009):

$$\mathrm{P}_3^2 - \mathrm{P}_4^2 = \left[\frac{\mathrm{Q}^2 \times d_{iz} \times \mathrm{f}}{\mathrm{r} \times {D_{pipe}}^5}\right] \tag{7}$$

where r is the ratio of compression based on the absolute pressure.

Incorporating the Moody friction factor for steel f_{steel}, the equation for pressure drop becomes (McCulloch, 2003; Rollins, 1961):

$$\mathrm{f}_{\mathrm{steel}} = \frac{0.1025}{{D_{pipe}}^{0.31}} \tag{8}$$

Hence,

$$\mathrm{P}_3^2 - \mathrm{P}_4^2 = \frac{d_{iz} \times \mathrm{Q}^2}{\mathrm{r} \times \mathrm{d}^5} \times \frac{0.1025}{{D_{pipe}}^{0.31}} = \frac{0.1025 \times d_{iz} \times \mathrm{Q}^2}{\mathrm{r} \times {D_{pipe}}^{5.31}} \tag{9}$$

Equation 9 is used to calculate the pressure drop losses in the model.

4.1.5 Air Leaks in System

Air leaks in the system are the second type of loss accounted for in the proposed model. Air leaks (V_{Leaks}) are calculated using the equation from the Compressed Air and Gas Handbook (McCulloch, 2003) as follows:

$$V_{Leaks} = \frac{\frac{d_{iz}}{d_{leaks}} \times (T_i + 460) \times \frac{P_2}{P_1} \times C_1 \times C_2 \times C_d \times \frac{\pi {D_{size}}^2}{4}}{C_3 \times \sqrt{T_i + 460}} \tag{10}$$

Recall that P_1 denotes absolute initial atmospheric pressure (psi), and P_2 is the absolute final pressure after compression (psi). C_1 represents the isentropic sonic volumetric flow constant (28.37 ft/sec-°$R^{0.5}$), C_2 and C_3 are conversion constants (60 sec/min and 144 in^2/ft^2 respectively), and C_d denotes the coefficient of discharge for a square edged orifice (0.8). T_i denotes the average line temperature. D_{size} is the size of each air leak in the model and d_{leaks} is the distance between air leaks in the system and is measured in feet. Both the size of air leaks (D_{size}) and the distance between air leaks *(*d_{leaks}) fluctuate in the model and as such are treated as uncertain parameters.

4.1.6 Actual Power of Air Compressor

Incorporating equations (9) and (10) into equation (5) will result in the actual work term (β) that considers both pressure drop and air leak losses in the system. As mentioned earlier, the rectilinear distances in the facility are used in part to calculate the pressure drop and number of air leaks in the system. By using the rectilinear distance (d_{iz}) in the work term, the model is able to adjust the losses in the system for distance, while calculating the power required to supply compressed air to all demand zones. The actual power equation measure in kW is then:

$$\beta_{kW} = 0.746 \left[\frac{144 \times P_1 \times \left[l_{ij} \left[V_{ij} + \frac{\frac{d_{iz}}{D_{Leaks}} * (T_i + 460) \times \frac{P_2}{P_1} \times C_1 \times C_2 \times C_d \times \frac{\pi D_{size}^2}{4}}{C_3 \times \sqrt{T_i + 460}} \right] \right] \times k}{33{,}000 \times (k-1)} \right] \times \left[\left(\frac{P_2 + \frac{0.1025 \times d_{iz} \times V_{ij}^2}{\left(\frac{P_2 + P_1}{P_1}\right) D_{pipe}^{5.31}}}{P_1} \right)^{\frac{(k-1)}{N \times k}} - 1 \right] \tag{11}$$

4.2 User Preference Variable in the Mathematical Model

Identifying the air compressor location that is most energy efficient will result in the lowest compressed air cost to the facility. This however will not always result in the most desired optimal air compressor location for the user. By incorporating the user's preference in the model, the user is allowed to designate preferred zones in the facility. User's preference is an important consideration as air compressors can be very noisy and can occupy a large footprint, which might get in the way of a production process and/or material routes. The User's preference is also important as it allows facilities to consider future expansions and changes that cannot be predicted by the model. By incorporating user preference, the user can prioritize areas of air compressor location. The user preference index adds an additional degree of freedom to effectively manage the trade-off between the energy cost and the user's location preference.

A five level user *preference index* I_z is introduced for each zone. The model uses an *a priori* preference (Kaya et al., 2002) approach to represent the preference level of I_z at each zone. The index has a range from 1 to 5 where 1 is the most desired (least inconvenient) location and 5 is the least desired (most inconvenient) location for the user.

A set of feasible zones (Z) is defined in the model. A zone is considered feasible if it exists within the true facility layout and has a temperature that always falls within the defined temperature range $15°\mathrm{F} \leq \mathrm{Tz} \leq 125°\mathrm{F}$ (Kaya et al., 2002). Room and outside temperature affect the performance of the air compressor and are therefore both considered in the model.

4.3 Proposed Mathematical Model

As indicated earlier, the proposed model has two objectives: (1) minimizing the total energy consumption of the air compressor and hence the system losses and energy cost for the facility, and (2) maximizing the user's preference for the air compressor location in the facility. The two goals are incorporated into a single objective function via a weighted sum of:

1) The product of i) distance between the air compressor location and each defined facility zone, ii) the compressed air total demand (machine demand plus air leaks), and iii) pressure at each respective zone (pressure set point for machines plus the related pressure drops);

2) The level of user preference for air compressor location, while incorporating zone availability with respect to zone temperature within the set of feasible zones (Z).

The resulting mathematical model can be written as follows:

$$\hat{z} = \underset{z\in\{Z\}}{\operatorname{argmin}} \, [w\mathrm{I}_z + 1]$$

$$\sum_{i=1}^{N}\sum_{j=1}^{R_i}\left[0.746\left[\frac{144\times P_1\times[l_{ij}[V_{ij}+V_{Leaks}]]\times k}{33{,}000\times(k-1)}\right]\times\left[\left(\frac{P_{ij}+P_{Drop}}{P_1}\right)^{\frac{(k-1)}{N\times k}}-1\right]\right] \tag{12}$$

where N and R_i represents the number of zones in the facility and the number of machines in each zone respectively. Also note that $\mathrm{P_{Drop}}$ and $\mathrm{V_{Leaks}}$ are defined in (9) and (10) respectively. For machine j in zone i, P_{ij} is the pressure (psig) requirement, V_{ij} is the volume (cfm) requirement, and l_{ij} is the load factor (%). Recall that d_{iz} is the distance between zone i, and the potential air compressor zone z is as defined in (1). The user preference for air compressor location of zone z is represented by a multi-level user preference index, I_z, which is captured by the weight,w in the objective function. w is a linear function of the term $z.w \in [0, 1]$ designates the level of w used. When w 0 is the model will only minimize the energy consumption of the air compressor. When w 1 is the model will minimize both energy consumption and user preference equally. Clearly higher values of w indicate cases where the user preference for air compressor location is more important to the user than the energy cost to the facility. A higher w normally leads to greater air compressor energy consumption for a given facility. Later in the numerical analysis, a sensitivity analysis is carried out to test the effect of a hundred possible weights of the user's preference on the resulting optimal location. The optimal zone for air compressor location is the zone z that minimizes the objective function given in (12).

The mathematical model is graphically represented in Fig. 21 below:

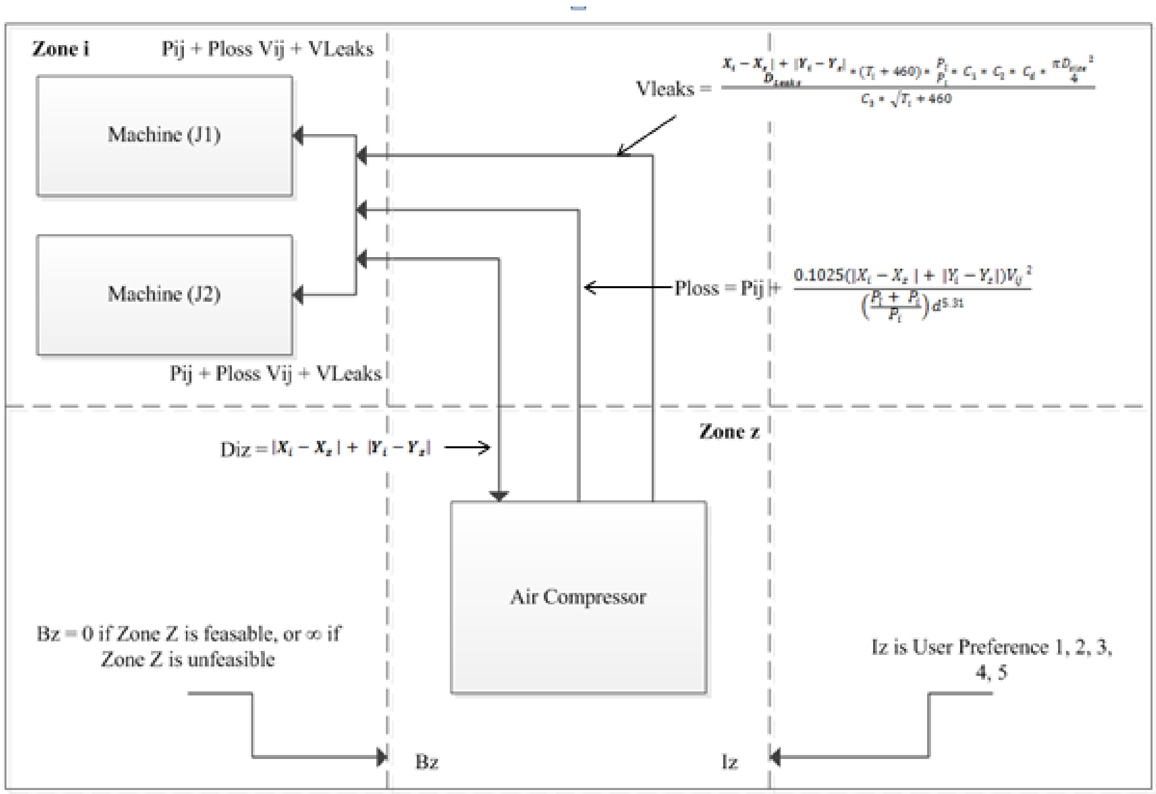

Fig. 21. Air Compressor Set-up and Relationship of Variables in Optimization Model

To follow the figure, recall that the air compressor positioned at zone z feeds air to machine's $J1$ & $J2$ in zone i. V_{Leaks} are the leaks that form as air is delivered from the air compressor in zone z to the machines in zone i. On the other hand, P_{Drop} is the pressure loss resulted due to leaks and friction in the pipe as air is delivered from the air compressor to the machines. The compressor must maintain the designated machine pressure set point. As the pressure drop from zone z to zone i increases, so will the energy cost to the facility.

4.3.1 Variable Parameters of the Mathematical Model

There are four main variables that are treated as uncertain parameters in the proposed model. These are 1) the load factor of the machines in the facility (L_{ij}); 2) the pressure set point required in the facility (P_{ij}); 3) the frequency of air leaks (ft) in the system; and 4) the size of each leak in the facility (D_{size}). These parameters vary depending on the state of the manufacturing process and other factors.

The load factor $L_{ij}$of the machines at each zone is in part determined by data published by Ingersoll Rand (Rand, 2013). The percent load of a machine determines the length of time the machine is on, and directly affects demand (cfm) in the facility. Obviously higher machine load factors result in higher compressed air demand in the facility, making them both directly correlated.

The load of each machine in a facility can be very dynamic and is dependent on various factors such as production schedules, product demand, availability of raw materials, etc. Although Ingersoll Rand suggests loads for various types of machines, there can still be some unexpected load change or uncertainty that can affect air demand in a facility. This will be discussed further in section 3.2.

The overall air compressor pressure set point in a facility is variable and uncertain mainly because of the controls installed on the air compressor. In this research the pressure set-point of the system is kept steady at 100 psi. However the compressor pressure set point is designed to fluctuate within a specific parameter interval preset by the manufacturer to prevent the air compressor from loading and unloading continuously. The model is therefore designed to alter the compressor pressure set-points between certain upper and lower bounds ($95\ psi \leq\ P_{ij} \leq 105\ psi$). This directs the compressor to unload once the pressure hits 105 psi and cycle back on once the pressure hits 95 psi.

Air leaks in a facility are very unpredictable. On average, compressed air systems lose up to 15% to 40% of the air produced to air leaks, which, makes it the greatest loss of energy in compressed air systems at manufacturing facilities (Saidur et al., 2010). Air leaks occur at the joints, flange connections, elbows, reducing bushes, sudden expansions, valve systems, filters, hoses, check valves, relief valves, extensions, worn out piping, and the equipment connected to the compressed-air lines (Saidur et al., 2010). Since the number of machines in the facility does not change, the frequency of air leaks is considered only for the main pipe line and not downstream at the machine level. The model randomly varies the distance of each leak occurrence and the size of each leak in the system. Based on the existing literature and observed data for most common air leak sizes, the model is able to generate a realistic representation of air leaks in the system.

4.3.2 Other Assumptions in the Mathematical Model

In this section the assumptions used in the model are discussed. Some assumptions are already mentioned and discussed further in their respective sections. For simplification of the mathematical model, it is assumed that the compressed air flow in the pipe will only be laminar and not turbulent. This assumption is made since the model is only designed for one compressor in one location and will have minimal turbulence in the pipes. Furthermore the model considers a well-designed piping system that minimizes T-joints and dead ends, which reduces the turbulence in the piping structure significantly.

When calculating pressure drop in a pipe, it is assumed that the behavior of air in a compressed air setting is within the temperature and pressure ranges for ideal gas. Friction is also assumed to be constant throughout the piping system as the piping size will not change.

For kW calculations, it is assumed that for every 2 psi increase in the air pressure set point, the air compressor experiences a 1 % increase in energy consumption (Christina & Worrell, 2008; D'Antonio et al., 2005; Kaeser, 2000). Furthermore since the number of demand locations and number of machines do not change in the model, piping drops are assumed to be uniform for each compressor location and do not affect the pressure drop analysis.

Last, the compressed air system is set-up in the same way at each selected zone. Therefore the performance of the air compressor and the cost of the set-up are not affected by the selected zone.

4.4 Simulation Driven Analysis of New Optimization Model

A simulation-based optimization approach is used to investigate the effectiveness of the proposed model in finding the optimal air compressor location and capacity in a manufacturing facility. The facility model and characteristics are discussed and the simulation analysis and results are presented in the corresponding sections.

Simulation-optimization provides the ability to determine the best zones in the facility by evaluating all possible facility designs and determining the optimal solution for the specific set of constraints and parameters. As discussed in Section 2.5, the model has various uncertainties that help account for the dynamic environment in manufacturing facilities. These uncertainties are best incorporated into the model using simulation optimization. In this paper 1,600 instances are simulated to test the effect of the various facility setups and uncertainties on the identified optimal air compressor location. Each instance is defined as a combination of the cfm demand, pressure set point, and the number and size of leaks in the facility. A sensitivity analysis is also conducted to test the effect of a hundred possible weights of the user's preference on the resulting optimal location.

The proposed model considers all instances of the facility when finding the optimal location. The power (kW) required to supply compressed air is calculated for each zone in the facility. Minimizing the power of the air compressor (kW), while considering the user preferences, results in the optimal location that satisfies the multi-criteria objective function. The following subsection elaborates on the facility model and its characteristics.

4.4.1 Facility Model Development and Zone Characteristics

In order to benchmark the performance of the proposed model, the facility layout and characteristics employed In (Joel Zahlan & Asfour, 2015) are also used in this paper. A quick summary of the facility layout is given in Fig. 22. For a more detailed description of the layout the reader is referred to (Joel Zahlan & Asfour, 2015).

The considered facility has a length of 260 feet and a width of 160 feet resulting in an area of 41,600 sq/ft. A square zone size of 20ft x 20ft is used as it is representative of an ideal compressed air demand area in a facility. Using this zone size results with 150 zones. 104 zones fall the inner facility area while 46 zones encompass the perimeter of the facility.

For the facility model, 59 machines are defined in 24 zones as depicted in Fig. 22. The figure displays the different areas in the facility, the compressed air demand zones, and the associated user preference for each zone in the facility.

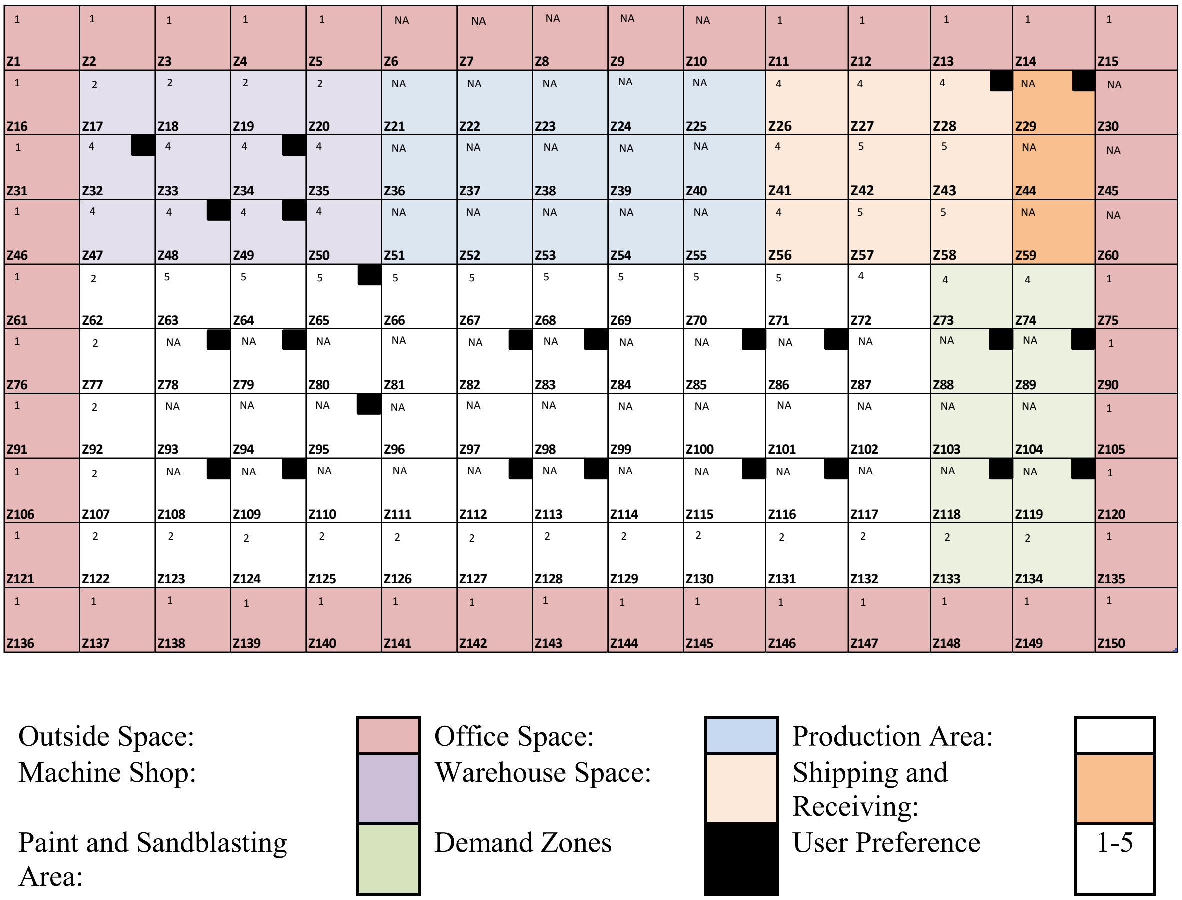

Fig. 22. Modeled facility layout with demand zones and associated user preferences

It is important to note that the facility is designed to be representative of a typical medium sized manufacturing facility in the United States. Machines are picked to correspond with appropriate areas in the facility. Furthermore, the air volume, pressure, and load factor levels of each machine/tool are taken from research provided by (Rand, 2013) and are also representative of actual tool settings and usage in practice.

4.4.2 Simulation Driven Analysis and Results

The simulation-based optimization framework is developed and implemented in VBA in the MS Excel environment due to its ease of use, availability, and effectiveness. The optimization approach is performed to generate the following outputs of the model:

i. Determination of the most energy efficient air compressor location with no articulation of location preference by the user or consideration of zone feasibility (Energy Baseline).
ii. Determination of the most optimal compressor location by conducting a sensitivity analysis by explicitly incorporating all weights ($w \in [0, 1]$) for user preference and zone availabilities.

As mentioned earlier, 1600 instances of the model are generated to account for facility variations and uncertainties defined in Section 2.5. For each instance, the model randomly varies the suggested load factor of each machine by $\pm 20\%$. This allows for the model to incorporate uncertainty of load variations in the facility. The model also varies the pressure set-point in the system so as to realistically account for the control strategies used in industry. Lastly the instances also involve variations in the number of air leaks and size of air leaks present in the facility. The model is simulated to calculate the required kW at each of the 150 zones in the facility. Fig. 23 illustrates, using the output of an instance, the average effect of air leaks and pressure drop losses on the kW consumption of the air compressor at each zone in the facility.

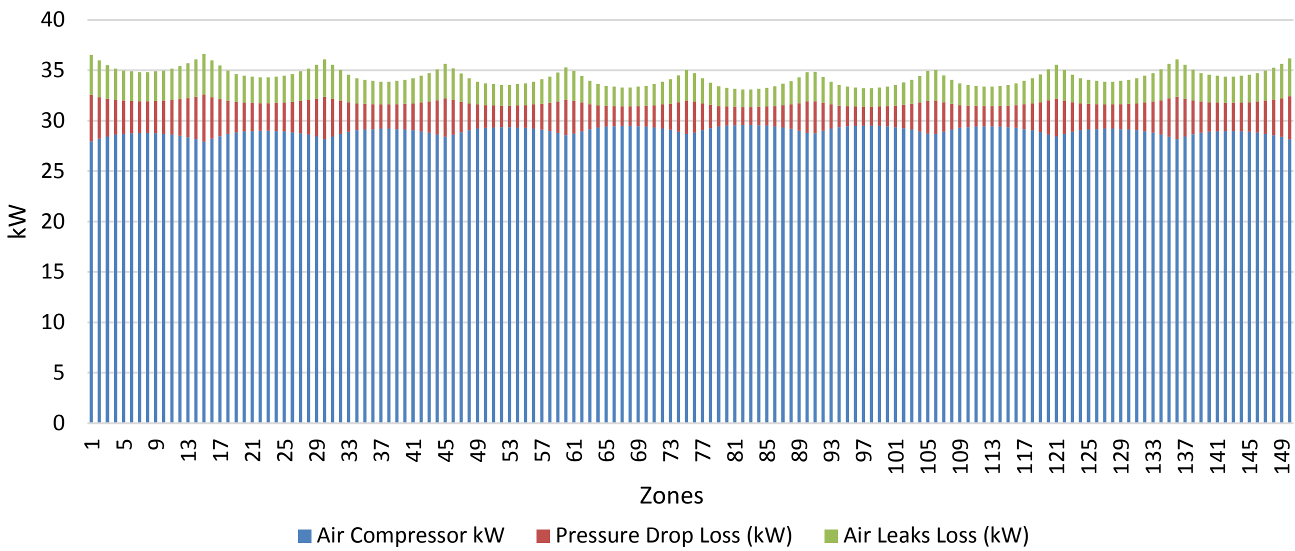

Fig. 23. Average Effect of Air Leak and Pressure Drop on Overall System

For this particular facility model, air leak and pressure drop losses account for an average of18.6% of the total kW required by the air compressor. Air leak and pressure drop losses in typical manufacturing facilities can range from 15% to 40%, indicating that the modelled facility is more efficient than most (Kaya et al., 2002; Laboratory; Risi, 1995; Saidur et al., 2010). This is important to note as less efficient the facilities offer higher opportunities for energy savings.

Table 6 below displays the average compressor power (kW) required to supply each zone in the facility. These averages are calculated from the 1600 instances generated in the simulation phase. The average air leaks (cfm) and pressure drop (psi) can also be seen for each zone. Finally the air leaks and pressure drop are converted to kW losses which are seen in the last column.

Table 6

Table 6 displays the average parameters of all 1600 instances generated in the simulation phase. As illustrated the optimal location with no articulation of user preference is zone 83.

Zone	Avg. Compressor Power (kW)	Air Leaks (CFM)	Air Leaks (kW)	Pressure Drop (psi)	Pressure Drop (kW)	Total Losses (kW)
1	36.55	26.46	3.99	12.32	4.60	8.58
2	36.00	24.22	3.65	11.08	4.13	7.78
3	35.51	22.17	3.34	9.98	3.72	7.06
4	35.16	20.68	3.12	9.20	3.43	6.55
5	34.99	19.94	3.00	8.81	3.29	6.29
6	34.90	19.57	2.95	8.62	3.22	6.16
7	34.81	19.19	2.89	8.43	3.15	6.04
8	34.81	19.19	2.89	8.43	3.15	6.04
9	34.90	19.57	2.95	8.62	3.22	6.16
10	34.99	19.94	3.00	8.81	3.29	6.29
11	35.16	20.68	3.12	9.20	3.43	6.55
12	35.42	21.80	3.28	9.78	3.65	6.93
13	35.69	22.92	3.45	10.38	3.87	7.32
14	36.09	24.60	3.70	11.29	4.21	7.91
15	36.64	26.83	4.04	12.53	4.67	8.72
16	36.00	24.22	3.65	11.08	4.13	7.78
17	35.47	21.99	3.31	9.88	3.69	7.00
18	34.99	19.94	3.00	8.81	3.29	6.29
19	34.64	18.45	2.78	8.06	3.01	5.78
20	34.47	17.70	2.67	7.68	2.87	5.53
21	34.38	17.33	2.61	7.50	2.80	5.41
22	34.30	16.96	2.55	7.32	2.73	5.28
23	34.30	16.96	2.55	7.32	2.73	5.28
24	34.38	17.33	2.61	7.50	2.80	5.41
25	34.47	17.70	2.67	7.68	2.87	5.53
26	34.64	18.45	2.78	8.06	3.01	5.78
27	34.90	19.57	2.95	8.62	3.22	6.16
28	35.16	20.68	3.12	9.20	3.43	6.55
29	35.56	22.36	3.37	10.08	3.76	7.13
30	36.09	24.60	3.70	11.29	4.21	7.91
31	35.56	22.36	3.37	10.08	3.76	7.13
32	35.03	20.13	3.03	8.91	3.32	6.35
33	34.56	18.08	2.72	7.87	2.94	5.66
34	34.22	16.58	2.50	7.13	2.66	5.16
35	34.05	15.84	2.39	6.77	2.53	4.91
36	33.96	15.47	2.33	6.59	2.46	4.79
37	33.88	15.09	2.27	6.41	2.39	4.67
38	33.88	15.09	2.27	6.41	2.39	4.67
39	33.96	15.47	2.33	6.59	2.46	4.79
40	34.05	15.84	2.39	6.77	2.53	4.91
41	34.22	16.58	2.50	7.13	2.66	5.16
42	34.47	17.70	2.67	7.68	2.87	5.53
43	34.73	18.82	2.83	8.24	3.08	5.91
44	35.12	20.50	3.09	9.10	3.39	6.48
45	35.65	22.73	3.42	10.28	3.83	7.26
46	35.20	20.87	3.14	9.29	3.47	6.61
47	34.68	18.63	2.81	8.15	3.04	5.85
48	34.22	16.58	2.50	7.13	2.66	5.16
49	33.88	15.09	2.27	6.41	2.39	4.67
50	33.71	14.35	2.16	6.06	2.26	4.42
51	33.63	13.98	2.10	5.88	2.19	4.30
52	33.55	13.60	2.05	5.71	2.13	4.18
53	33.55	13.60	2.05	5.71	2.13	4.18
54	33.63	13.98	2.10	5.88	2.19	4.30
55	33.71	14.35	2.16	6.06	2.26	4.42
56	33.88	15.09	2.27	6.41	2.39	4.67
57	34.13	16.21	2.44	6.95	2.59	5.03
58	34.38	17.33	2.61	7.50	2.80	5.41
59	34.77	19.01	2.86	8.34	3.11	5.97
60	35.29	21.24	3.20	9.49	3.54	6.74
61	34.94	19.75	2.98	8.72	3.25	6.23

Zone	Avg. Compressor Power (kW)	Air Leaks (CFM)	Air Leaks (kW)	Pressure Drop (psi)	Pressure Drop (kW)	Total Losses (kW)
62	34.43	17.52	2.64	7.59	2.83	5.47
63	33.96	15.47	2.33	6.59	2.46	4.79
64	33.63	13.98	2.10	5.88	2.19	4.30
65	33.46	13.23	1.99	5.54	2.06	4.06
66	33.38	12.86	1.94	5.36	2.00	3.94
67	33.30	12.48	1.88	5.19	1.94	3.82
68	33.30	12.48	1.88	5.19	1.94	3.82
69	33.38	12.86	1.94	5.36	2.00	3.94
70	33.46	13.23	1.99	5.54	2.06	4.06
71	33.63	13.98	2.10	5.88	2.19	4.30
72	33.88	15.09	2.27	6.41	2.39	4.67
73	34.13	16.21	2.44	6.95	2.59	5.03
74	34.51	17.89	2.69	7.78	2.90	5.60
75	35.03	20.13	3.03	8.91	3.32	6.35
76	34.73	18.82	2.83	8.24	3.08	5.91
77	34.22	16.58	2.50	7.13	2.66	5.16
78	33.75	14.53	2.19	6.15	2.29	4.48
79	33.42	13.04	1.96	5.45	2.03	4.00
80	33.26	12.30	1.85	5.11	1.90	3.76
81	33.18	11.93	1.80	4.94	1.84	3.64
82	33.10	11.55	1.74	4.81	1.79	3.53
83	33.09	11.55	1.74	4.77	1.78	3.52
84	33.18	11.93	1.80	4.94	1.84	3.64
85	33.26	12.30	1.85	5.11	1.90	3.76
86	33.42	13.04	1.96	5.45	2.03	4.00
87	33.67	14.16	2.13	5.97	2.23	4.36
88	33.92	15.28	2.30	6.50	2.43	4.73
89	34.30	16.96	2.55	7.32	2.73	5.28
90	34.81	19.19	2.89	8.43	3.15	6.04
91	34.86	19.38	2.92	8.53	3.18	6.10
92	34.34	17.14	2.58	7.41	2.76	5.35
93	33.88	15.09	2.27	6.41	2.39	4.67
94	33.55	13.60	2.05	5.71	2.13	4.18
95	33.38	12.86	1.94	5.36	2.00	3.94
96	33.30	12.48	1.88	5.19	1.94	3.82
97	33.22	12.11	1.82	5.02	1.87	3.70
98	33.22	12.11	1.82	5.02	1.87	3.70
99	33.30	12.48	1.88	5.19	1.94	3.82
100	33.38	12.86	1.94	5.36	2.00	3.94
101	33.55	13.60	2.05	5.71	2.13	4.18
102	33.80	14.72	2.22	6.24	2.33	4.54
103	34.05	15.84	2.39	6.77	2.53	4.91
104	34.43	17.52	2.64	7.59	2.83	5.47
105	34.94	19.75	2.98	8.72	3.25	6.23
106	35.03	20.13	3.03	8.91	3.32	6.35
107	34.51	17.89	2.69	7.78	2.90	5.60
108	34.05	15.84	2.39	6.77	2.53	4.91
109	33.71	14.35	2.16	6.06	2.26	4.42
110	33.55	13.60	2.05	5.71	2.13	4.18
111	33.46	13.23	1.99	5.54	2.06	4.06
112	33.38	12.86	1.94	5.36	2.00	3.94
113	33.38	12.86	1.94	5.36	2.00	3.94
114	33.46	13.23	1.99	5.54	2.06	4.06
115	33.55	13.60	2.05	5.71	2.13	4.18
116	33.71	14.35	2.16	6.06	2.26	4.42
117	33.96	15.47	2.33	6.59	2.46	4.79
118	34.22	16.58	2.50	7.13	2.66	5.16
119	34.60	18.26	2.75	7.96	2.97	5.72
120	35.12	20.50	3.09	9.10	3.39	6.48
121	35.56	22.36	3.37	10.08	3.76	7.13
122	35.03	20.13	3.03	8.91	3.32	6.35
123	34.56	18.08	2.72	7.87	2.94	5.66
124	34.22	16.58	2.50	7.13	2.66	5.16
125	34.05	15.84	2.39	6.77	2.53	4.91
126	33.96	15.47	2.33	6.59	2.46	4.79
127	33.88	15.09	2.27	6.41	2.39	4.67
128	33.88	15.09	2.27	6.41	2.39	4.67
129	33.96	15.47	2.33	6.59	2.46	4.79

Zone	Avg. Compressor Power (kW)	Air Leaks (CFM)	Air Leaks (kW)	Pressure Drop (psi)	Pressure Drop (kW)	Total Losses (kW)
130	34.05	15.84	2.39	6.77	2.53	4.91
131	34.22	16.58	2.50	7.13	2.66	5.16
132	34.47	17.70	2.67	7.68	2.87	5.53
133	34.73	18.82	2.83	8.24	3.08	5.91
134	35.12	20.50	3.09	9.10	3.39	6.48
135	35.65	22.73	3.42	10.28	3.83	7.26
136	36.09	24.60	3.70	11.29	4.21	7.91
137	35.56	22.36	3.37	10.08	3.76	7.13
138	35.07	20.31	3.06	9.00	3.36	6.42
139	34.73	18.82	2.83	8.24	3.08	5.91
140	34.56	18.08	2.72	7.87	2.94	5.66
141	34.47	17.70	2.67	7.68	2.87	5.53
142	34.38	17.33	2.61	7.50	2.80	5.41
143	34.38	17.33	2.61	7.50	2.80	5.41
144	34.47	17.70	2.67	7.68	2.87	5.53
145	34.56	18.08	2.72	7.87	2.94	5.66
146	34.73	18.82	2.83	8.24	3.08	5.91
147	34.99	19.94	3.00	8.81	3.29	6.29
148	35.25	21.06	3.17	9.39	3.50	6.67
149	35.65	22.73	3.42	10.28	3.83	7.26
150	36.18	24.97	3.76	11.49	4.29	8.05

Fig. 22, Fig. 23 and Fig. 24 all show that Zone 83 is the optimal location with no articulation of user preference. As it can be seen in Table 7 and 8 Zone 83 has a 99.6% probability of correct selection. Zone 82 was the alternative optimal location for 0.4% of the instances. The difference in energy savings at Z83 in comparison to Z82 is 0.023%.

To investigate the effect of user preference in the model, a sensitivity analysis is performed by assigning a hundred different weights (α) to the model. Fig. 25, Fig. 26, and Fig. 27 show three tested weights of α. The sensitivity analysis resulted in a high probability of three distinct optimal zones. For $w \in [0, 0.05]$, Zone 68 (Z68) is found to be optimal for 95.3% of instances. Zone 128 (Z128) is found to be optimal for $w \in (0.05, 0.15]$ for 97.8 of instances. Finally for $w \in (0.15, 1]$, Zone 143 (Z143), is found to be optimal for 97.7%, of instances. Fig. illustrates the locations of these zones in the facility. Table 7 and Table 8 show the effect of locating the air compressor at each identified zone.

A 3-D representation of the power (kW) required by the air compressor to meet air demand at each zone in the facility is shown in Fig. 24. Since the model is set to minimize the power, the lowest calculated kW results in the most energy efficient zone. By increasing the weight of α in the model, the optimal zone shifts to a zone with a higher kW output, but a more acceptable user preference.

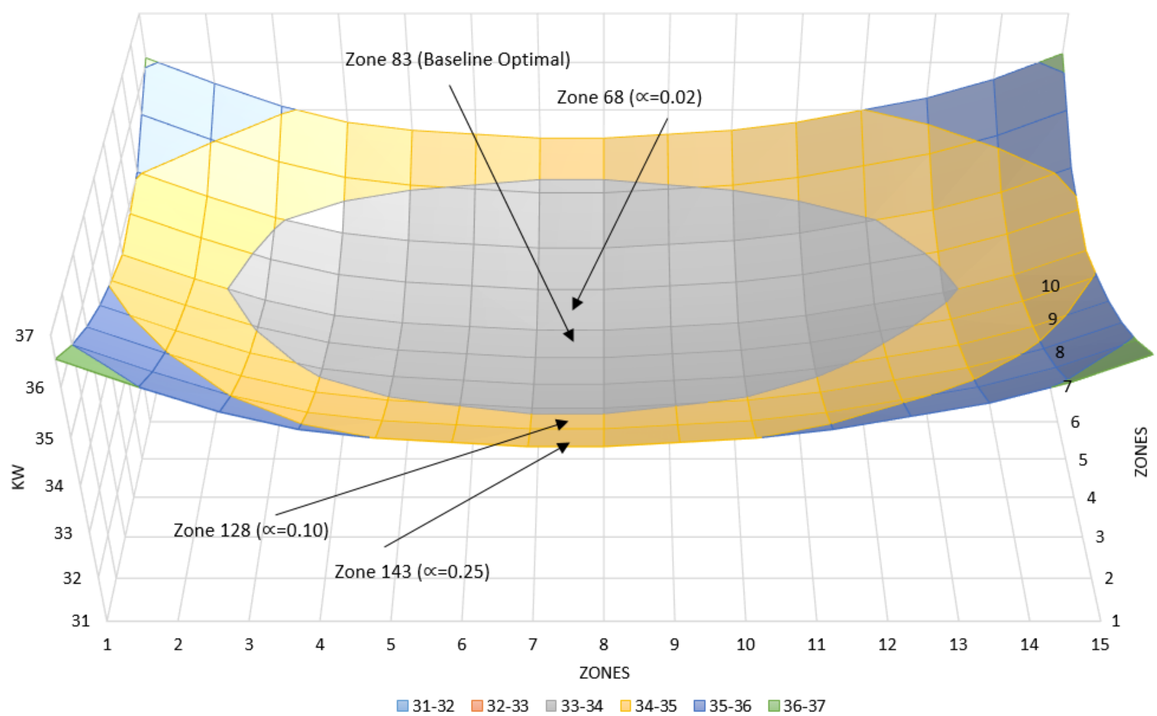

Fig. 24. 3-D Representation of kW at Each Zone

As it can be seen, the higher the user preference in the model, the further away the air compressor is from the most energy efficient zone (Z83). The energy intensity of the air compressor at each zone in the facility is further illustrated in Fig. 25. The shades of orange represent the level of kW required by the air compressor to meet demand. As the orange shade darkens the kW requirement increases. The lightest shade of orange can be found towards the center of the facility, where Z83 and Z68 are located.

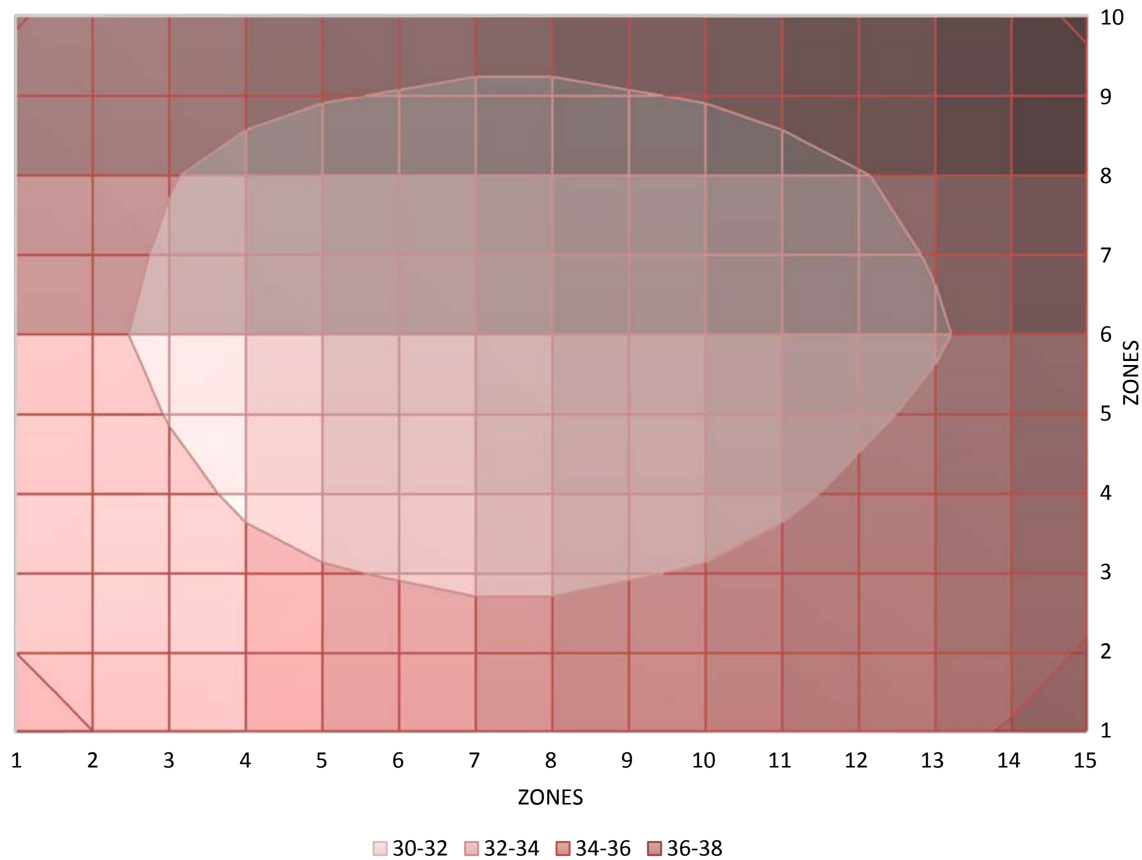

Fig. 25. Energy Intensity of Air Compressor at Each Zone

The variations in the model represented by the instances also causes a change in cfm demand for each replication. The resulting range of cfm demand in the facility is 199.2 cfm to 261.18 cfm. This will similarly result in a variation in potential energy savings for each iteration. Table 7 and Table 8 display the resulting range in energy savings for each of the identified optimal zones.

Savings are benchmarked on the least efficient air compressor location (Zone 15) in the facility. For the modelled facility, Zone 15 will require an average compressor size of 36.69 kW to meet demand. Placing the air compressor in the most energy efficient location (Z83) results in an energy savings of 8.52% to 12.22% with an average energy savings of 10.703%. Zone 83 will require an average compressor size of 33.1 kW to meet demand. When considering user preference, energy savings are less but still significant with Zone 68 saving 7.90% to 11.61% with an average savings of 10.025%. Zone 68 will require a compressor size of 33.299 kW to meet demand. Zone 128 resulted in possible energy savings range of 6.14% to 9.77% with an average savings of 8.14% and finally Zone 143 resulted in possible energy savings range of 4.92% to 8.21% with an average savings of 6.55%. Zone 128 and 143 will require 33.879 kW and 34.385 kW respectively to meet air demand. There is a 4.53% loss in energy efficiency as the air compressor location shifts from Zone 83 (most energy efficient) to Zone 143 (most user preferred zone). Nonetheless Zone 143 will still save an average of 6.55% in energy use. More detailed results can be seen in Table 7 below.

Table 7

Results of instances on the model. Table 7 display the probability of correct selection of the optimal zones, the range of savings and average savings for all instances of the model. Savings are benchmarked on the least efficient air compressor location (Zone 15) in the facility. Table 7 displays savings in percent format. Note that Zone 83 and 82 are optimal with no articulation of user preference.

Table 7	Optimal Zone Savings				Alternative Zone Savings				
	Zone	Probability (%)	Range (%)	Average (%)	Zone	Probability (%)	Range (%)	Average (%)	Difference in Avg. Savings (%)
Energy Optimal No Preference	83	99.6	8.52-12.22	10.703	82	0.4	8.52-12.24	10.68	0.023
Optimal for w∈ [0, 0.05]	68	95.3	7.90-11.61	10.025	67	4.7	7.65-11.34	9.75	0.271
Optimal for w∈(0.05, 0.15]	128	97.8	6.14-9.77	8.142	127	2.2	6.01-9.64	8.09	0.133
Optimal for w∈ (0.15, 1]	143	97.7	4.92-8.21	6.550	143	2.3	4.82-8.08	6.42	0.132

Table 8

Table 8 displays the probability of correct selection of the optimal zones, the range of savings and average savings for all instances of the model. Savings are benchmarked on the least efficient air compressor location (zone 15) in the facility. Table 8 displays savings in kW format.

Table 8	Optimal Zone Savings				Alternative Zone Savings				
	Zone	Probability (%)	Range (kW)	Average (kW)	Zone	Probability (%)	Range (kW)	Average (kW)	Difference in Avg. Savings (kW)
Energy Optimal No Preference	83	99.6	28.5-39.8	33.095	82	0.4	28.5-38.8	33.102	0.0068
Optimal for w∈ [0, 0.05]	68	95.3	28.7-39.1	33.299	67	4.7	28.7-39.2	33.381	0.0820
Optimal for w∈(0.05, 0.15]	128	97.8	29.1-39.7	33.879	127	2.2	29.1-39.8	33.920	0.0418
Optimal for w∈ (0.15, 1]	143	97.7	29.5-40.3	34.385	143	2.3	29.5-40.3	34.428	0.0426
Least Efficient	15	N/A	30.6-43.6	36.637	N/A	N/A	N/A	N/A	N/A

As displayed in Fig. 24 and Fig. 25, as well as Table 7 and 8, as the weight of user preference increases the optimal location shifts from Zone 83, to Zone 68, Zone 128, and Zone 143 respectively. This shift reduces the potential energy savings achievable by the model, but increases the user's preference for location. Even with a high user preference the model can still achieve significant savings.

To show the effect of the effect of α on the shifting optimal zone location, the baseline kW profile of the air compressor($w = 0$) is plotted against three profiles of energy/performance output for w = 0.02, 0.10, and 0.25. The selected values of α fall within the range of α that identifies optimality at zones Z68, Z128, and Z143 (See Fig. 26, Fig. 27, and Fig. 28).

By selecting w of 0.02 (Fig. 26), the zone shifts from Z83 to Z68. This results in an increase in user preference from a non-applicable zone NA to a zone with a preference index of 4. The orange bars show the weighted kW consumption. To find the resulting kW consumption at the newly identified zone, the blue baseline bar is used with the corresponding orange bar. For example to find the optical zone with $w = 0.02$, the minimum orange bar is identified and traced down to the blue baseline which will give you the kW consumption at that zone. To illustrate the effect of α on the air compressor location, $w = 0.02, 0.10, and\ 0.25$ are displayed below.

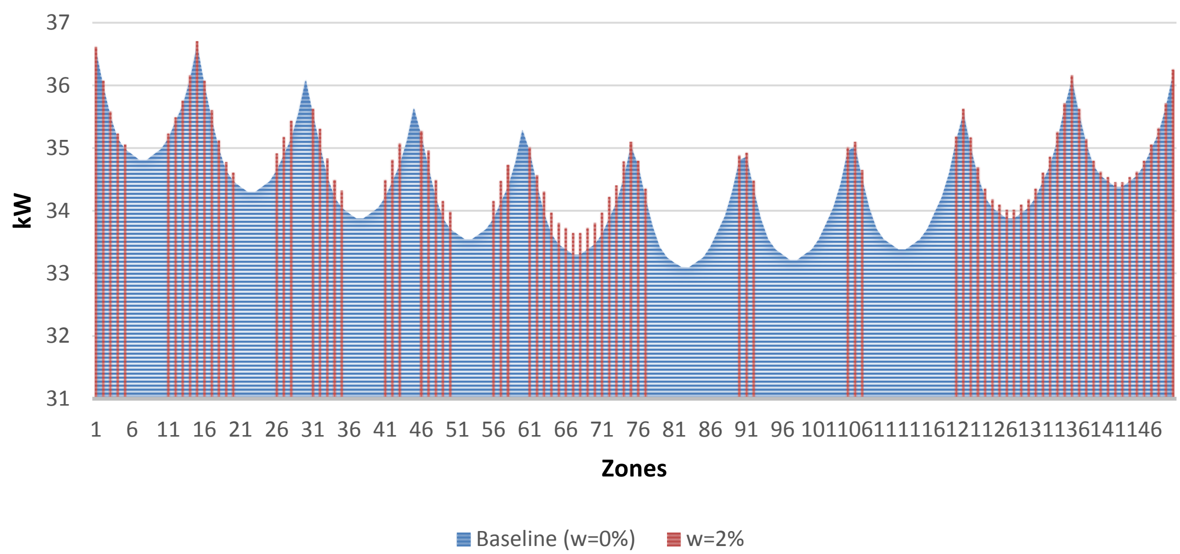

Fig. 26. Baseline Energy vs. Energy with User Preference w = 0.02

Using $w = 0.02$ will result in the optimal location shifting to zone 68. The model is next run using $\mathrm{w} = 0.10$ (Fig. 27) & $\mathrm{w} = 0.25$ (Fig. 28). The resulting optimal location shifts from Z68 to Z128 and Z128 to Z143 respectively. The shift also results in more convenient user preference of level 4 to level 2 and level 2 to level 1.

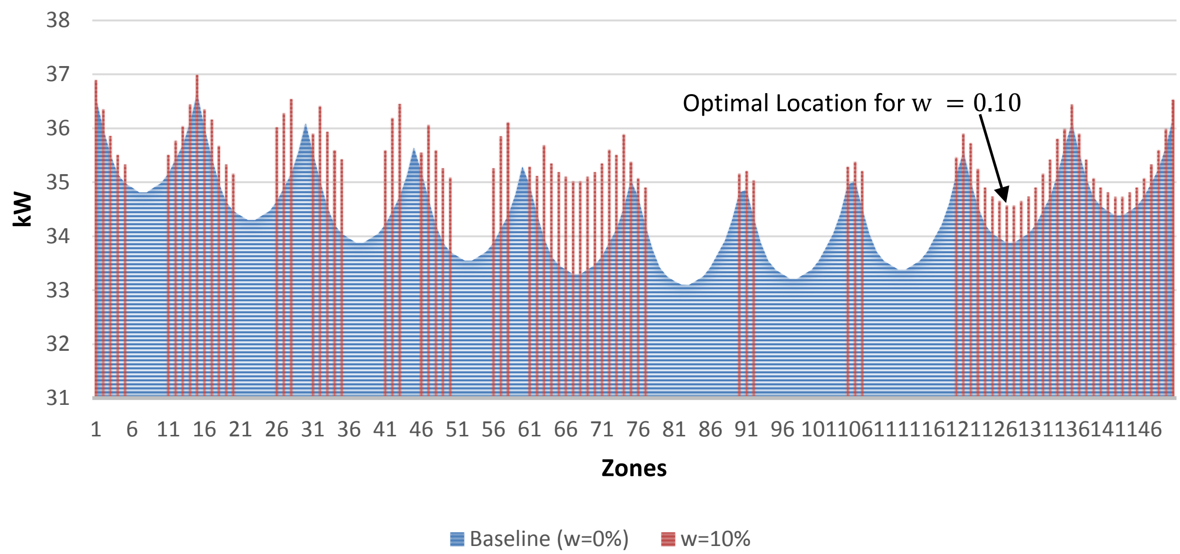

Fig. 27. Baseline Energy vs. Energy with User Preference w = 0.10

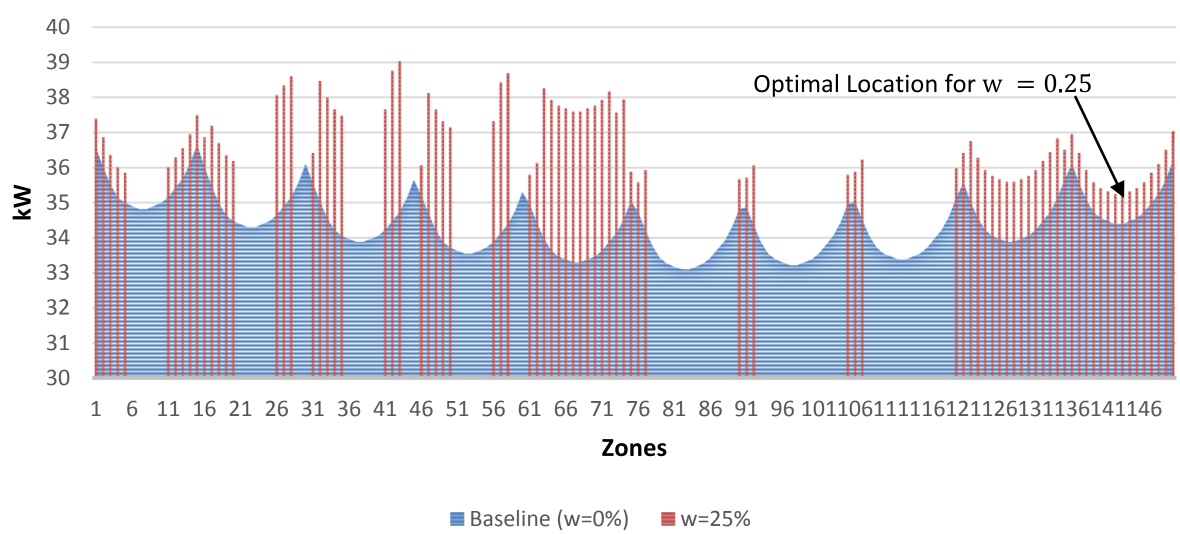

Fig. 28. Baseline Energy vs. Energy with User Preference w = 0.25

Lastly a sensitivity analysis is conducted to show the effect of increasing α on the resulting optimal locations (See Fig. 29). The kW of all zones are plotted in ascending order. Areas shaded red are zones that are not appropriate for air compressor location (NA). Similar to Fig. 26, Fig. 27, and Fig. 28 the w plots are only there to identify the optimal location. Once the lowest point on the w plot is located, the reader must trace down to the baseline plot to find the kW draw of the air compressor at that zone.

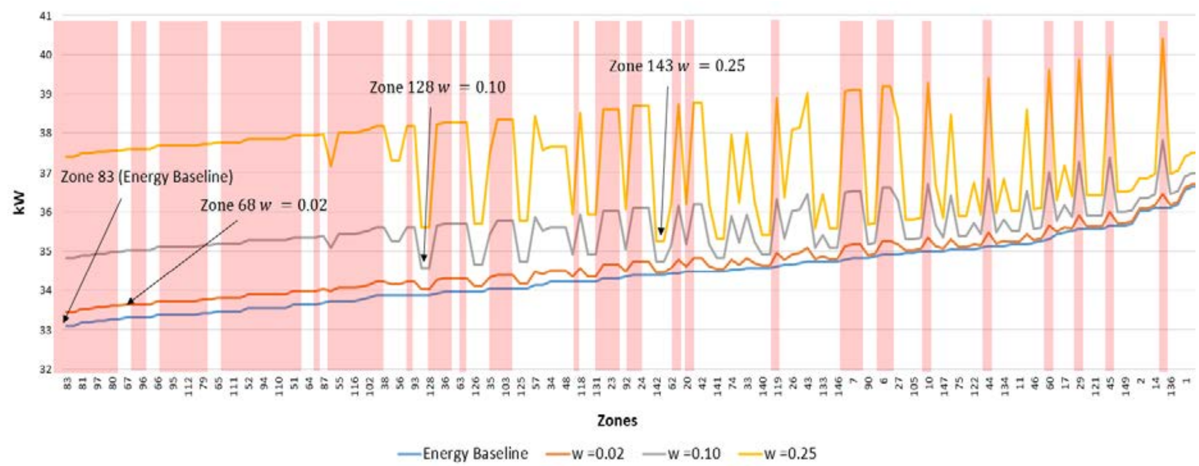

Fig. 29. Energy Baseline vs. Energy with User Preference w = 0.02, 0.10, and 0.25.

The energy cost (kW) vs. user preference trade-off is clearly displayed by the resulting optimal zone shifting to the right as the weight (wE) of the user preference increases.

4.5 Effects of Correct Compressor Sizing

Correct air compressor sizing is critical for improved energy efficiency. In fact, according to (Saidur et al., 2010), “two of the most important factors influencing the cost of compressed air are the type of compressor control and the proper compressor sizing. Oversized compressors and compressors operating in inefficient control modes have the highest unit energy and the highest annual operating costs”.

Air Demand (cfm) is the primary determinant of air compressor size in facilities. In this section the air demand required at each of the four identified optimal zones (Z83, Z68, Z128, and Z147) is compared with four traditional sizing methodologies used by highly respected industry leaders (Atlas Copco, Kaeser, Quincy Compressors, and Champion Compressors).

The air demand of zones Z83, Z68, Z128, and Z147 is calculated using the 1600 instances generated during the simulation phase in section 3.2. Two separate cfm requirements are considered. The first is the base air demand resulting from the machines present in the facility and their associated load factors, and the second is based on the simulated number and size of the air leaks in the facility. A normal density curve is generated for the 1600 instances and a 95% confidence interval is used to find both the upper and lower bounds of the air demand required. The upper bounds of both the base load cfm and air leak cfm are then added and used so as to compare our results with the other approaches. The upper and lower bounds of the base load remain the same for each zone in the facility, however, the upper and lower bounds of the air leaks vary depending on the location of the zone. The mean and standard deviation for the base cfm load were found to be 228.18 ± 10.25 cfm. A Z^* Score of 1.96 is used in the analysis. Using the upper bound of the 95% confidence interval allows the analysis to account for 97.5% of possible compressed air demand instances in the facility.

The four sizing methodologies are then independently applied to the machine, cfm requirement, and load factor data to determine the cfm demand for the entire facility. One of the compressor producers, Champion, suggests employing a load factor of 35% to all the required cfm's of the pneumatic machines in the facility as a rule of thumb. Once this is complete, they suggest a duty cycle of 75% for the air compressor and therefore increase the calculated cfm by 25% resulting in air demand for the modelled facility of 559.5 cfm (Sill, 2008).

Kaeser and Quincy suggest calculating cfm demand based on the machines in the facility and their respective load factors. However, once the base load is calculated, Keaser recommends adding an additional 10% of load for leakages, 15% for errors, and 20% as a reserve (Kaeser, 2000). Similarly Quincy adds 25% to the final cfm demand for future growth, additional tools, and eventual air leaks (Quincy, 2015).

Last, Atlas Copco recommends adding a safety margin of 10%-20% (Copco, 1998). To account for this in our analysis, a percent increase between 10% and 20% was simulated two hundred times. A normal density curve was then generated, and a 95% confidence interval was used. The mean and standard deviation for the base cfm load was found to be 262.4 ± 16.13 cfm.

To realistically account for energy losses by over-sized compressors, the type of compressor control is also considered in the analysis. Controls greatly affect the energy efficiency of the air compressor, hence seven different control strategies are accounted for.

Table 9 and 10 displays air compressor loads and their respective savings in cfm only. kW savings are not displayed as it significantly increases the complexity of the calculations. Introducing kW will result in the introduction of various manufacturers' compressor offerings and motor efficiencies, which is not within the scope of this research. Looking at Table 9 and 10, the compressor load column represents the average load of the air compressor at any given point. The required cfm displayed in the Upper Bound column is compared with 228.17 cfm, which is the average cfm demand calculated using the 1600 instances modelled in the simulation analysis section. The sizing losses section displays the losses using each methodology and each control strategy.

Table 9

Statistical distribution of the data for all instances. Selected zones are the zones used in the analysis of the capacity problem.

	Identified Zones for Analysis				
Zones	83	68	128	147	15
Average (kW)	33.09	33.30	33.88	34.99	36.64
SD (kW)	1.59	1.61	1.66	1.77	1.96
Sample Size	1600.00	1600.00	1600.00	1600.00	1600.00
Margin of Error (%)	0.08	0.08	0.08	0.09	0.10
Upper Bound (kW)	33.17	33.38	33.96	35.07	36.73
Lower Bound (kW)	33.02	33.22	33.80	34.90	36.54
Max (kW)	38.83	39.06	39.72	41.14	43.59
Min (kW)	28.48	28.65	29.13	29.85	30.91
Range (kW)	10.35	10.41	10.59	11.29	12.68
95% Confidence Upper	36.28	36.51	37.19	38.52	40.55
95% Confidence Lower	29.91	30.08	30.56	31.46	32.72

Table 10

Resulting cfm demand using the different sizing methodologies. 95% confidence interval was calculated for applicable fields. The sizing losses vary depending on the control strategy used on the air compressors, hence the sizing losses column. Min, Max, and average loss result from seven different control strategies used.

Table 3	Parameters				Sizing Losses (%)		
Sizing Methodology	95% Confidence Interval (cfm)	Upper Bound (cfm)	Cfm Oversize (%)	Compressor Load (%)	Min Loss (%)	Max Loss (%)	Avg. Loss (%)
Zone 83 (Best Energy Zone)	213.9-265.5	265.5	0	85.94	1	13.06	7.53
Zone 68 $\alpha \in [0, 0.05]$	214.4-266.9	266.9	0.51	85.50	1	13.5	7.75
Zone 128 $\alpha \in (0.05, 0.15]$	215.9-270.7	270.7	1.94	84.30	1	14.7	8.35
Zone 147 $\alpha \in (0.15, 1]$	218.5-277.8	277.8	4.61	82.15	1	15.85	8.925
Zone 15 (Worst Energy Zone)	222.2-287.80	287.80	8.39	79.28	1	18.72	10.36
Atlas Copco	230.1-294.7	294.7	10.98	77.44	1	19.56	10.78
Quincy Air Comp.	285.37	285.37	5.64	81.35	1	16.65	9.325
Kaeser	306.0	306.0	22.54	70.13	1	24.87	13.435
Champion Comp.	559.5	559.5	95.97	43.85	1	38.15	20.075

As illustrated in Table 10 and Fig. 30, the performance of oversized compressors depends considerably on the type of control strategy used by the facility. The calculated sizing losses are based on seven different control strategies seen in Fig. 30. Using VFD's to control oversized compressors results in the most efficient strategy with compressors experiencing minimal losses ($\leq 1\%$ loss). The capacity analysis shows that VFD compressors can save up to 37.15% irrespective of the sizing methodology used. This is similar to findings by Nadal et al and Saidur et al which state that VFD's can save over 30% in compressor applications (Nadel, Wang, Liu, & McKane, 2001; Saidur, Mekhilef, Ali, Safari, & Mohammed, 2012). Unfortunately VFD air compressors are not yet universally used in manufacturing facilities, as their initial startup cost is significantly more than alternative control strategies. Inlet Modulation (without blowdown) is the least efficient control strategy, resulting in oversized compressors losing between 13.06% and 38.15% (See Fig. 11 and Table 10).

Fig. 30 illustrates the percent load of the air compressor for each savings methodology, and the effect of each control strategy on the power required by the compressor.

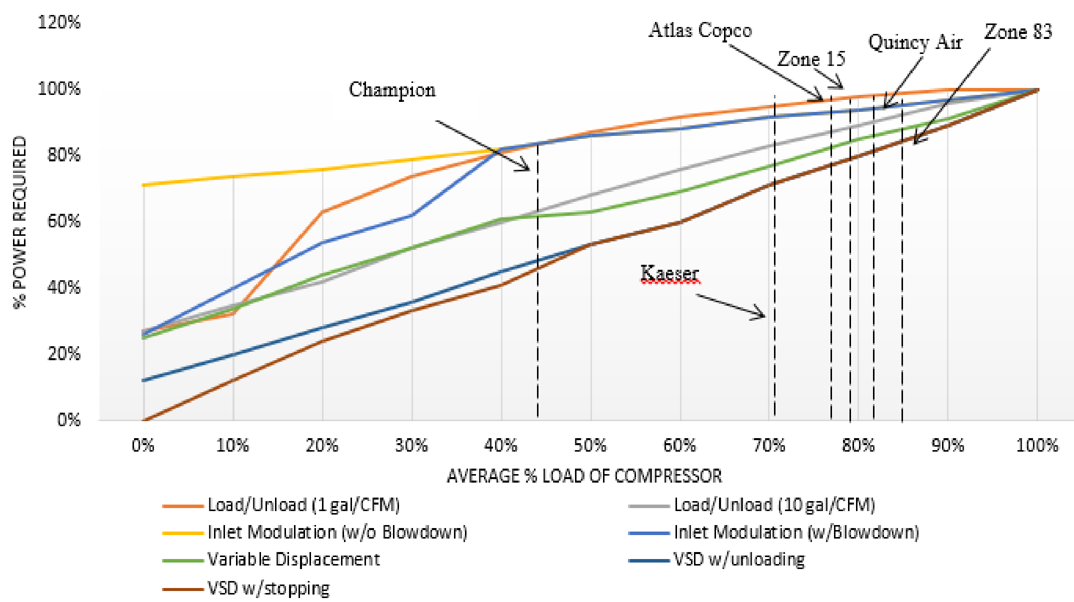

Fig. 30. Effect of Air Compressor Control Strategy on Compressor (Scales et al., 2009).

Clearly the higher the average load of the compressor the more efficient it is. This illustrates the importance of correct compressor sizing. As shown in Fig. 30 and Table 10, the sizing methodology that results in the highest average compressor load is by the proposed model. The model uses a simulation approach to calculate the air demand, actual leaks, and pressure drop in the system allowing for a more accurate compressor size. However it must be noted that using sizing methodologies provided by Atlas Copco and Quincy will lead to an acceptable air compressor sizing.

CHAPTER 5: CONCLUSION

A simulation-optimization framework is proposed to determine the best location and size of air compressors in dynamic manufacturing facilities. The developed model is designed to minimize the total energy consumption of the air compressor while simultaneously considering the user's preference for air compressor location. By considering air leak, air pressure losses, and various uncertainties, the model is able to simulate the performance of an air compressor at each zone in the manufacturing facility. Using this performance data the model identifies the zone that requires the least air compressor kW to meet cfm demand. The model simultaneously determines the size (cfm) of air compressor required at each zone. Additionally a sensitivity analysis is performed to investigate effect of the user's preference α on the model.

For the first model in Chapter 3, the results reveal a significant reduction in energy consumption. It is important to note that the model identified Zone 85 as the energy optimal location with 95.2% probability of correct selection. Placing the air compressor in the most energy efficient location (Z85) results in an energy savings of 8.59% to 13.94% with an average energy savings of 10.97%. Energy savings when the model considers user preference are less but still significant with Zone 70 saving 8.03% to 13.06% with an average savings of 10.31%. Zone 130 resulted in possible energy savings range of 6.27% to 11.38% with an average savings of 8.34% and finally Zone 145 resulted in possible energy savings range of 4.91% to 9.64% with an average savings of 6.28%. There is a 4.62% loss in energy efficiency as the air compressor location shifts from Zone 85 (most energy efficient) to Zone 145 (maximum user preference priority). Nonetheless Zone 145 is still able to save an average of 6.2% in energy usage.

To investigate and demonstrate the effectiveness of the simulation-driven analysis, a novel mathematical formulation using several standardized equations is used to compare the derived optimal zones (Z70, Z130, and Z145), with five other zones, including but not limited to the most energy efficient (Z85) and least energy efficient (Z1) zones.

The effect of the air compressor location is demonstrated by the impact that the eight selected locations have on the total kW required to meet compressed air demand in the facility. An energy savings of up to 2.98 kW or 10.8% can be achieved by placing the air compressor in the most energy efficient location Z85. Z70, Z130, and Z145 can save up to 10.11%, 8.1%, and 6.2% in energy consumption respectively. As expected, there is an increase in air compressor energy consumption as the weight of the user's preference increases in the objective function. The increase in energy consumption between Z85 (Energy Baseline) and Z70, Z130, and Z145 is 0.69%, 2.7%, and 4.6% respectively.

These results suggest that by using the proposed model to optimize the air compressor location in a facility, significant energy reductions can be achieved while maintaining the user's desired air compressor location. The results also suggest that as the weight of the user preference variable goes up, so will the associated energy of the resulting optimal zone.

The second model in Chapter 4 also reveals that a significant reduction in energy consumption is possible by optimizing the location of the air compressor (up to 10.7%). The level of energy reduction experienced using the proposed model is similar to that of the first model and of a preceding paper by Zahlan and Asfour. However by modelling the air leak and pressure losses, and by including additional uncertainties, the model can more accurately and realistically determine the best possible location and size of the air compressor.

To determine the correct air compressor size, several sizing methodologies including the proposed model are compared and tested using seven different air compressor control strategies. The savings resulting from correct air compressor sizing vary significantly between methodologies and control strategies. The sizing methodology resulting in the least losses to the facility was the proposed model. The VFD control strategy was most efficient with compressors experiencing minimal losses ($\leq 1\%$ loss). However using a modulating (without blowdown) control approach will result in oversized compressors losing between 13.06% and 38.15%. It must be noted that whichever sizing methodology or compressor control is selected, oversized compressors will lead to higher demand (kW), higher installation cost and higher maintenance cost to the facility.

This research shows that substantial energy savings can be achieved by correctly locating air compressors in manufacturing facilities. The introduced model also highlights potential savings from correctly sizing and controlling air compressors. The results also suggest that as the user's preference for location increases, the energy consumption of the air compressor will also increase. This correlation of user preference and energy consumption was expected. However, significant savings can still be achieved even at high user preference locations.

This novel mathematical model can be used by facility planners and energy managers to determine the best size and most efficient location for air compressors in manufacturing facilities. The established framework plays a significant role in creating an energy efficient and problem free compressed air system.

Compressed air systems account for 23% of total mechanical drive electric consumption in manufacturing facilities and are only 11% efficient. Having said that they are one of the least understood systems. In this research, the authors show that optimizing the location of air compressors in manufacturing facilities can reduce energy costs while considering the users preference for location. This is an important step to creating an optimal compressed air system.

FUTURE WORK

Improvements in efficiency of the air compressor motor have been investigated throughout the years by manufacturers. However there is a limited amount of research and time invested into total compressed air system efficiency. Therefore is significant opportunity for further research into improvements of the entire compressor system, the compressors relationship with the manufacturing process, the compressors performance in given particular orientation, and the opportunity to maximize the compressed air and losses (waste heat, air etc.) produced by the air compressor. This suggests that there are several opportunities to further research into compressed air location, capacity and waste optimization. These opportunities will be briefly detailed below.

Since the proposed objective function is tailored towards maximizing the energy efficiency of an air compressor system, its benefits are best realized when the model can effectively capture and integrate the user preference index in its analysis. Such a component will be more effective deducing the user’s preference via a prediction preference model. An applicable extension to this research would be to develop an effective and practical mechanism to accomplish this task.

A separate cost function, or penalty function can also be introduced into the model to better account for the effects of the air compressor noise in the facility and the costs of moving an existing compressor to a different location. This will allow the model to more accurately determine the best location in the facility and the practicality of moving the compressor. The cost function could also be used to determine the most effective number of compressors needed in a facility. It can help in calculating feasibility studies for incorporating alternative solutions.

It must also be noted that the proposed objective function is for a single air compressor setup in a facility. Further studies can be conducted to incorporate systems with multiple air compressors at multiple zones and variation in pressure at each zone into the model. For example if the facility has several machines that require very different pressure set points i.e. 50 psi versus 100 psi, a method of determining the feasibility of running two separate compressor systems can be incorporated into the model. This will allow for more efficient use of compressed air since the pressure set points will be better managed. This approach can be considered as a system of clustering several machines with similar attributes into groups.

Another expansion of research would be to incorporate waste heat into the model. Seventy percent of the energy put into the air compressor is wasted as heat. This heat is generally exhausted into the environment, but some facilities have begun recycling the waste heat produced by the air compressor and using it for space heating and process heat. By identifying zones that require waste heat in the facility, the model can incorporate the produced waste heat into the location optimization algorithm. This will allow the model to position the air compressor closest to locations were heat is most needed in the facility. Since waste heat accounts for 70% of compressor losses, it could have a significant effect on the resulting optimal air compressor location.

Acknowledgement

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APPENDIX A

RESULTS OF SENSITIVITY ANALYSIS ON OPTIMAL LOCATION (MODEL #1)

Supplementary Data File

Description:

The accompanying graphs show the results of the sensitivity analysis performed on the first model (Model #1) introduced. A sensitivity analysis is conducted to see the relationship between the energy consumption of the air compressor and the user's location preference. The sensitivity analysis is conducted on all possible weights of the user preference.

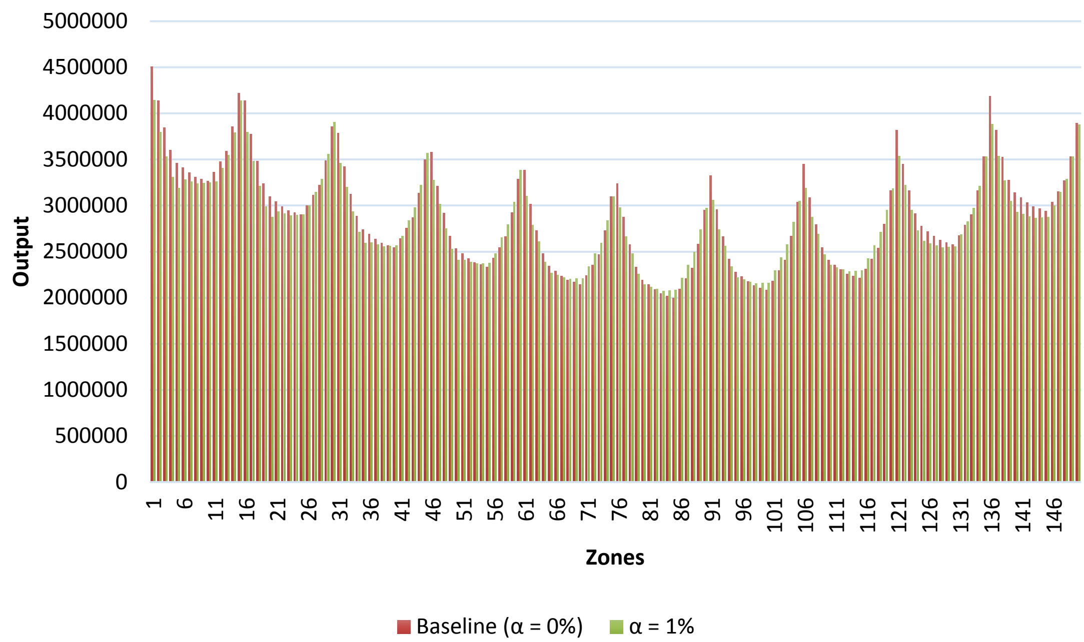


Fig. 1. Baseline Energy vs. Energy with User Preference α = 0.01 (Model #1)

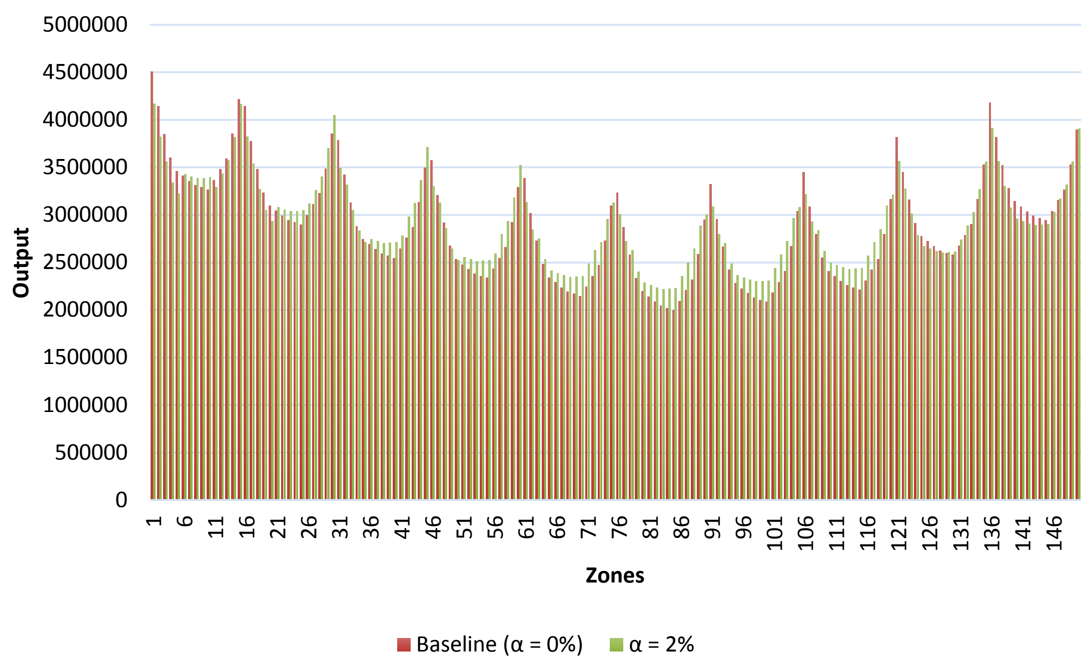


Fig. 2. Baseline Energy vs. Energy with User Preference α = 0.02 (Model #1)

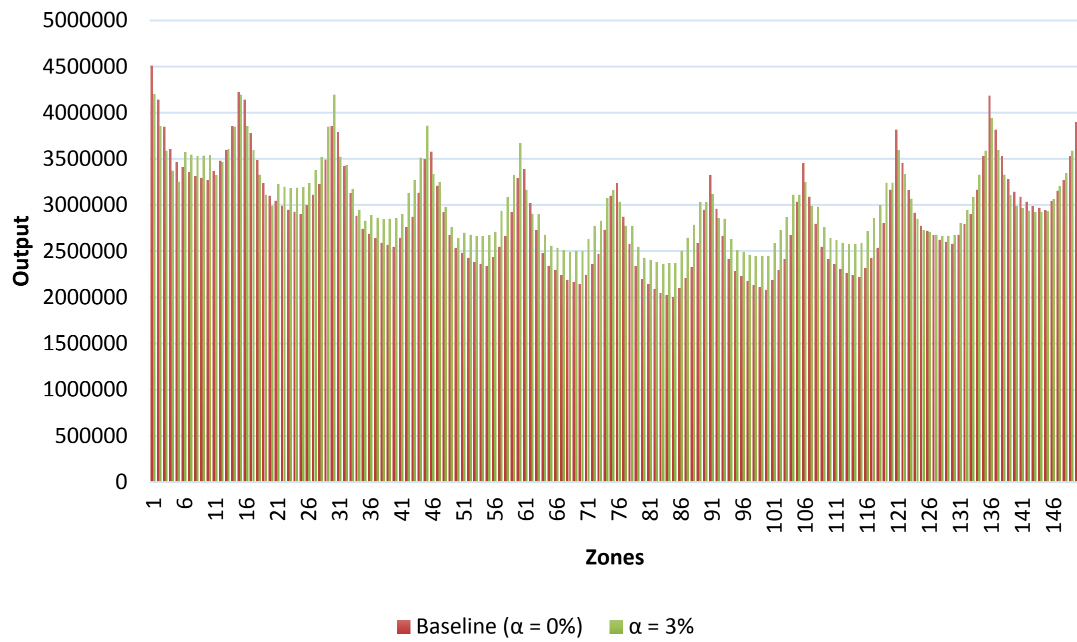


Fig. 3. Baseline Energy vs. Energy with User Preference α = 0.03 (Model #1)

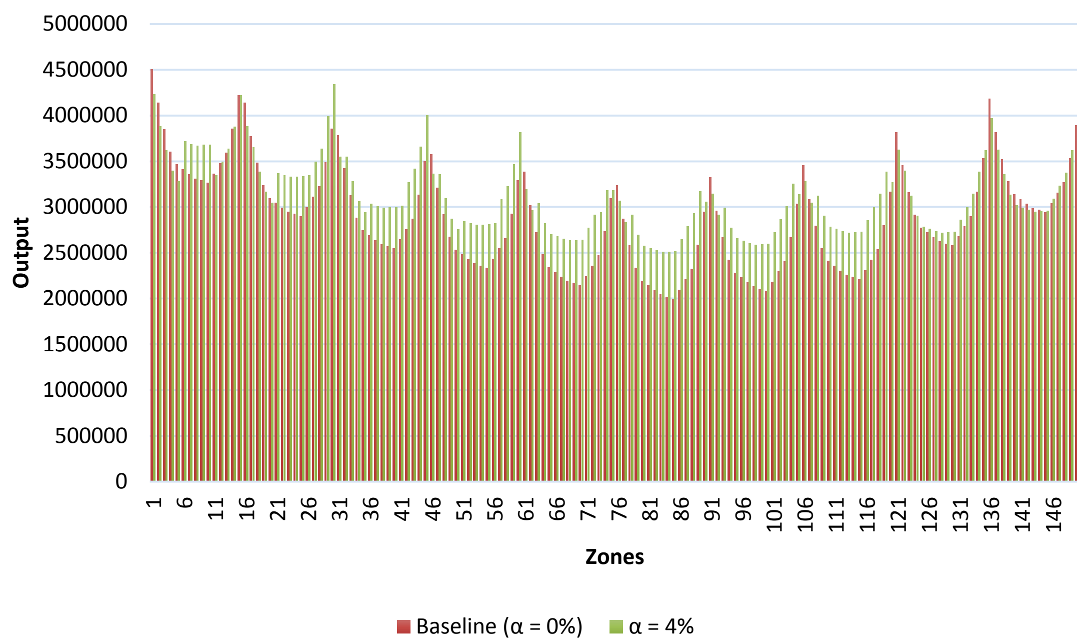


Fig. 4. Baseline Energy vs. Energy with User Preference α = 0.04 (Model #1)

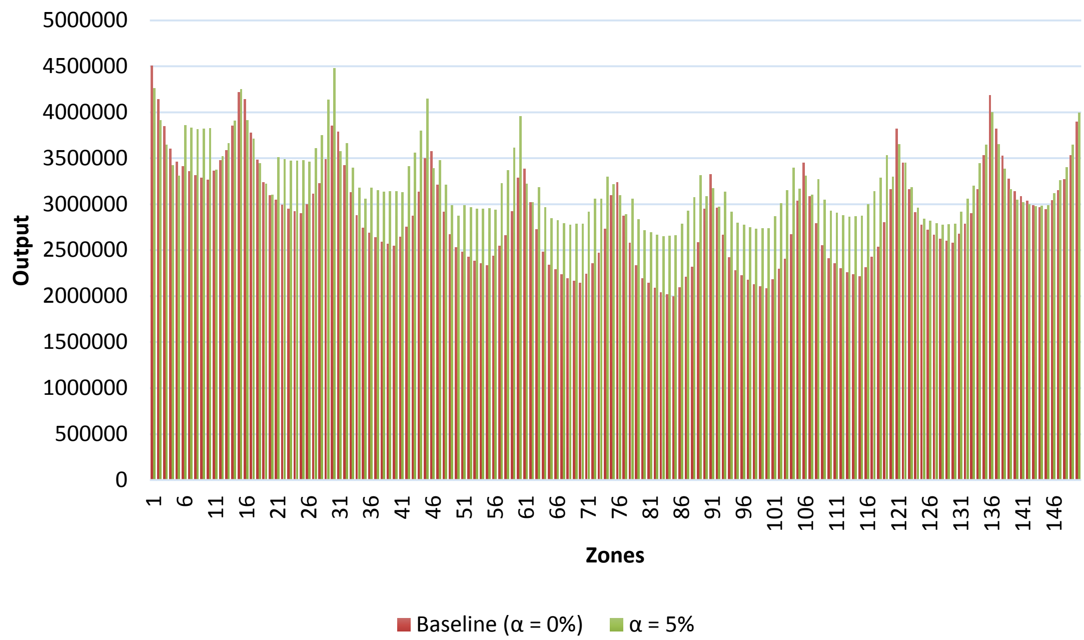


Fig. 5. Baseline Energy vs. Energy with User Preference α = 0.05 (Model #1)

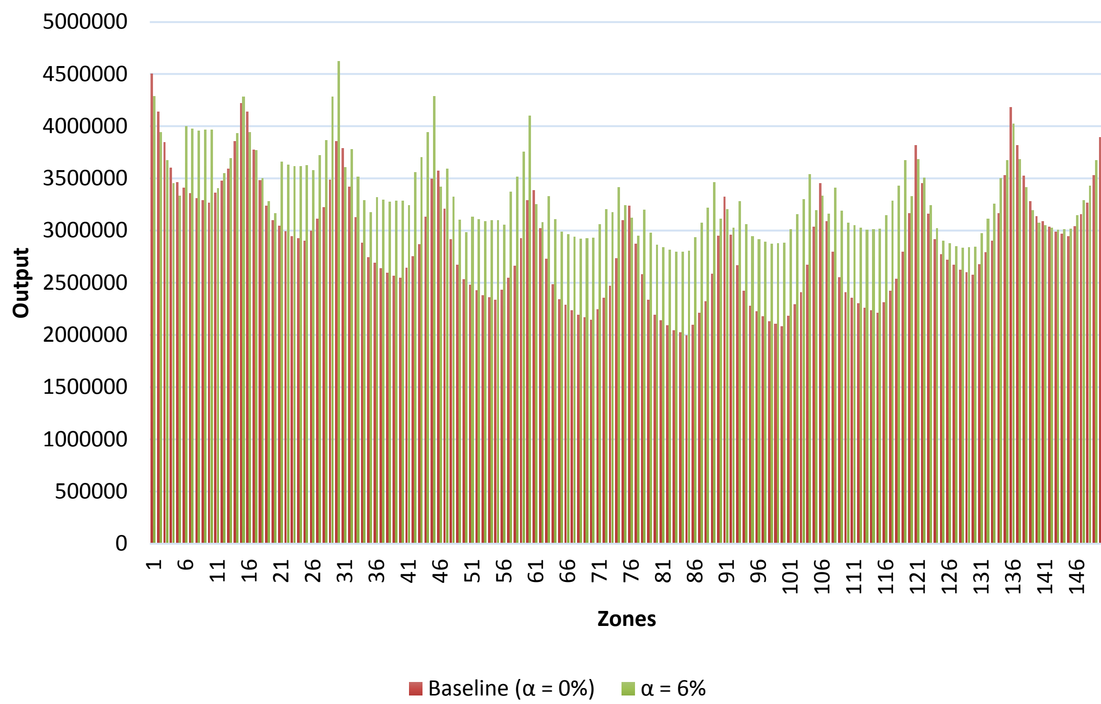


Fig. 6. Baseline Energy vs. Energy with User Preference α = 0.06 (Model #1)

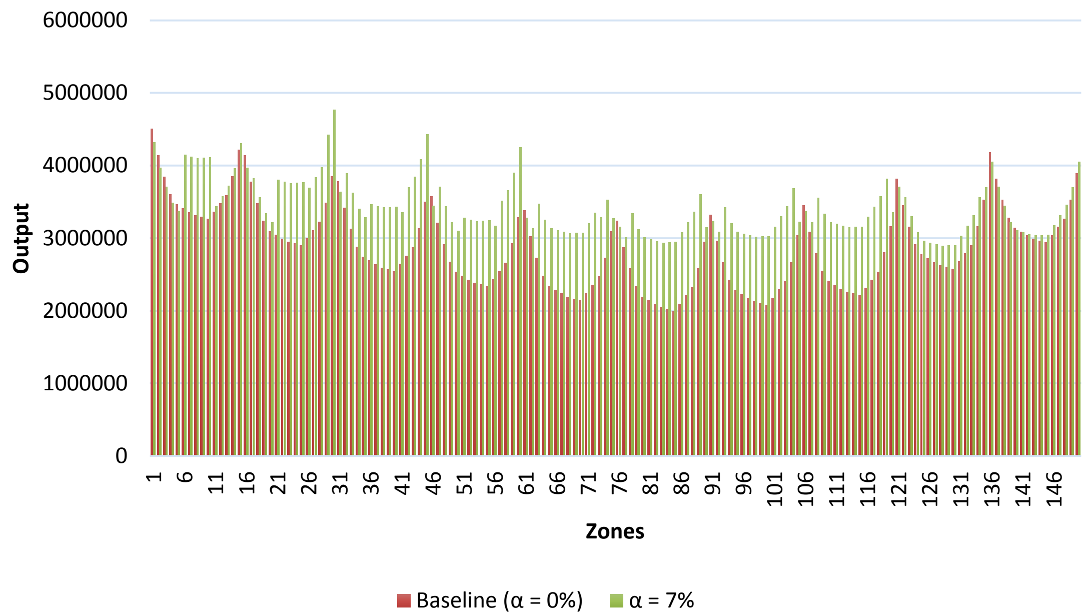


Fig. 7. Baseline Energy vs. Energy with User Preference α = 0.07 (Model #1)

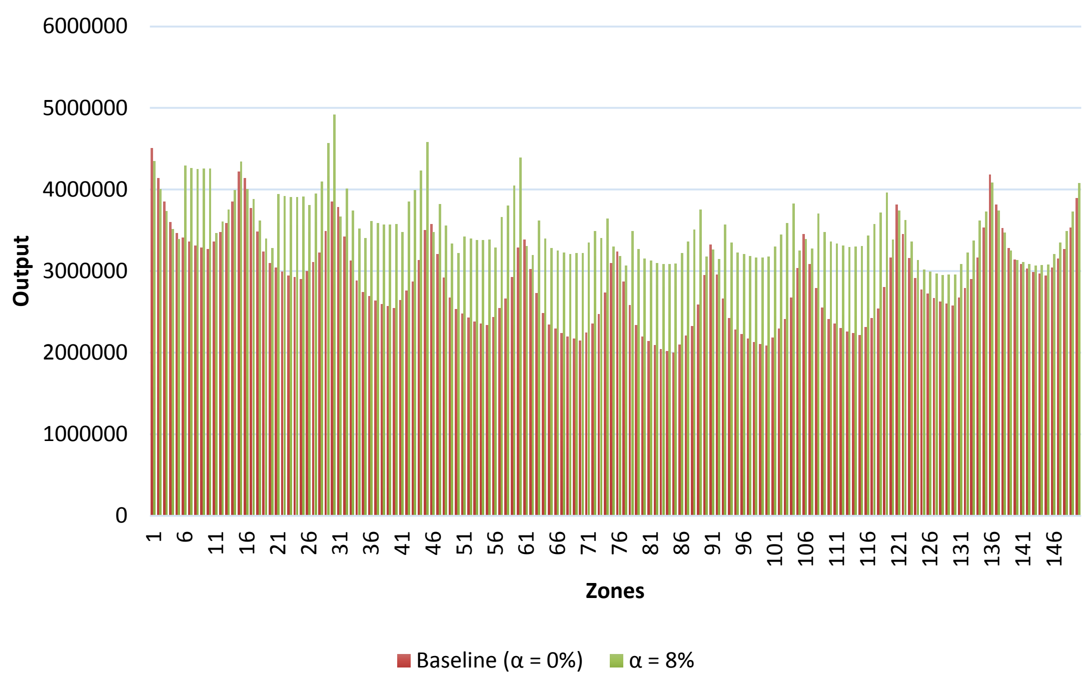


Fig. 8. Baseline Energy vs. Energy with User Preference α = 0.08 (Model #1)

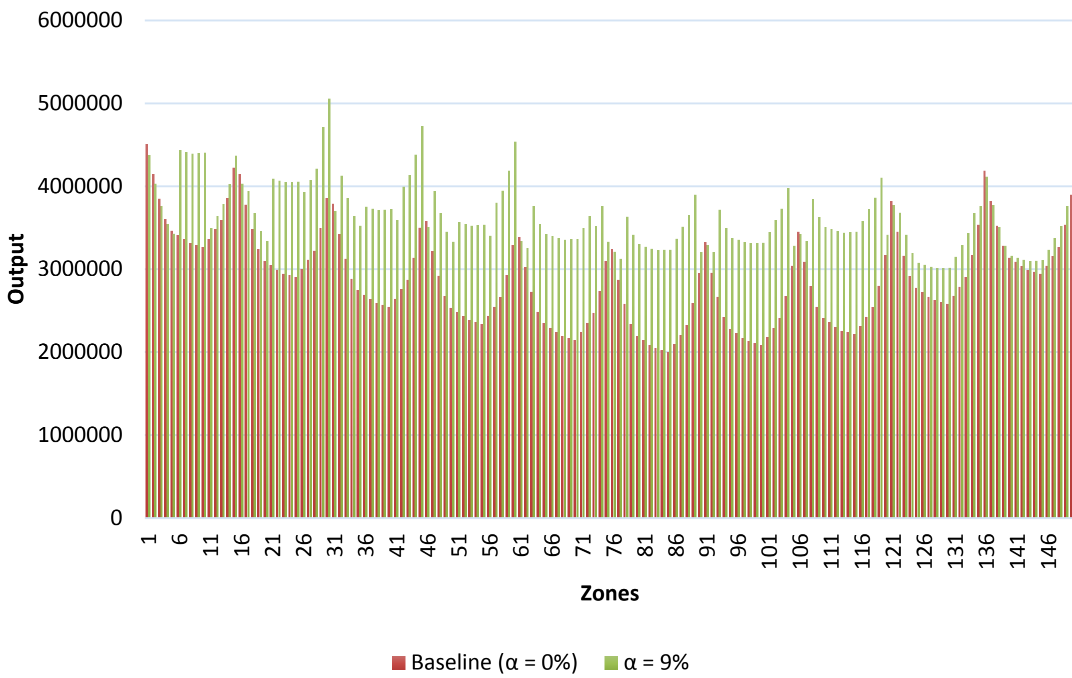


Fig. 9. Baseline Energy vs. Energy with User Preference α = 0.09 (Model #1)

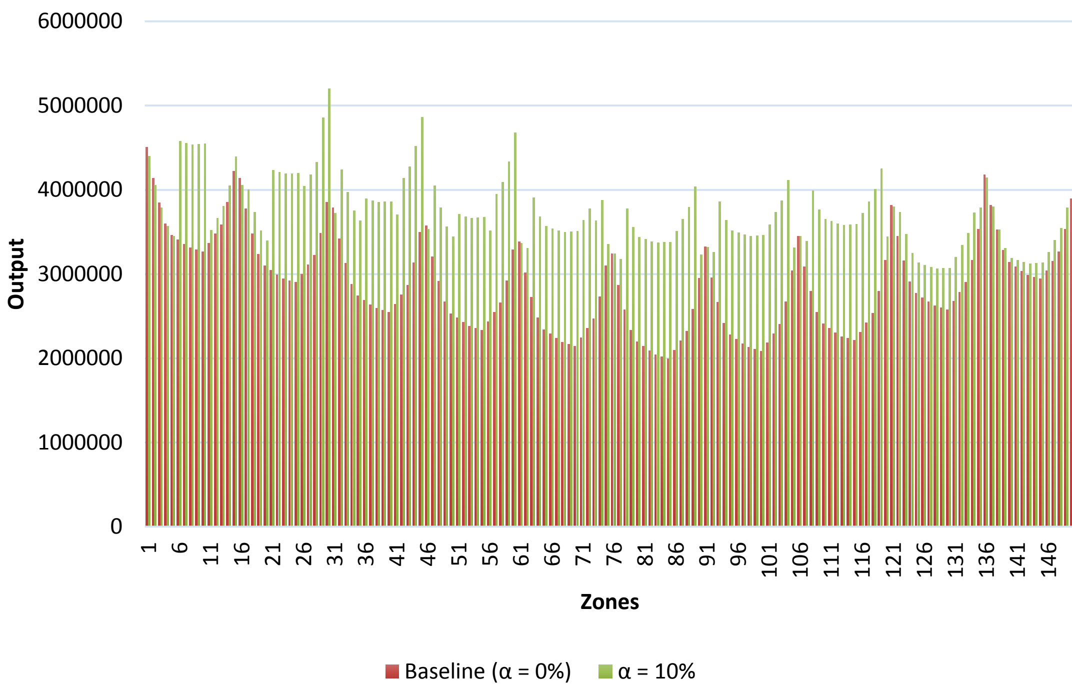


Fig. 10. Baseline Energy vs. Energy with User Preference α = 0.1 (Model #1)

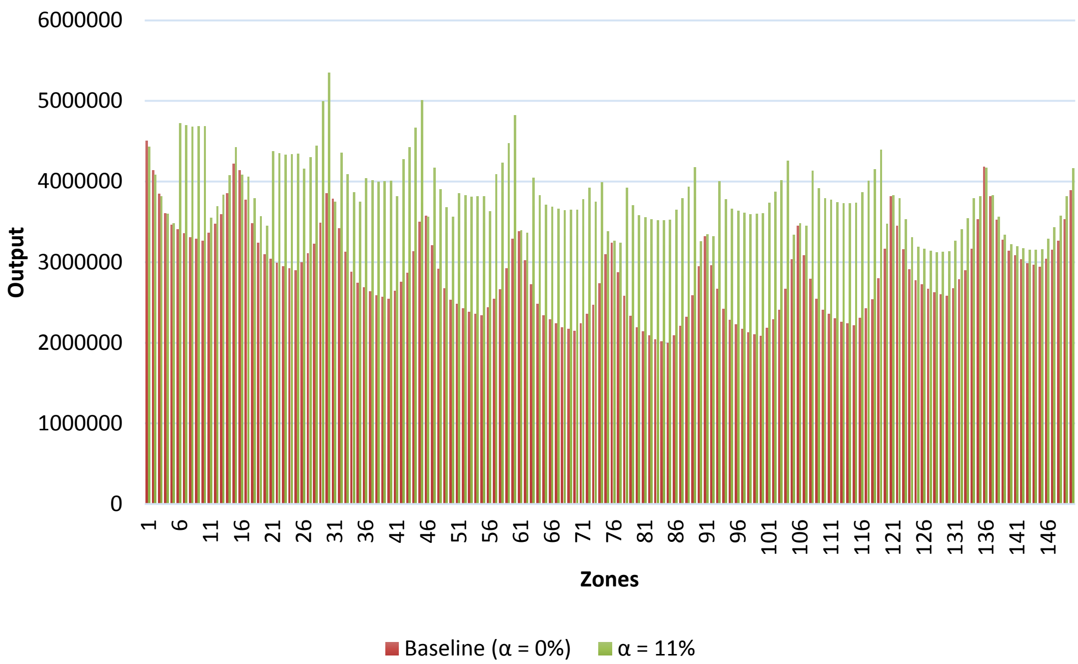
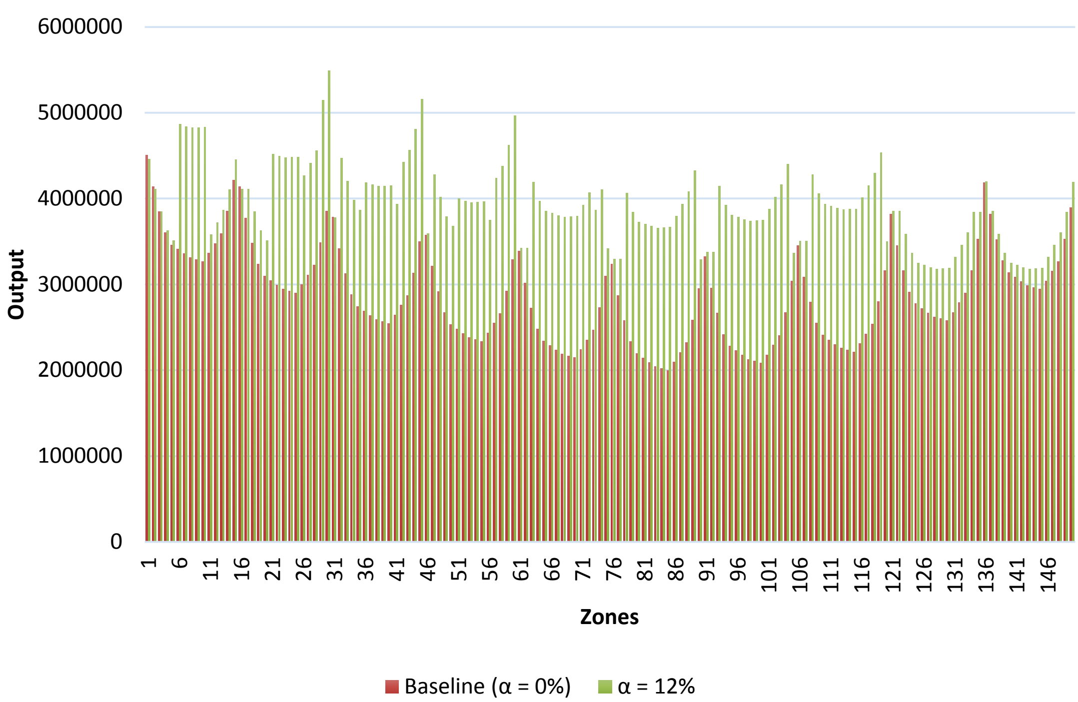


Fig. 11. Baseline Energy vs. Energy with User Preference α = 0.11 (Model #1)



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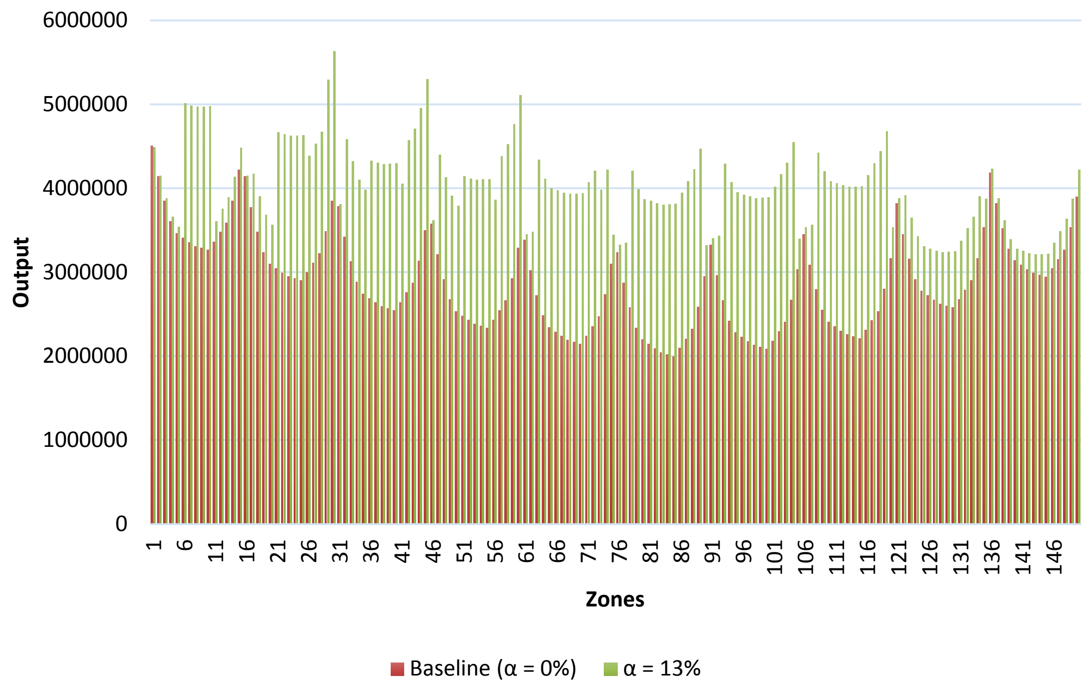
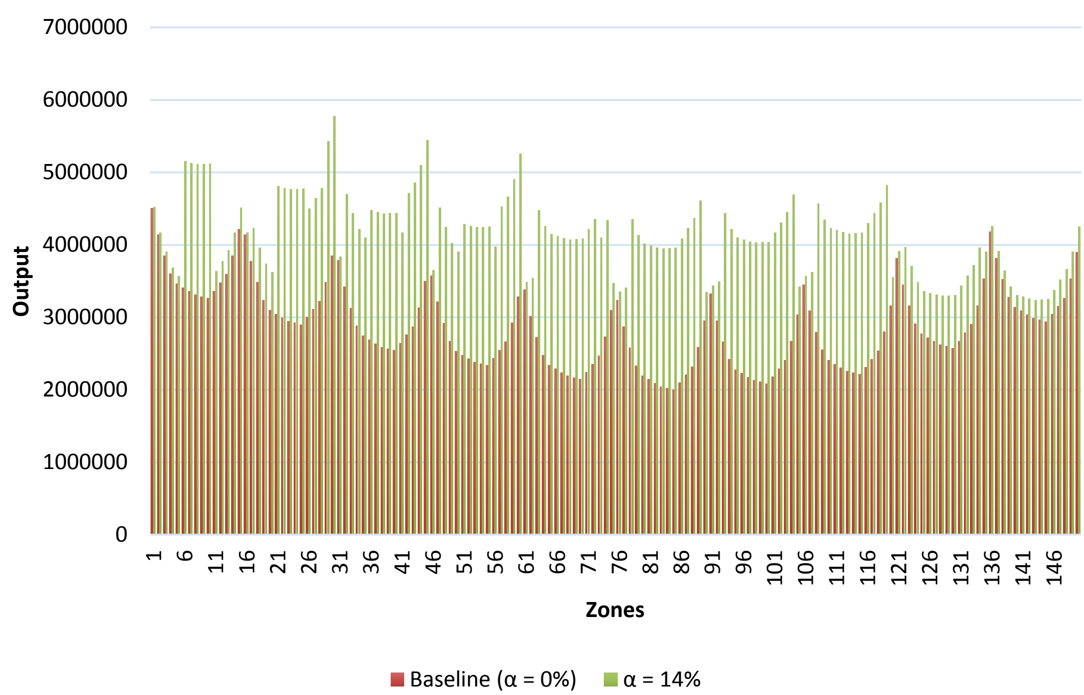


Fig. 13. Baseline Energy vs. Energy with User Preference α = 0.13 (Model #1)



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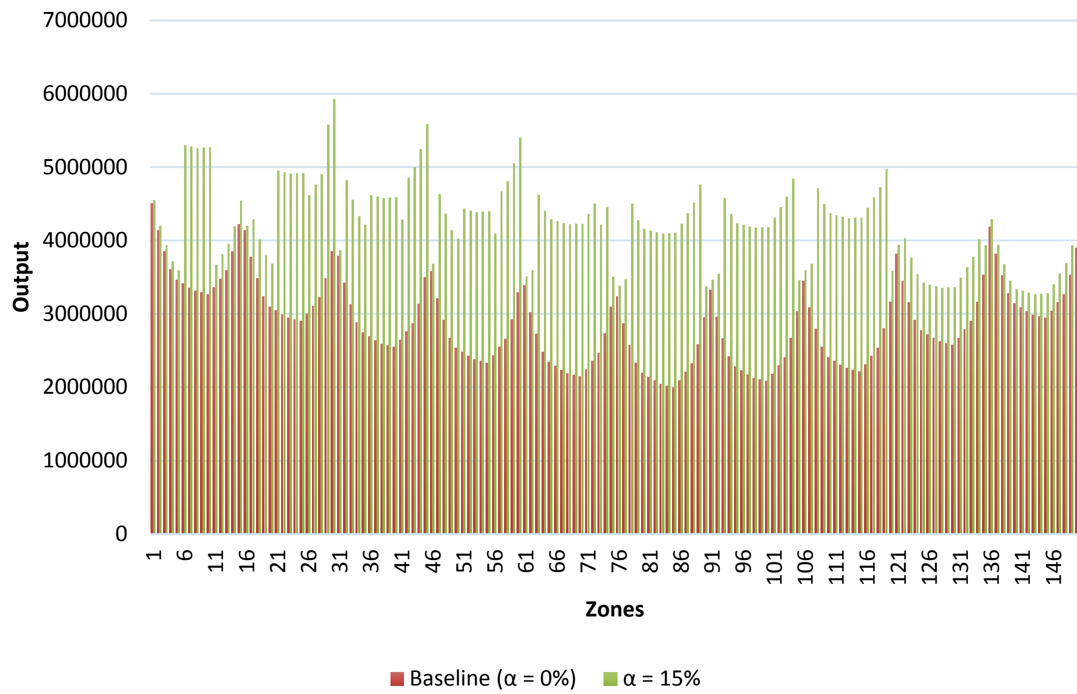

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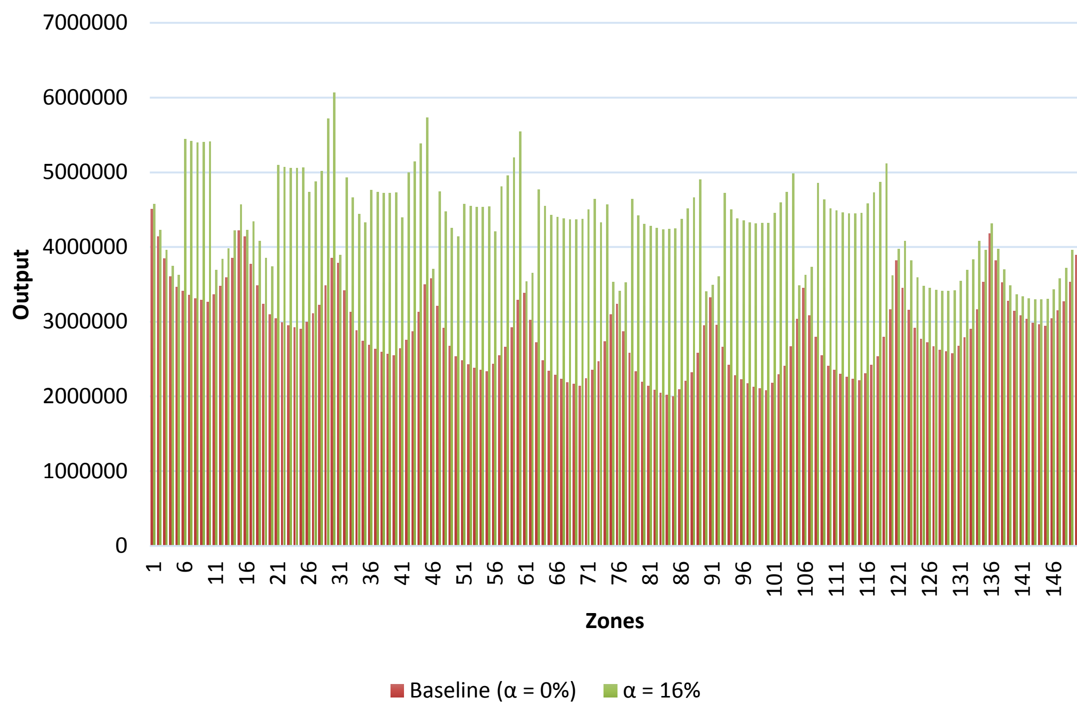

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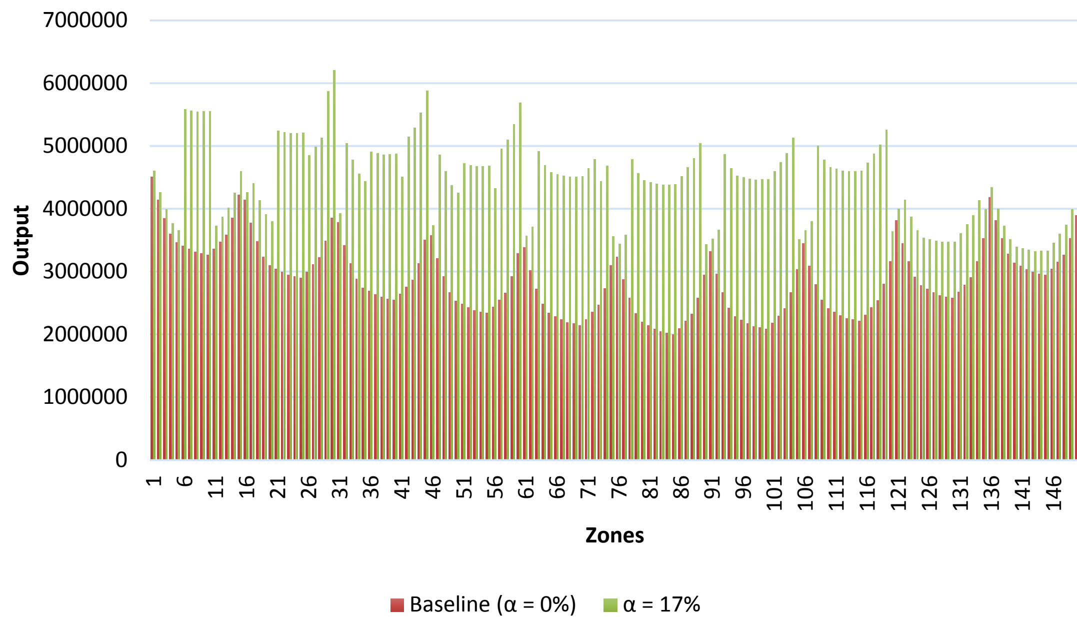


Fig. 17. Baseline Energy vs. Energy with User Preference α = 0.17 (Model #1)

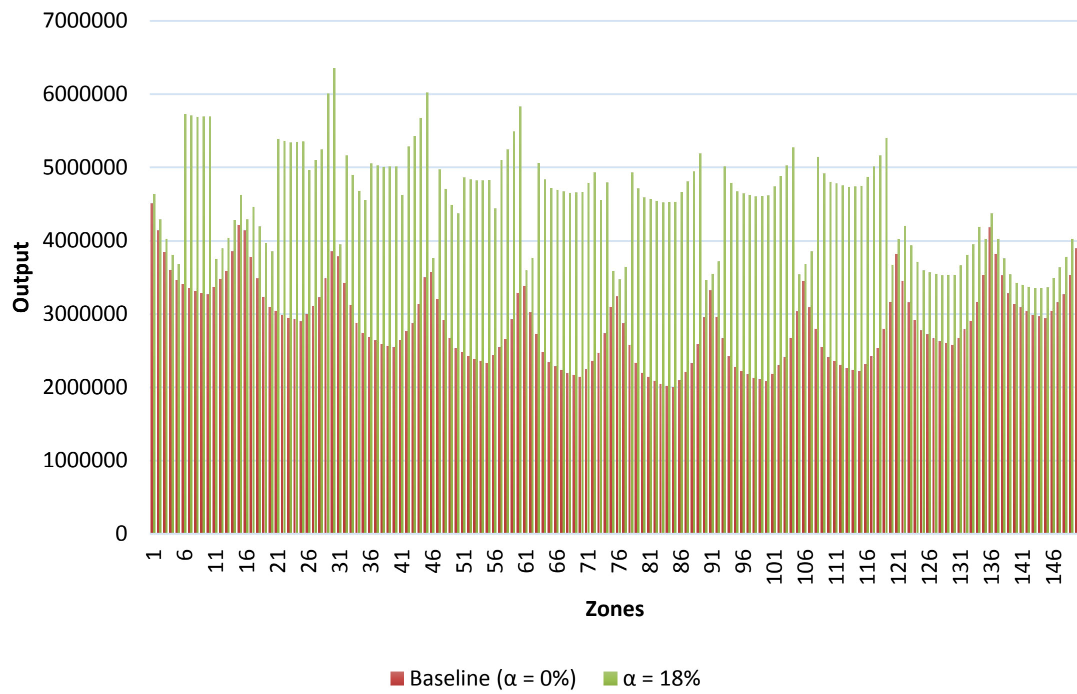


Fig. 18. Baseline Energy vs. Energy with User Preference α = 0.18 (Model #1)

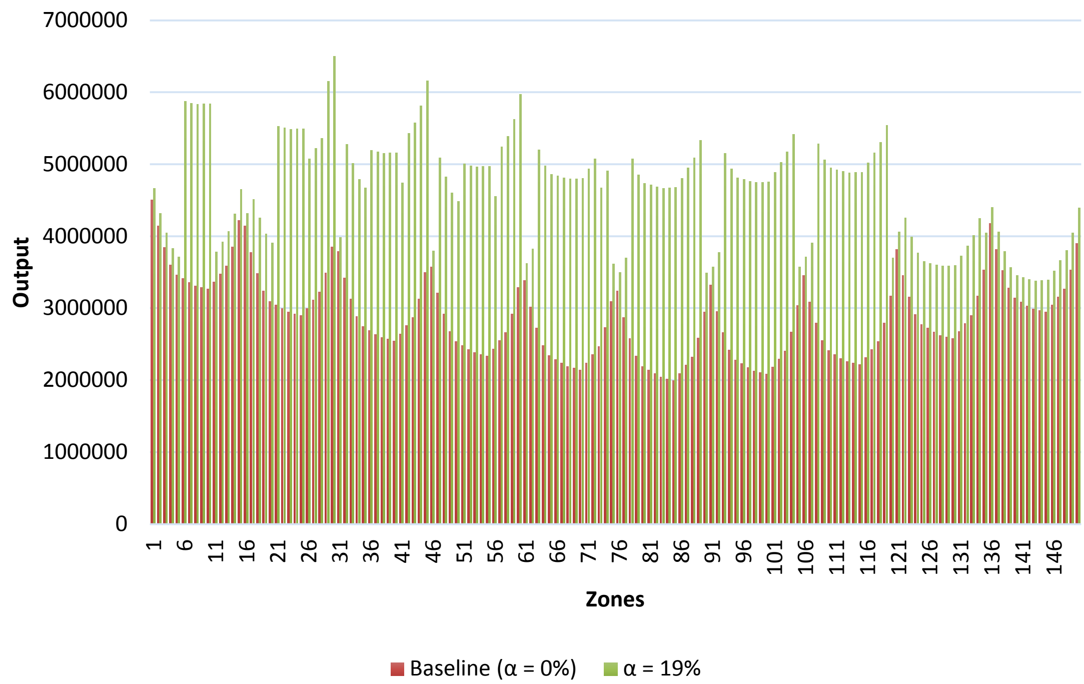


Fig. 19. Baseline Energy vs. Energy with User Preference α = 0.19 (Model #1)

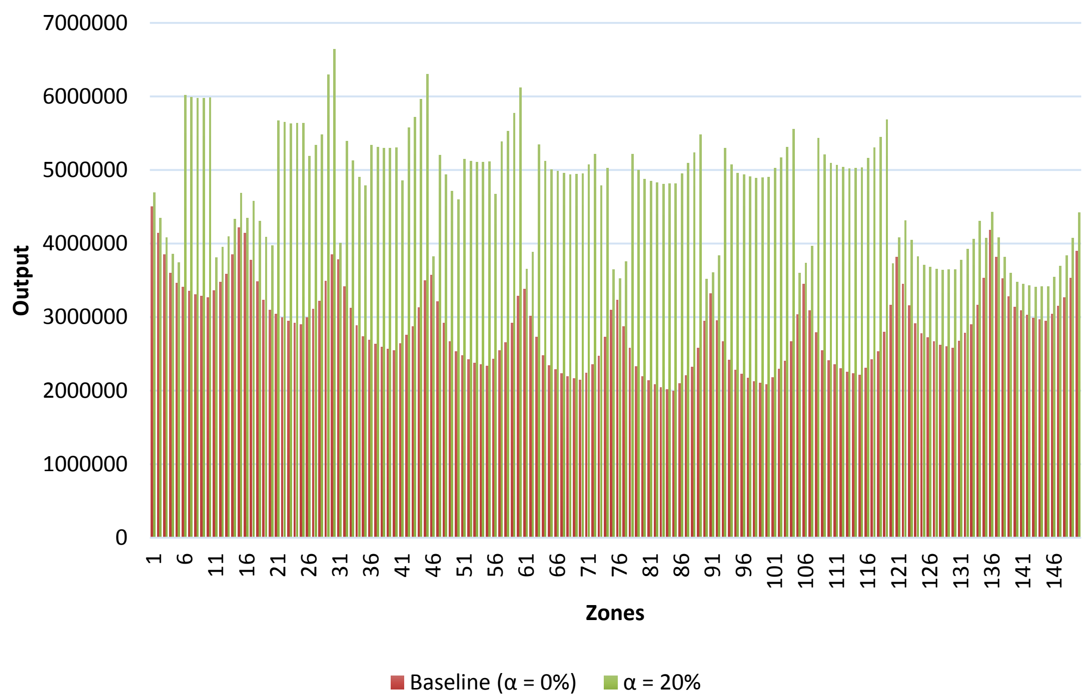


Fig. 20. Baseline Energy vs. Energy with User Preference α = 0.20 (Model #1)

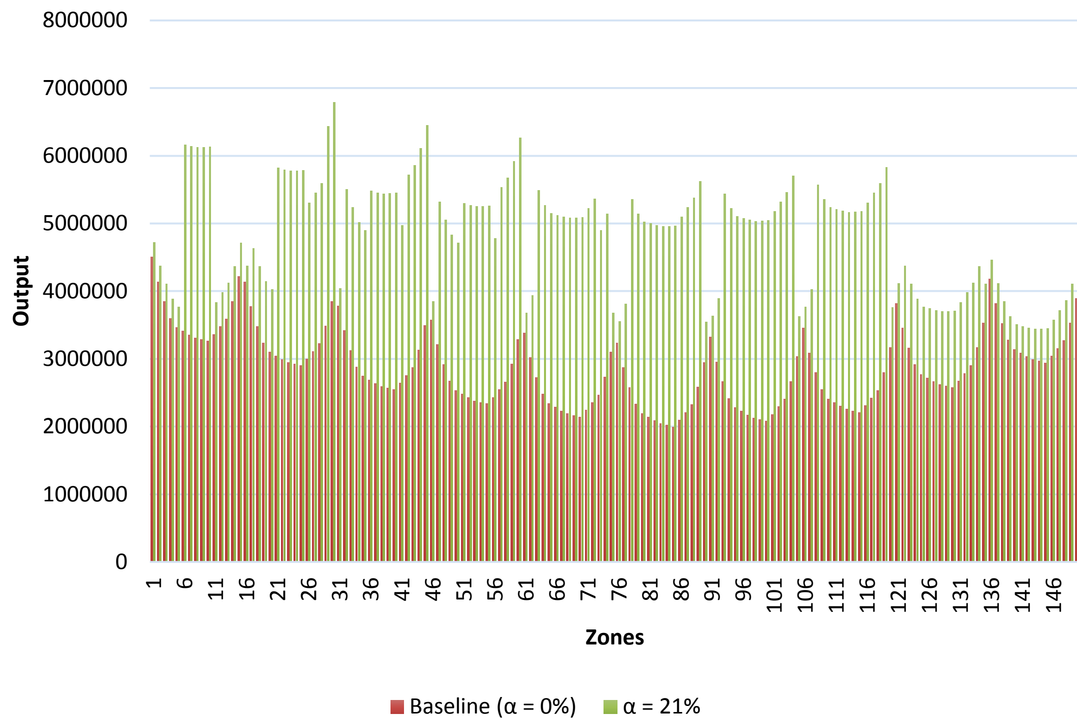
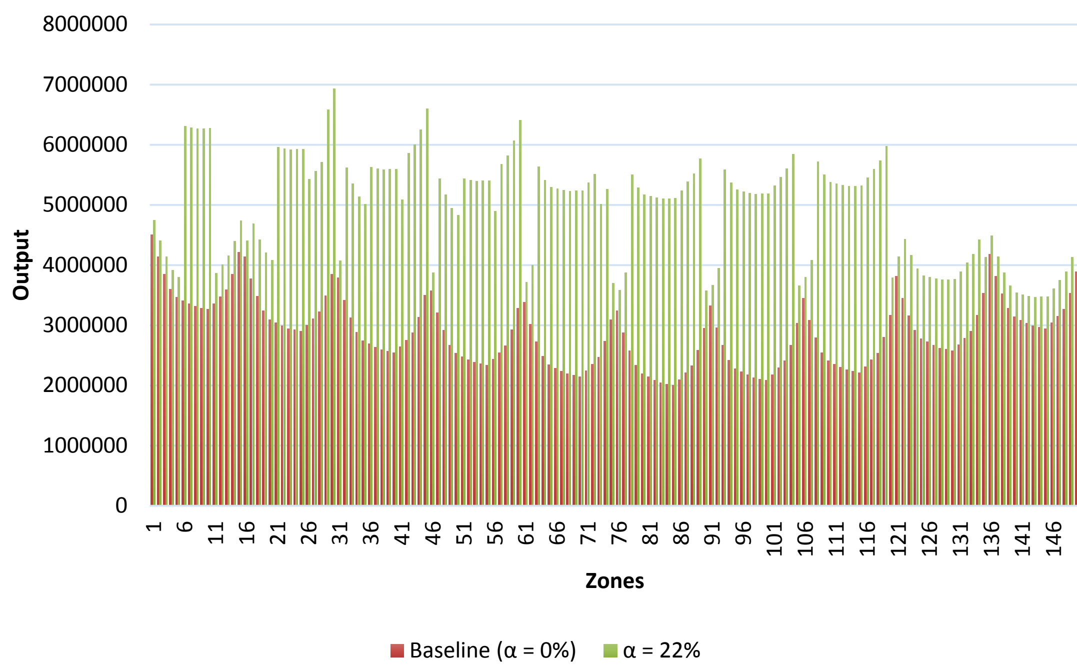


Fig. 21. Baseline Energy vs. Energy with User Preference α = 0.21 (Model #1)



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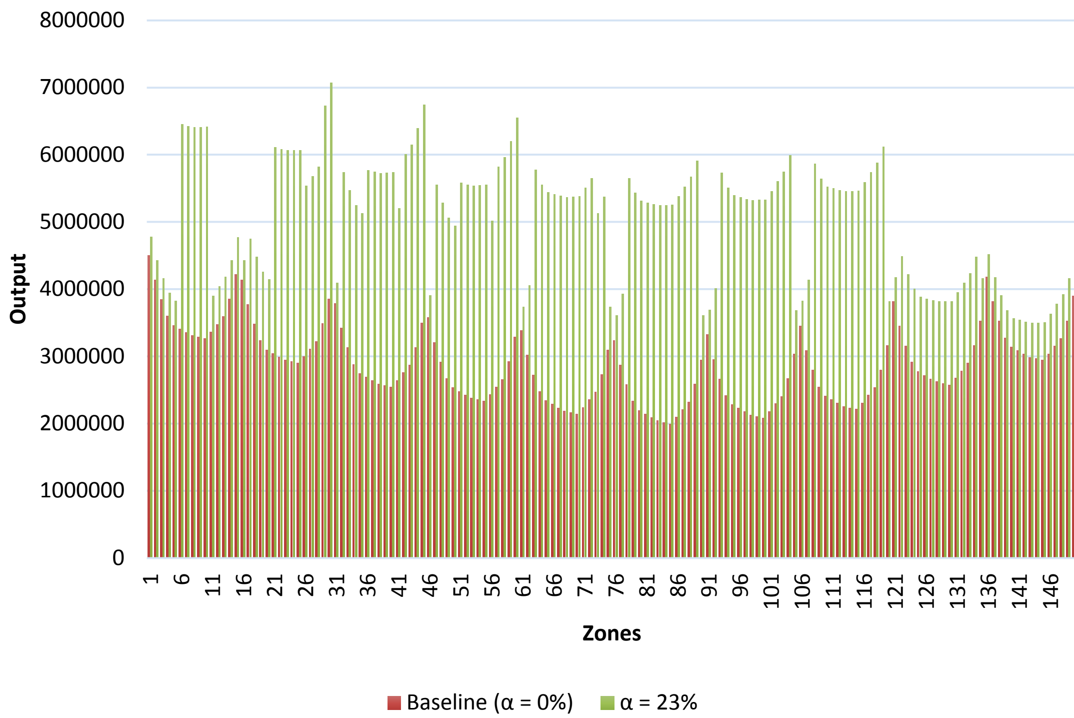

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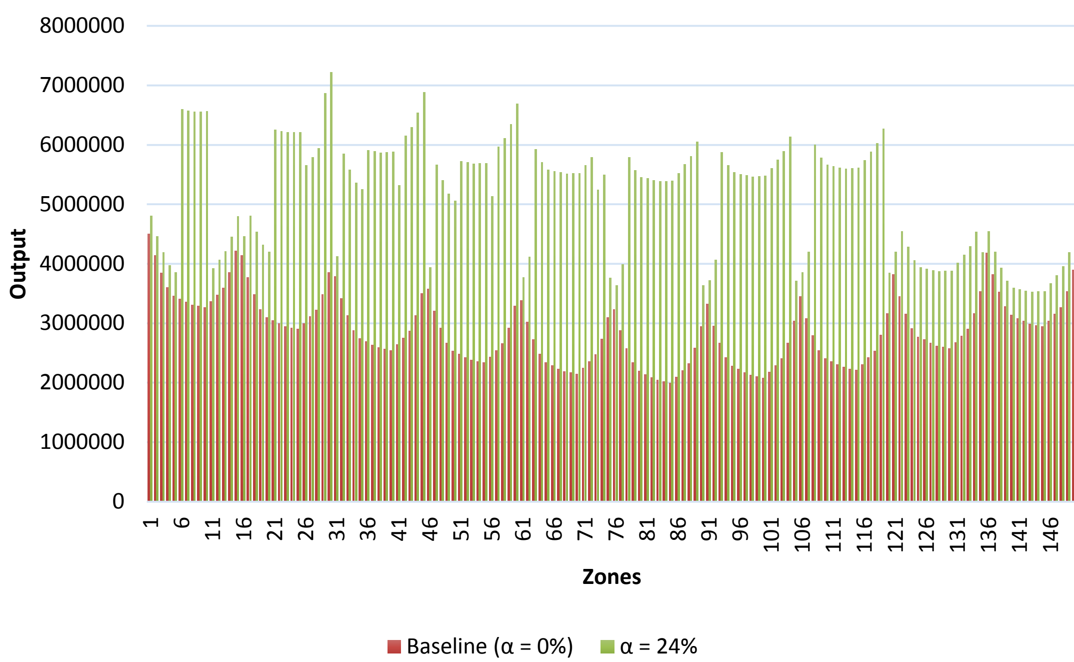


Fig. 24. Baseline Energy vs. Energy with User Preference α = 0.24 (Model #1)

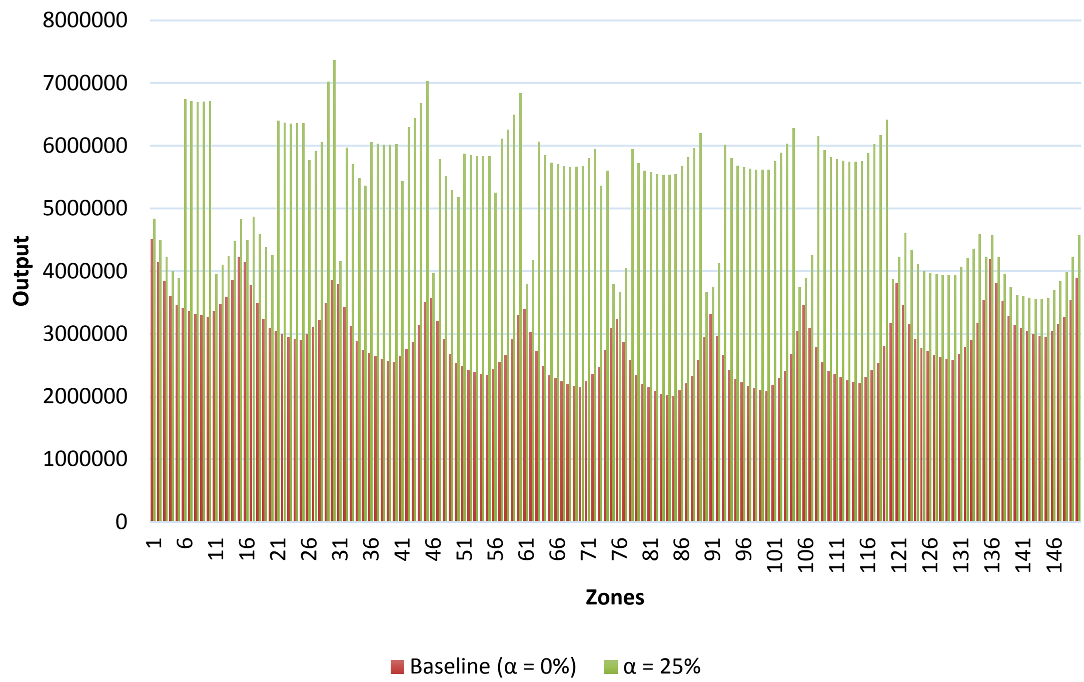


Fig. 25. Baseline Energy vs. Energy with User Preference α = 0.25 (Model #1)

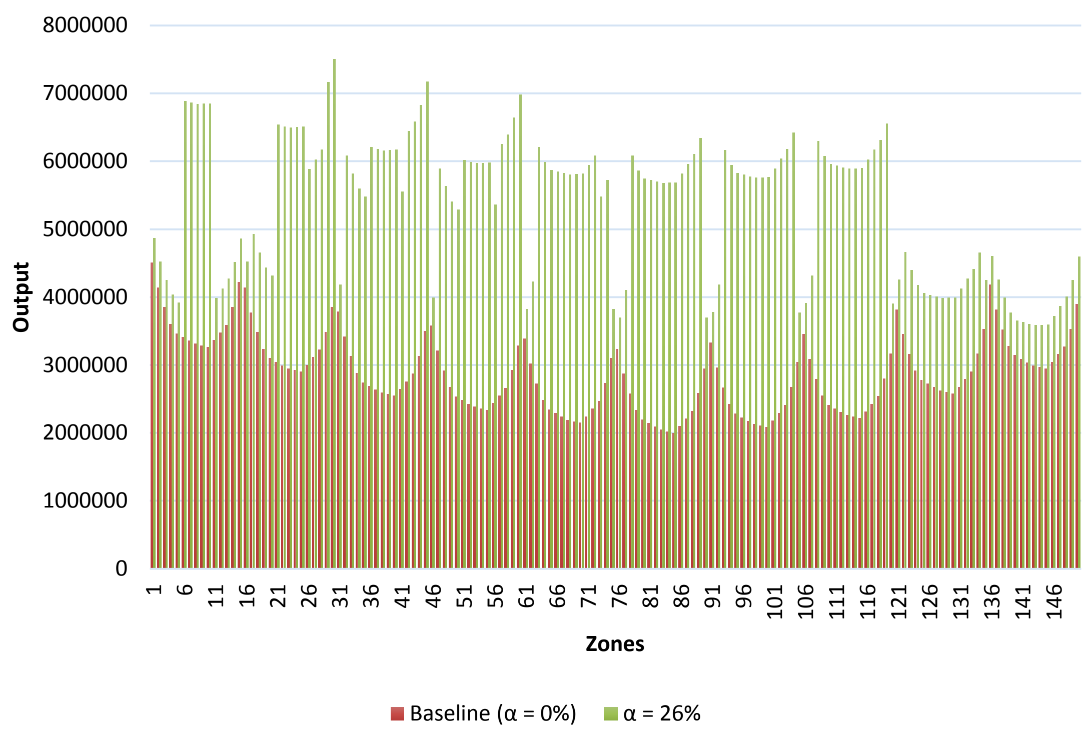


Fig. 26. Baseline Energy vs. Energy with User Preference α = 0.26 (Model #1)

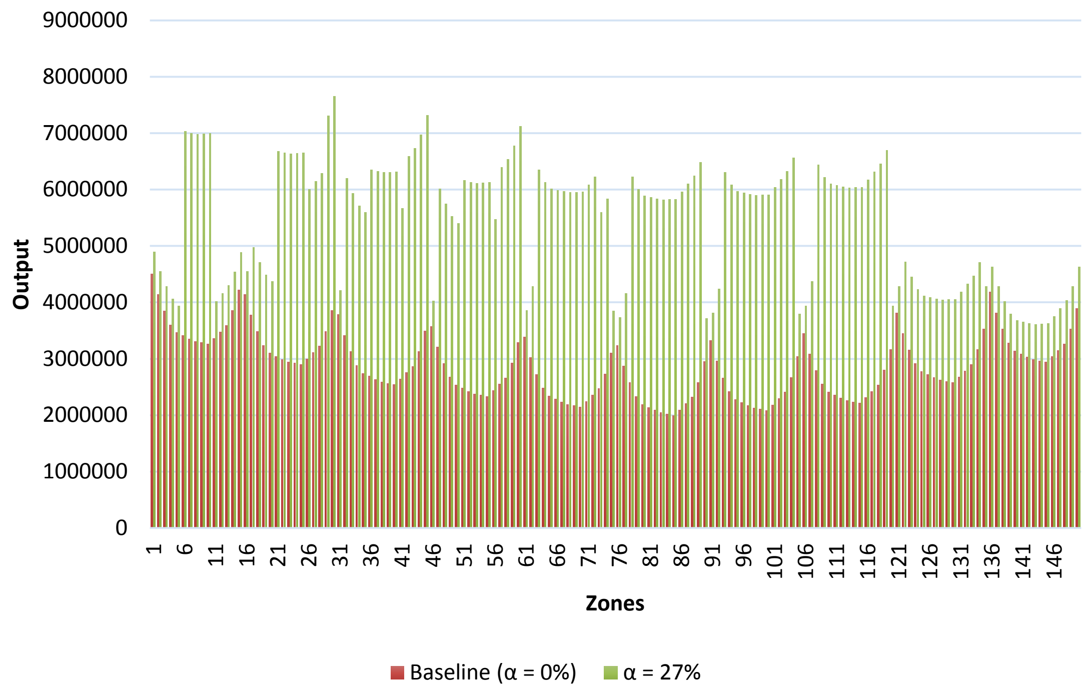
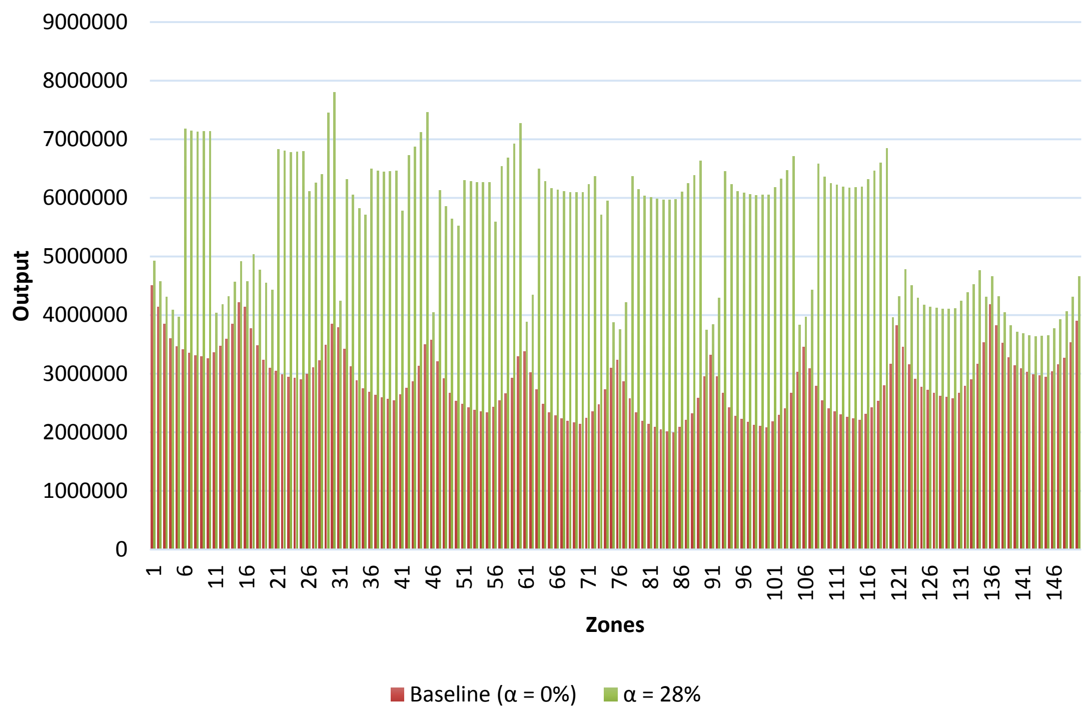


Fig. 27. Baseline Energy vs. Energy with User Preference α = 0.28 (Model #1)



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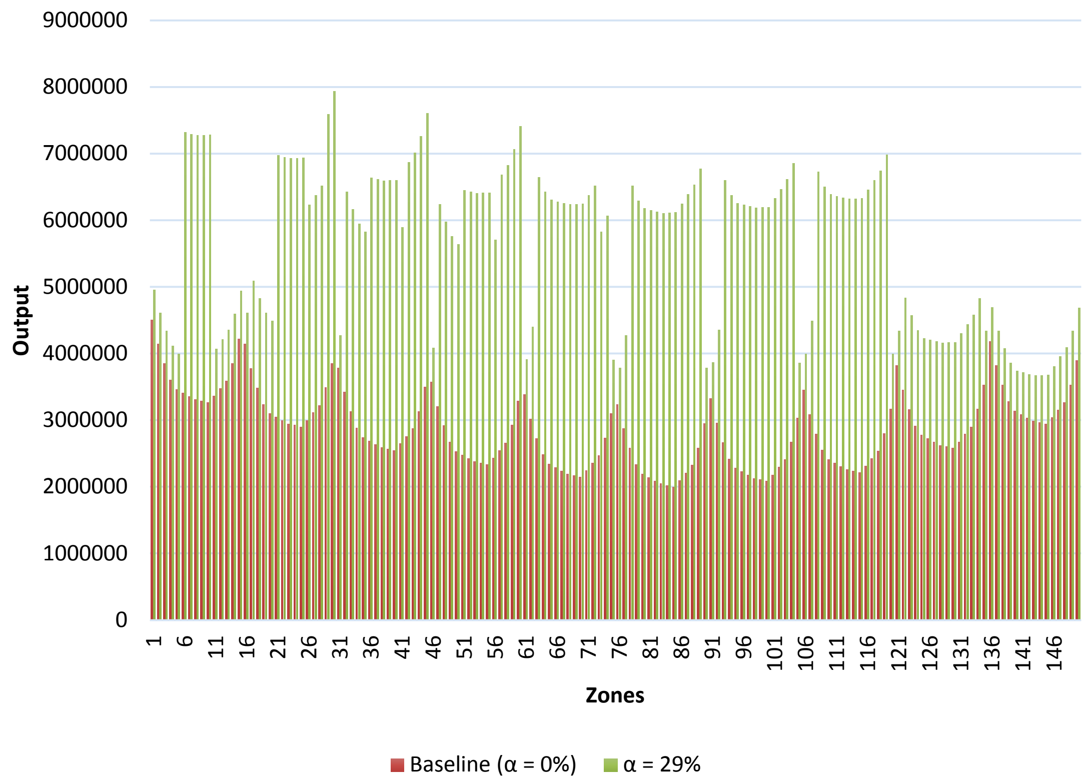

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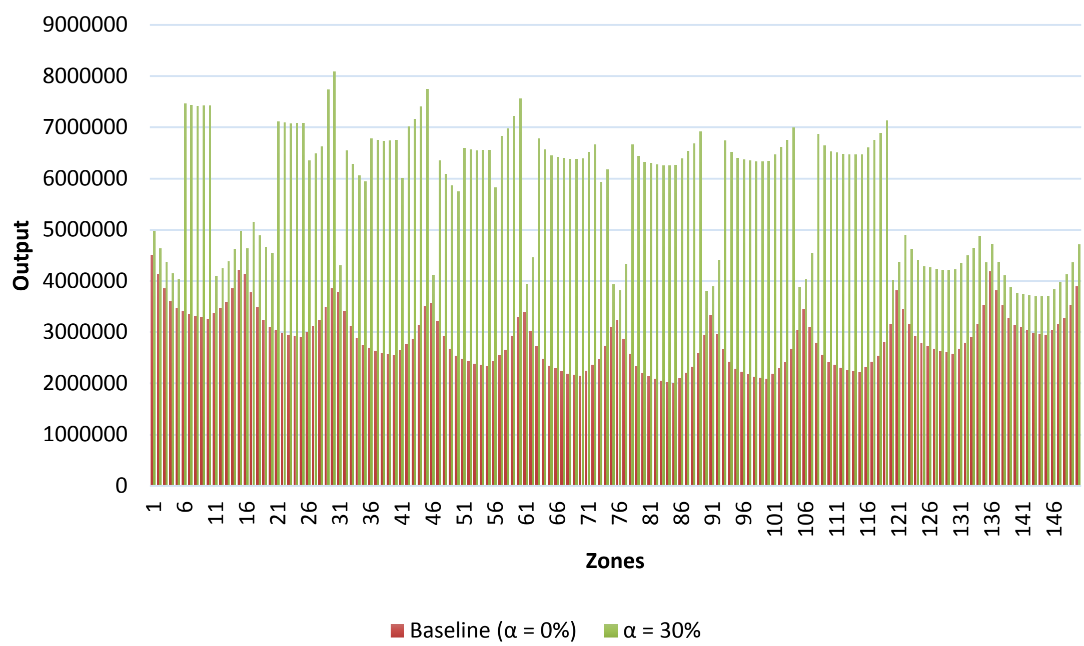

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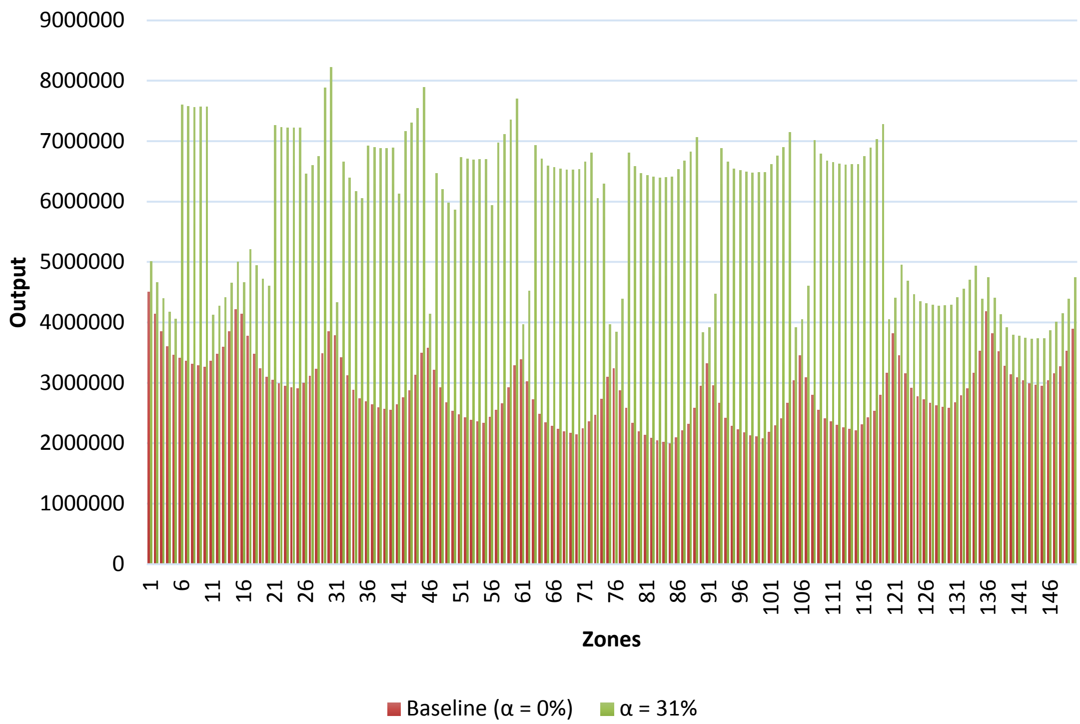

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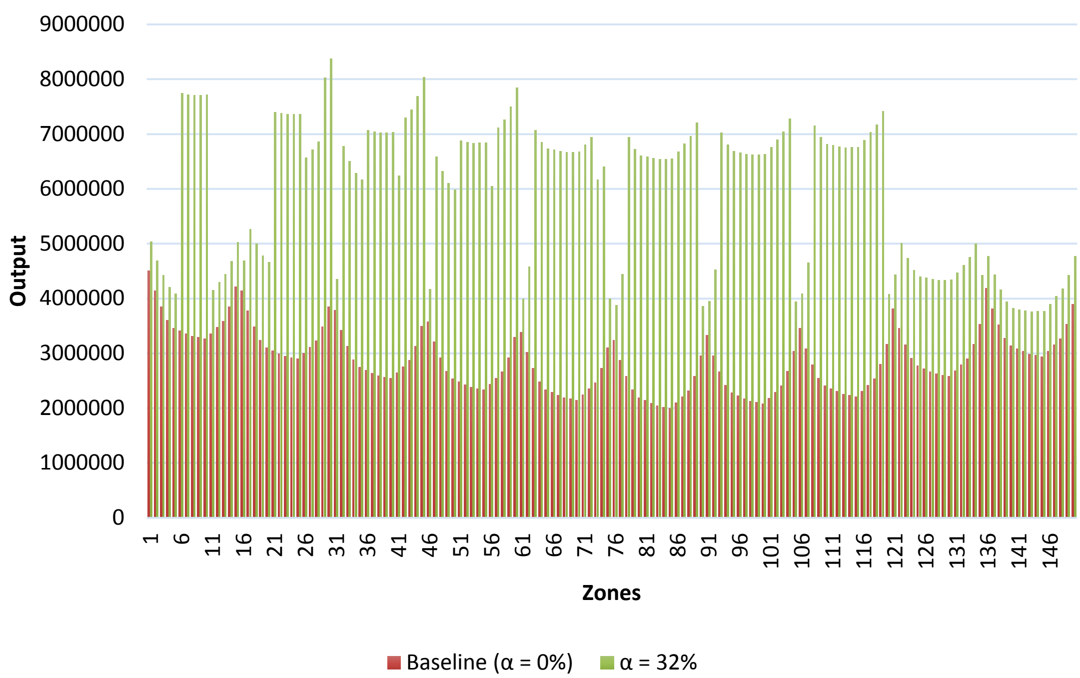

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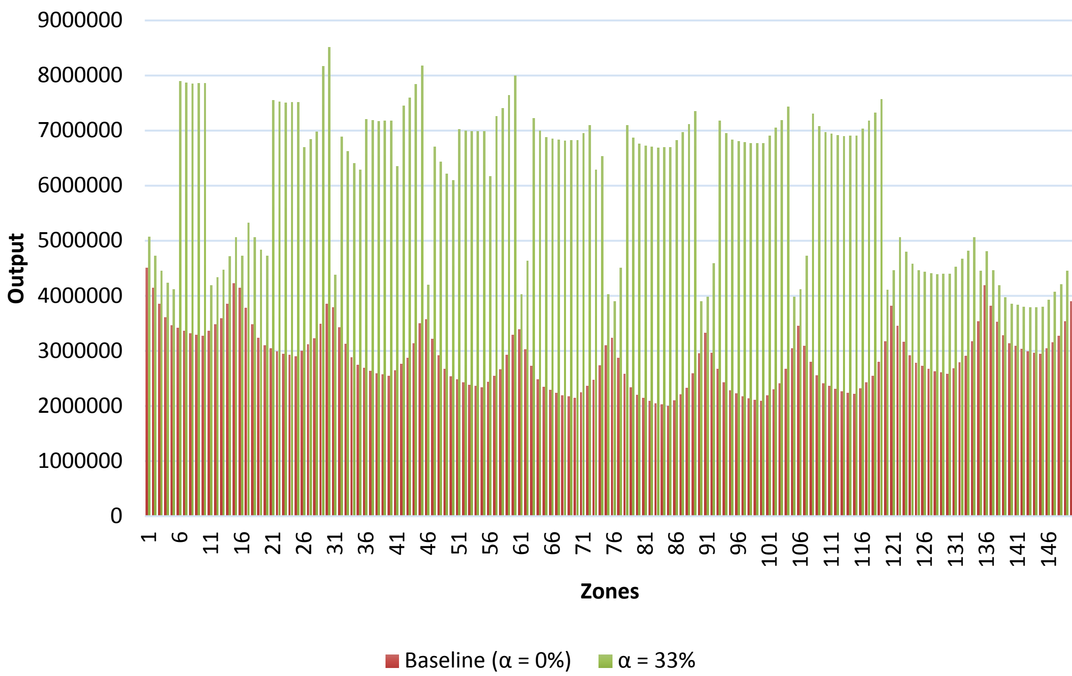

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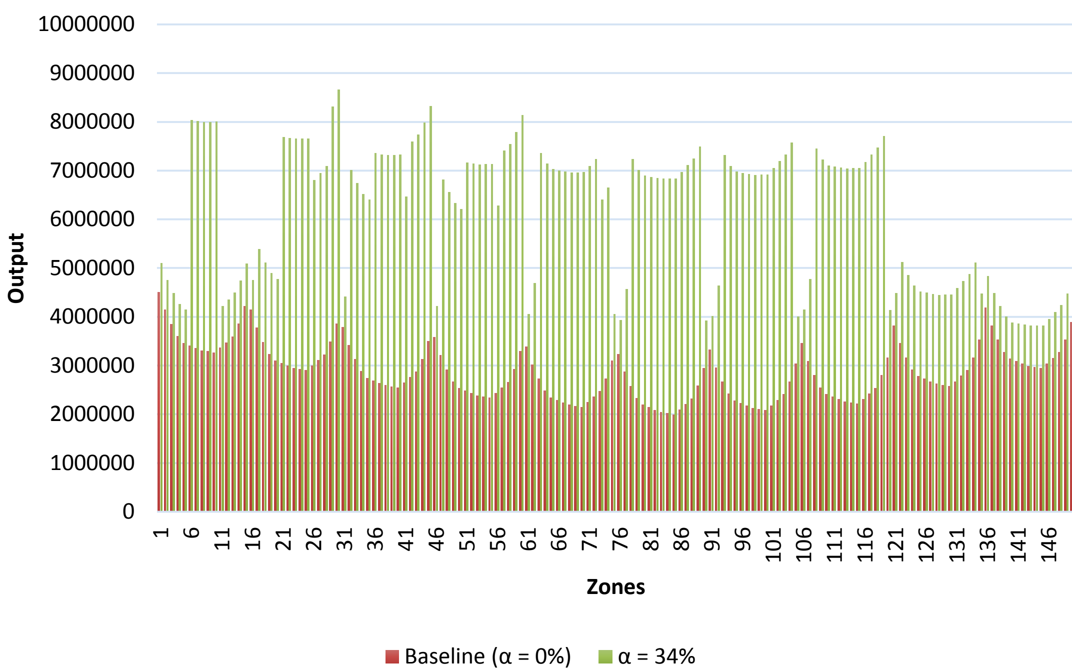
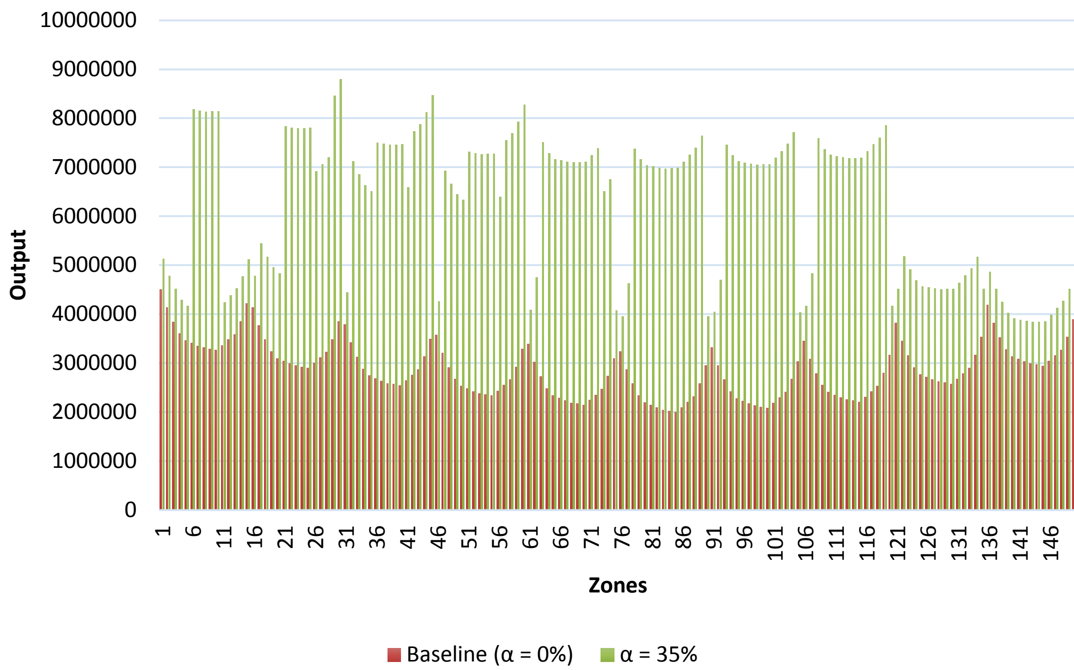


Fig. 34. Baseline Energy vs. Energy with User Preference α = 0.34 (Model #1)



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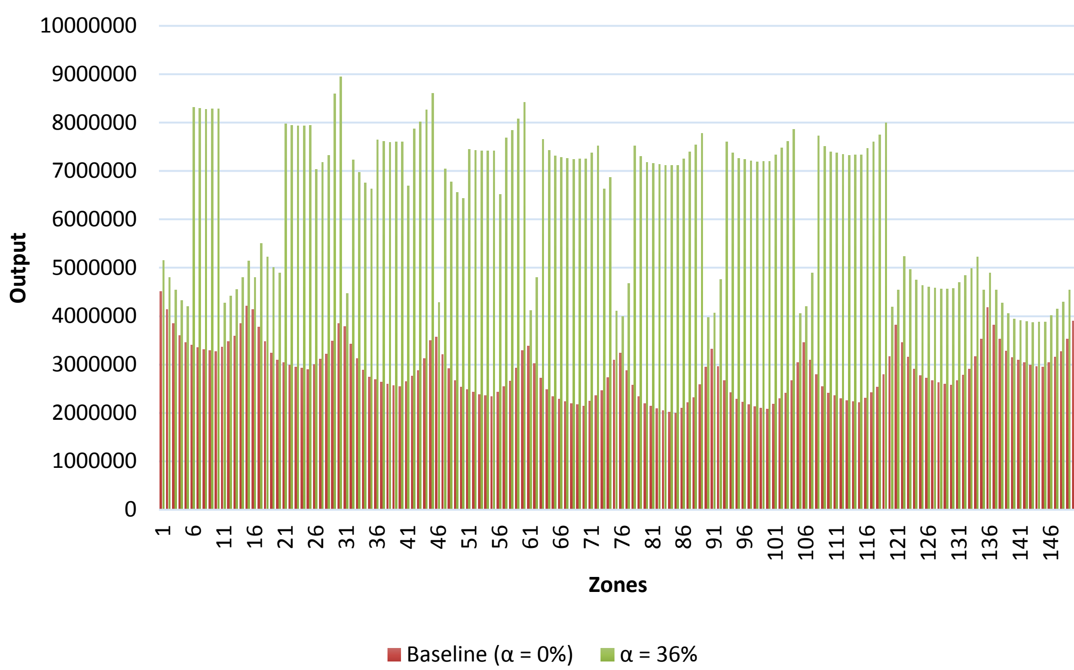

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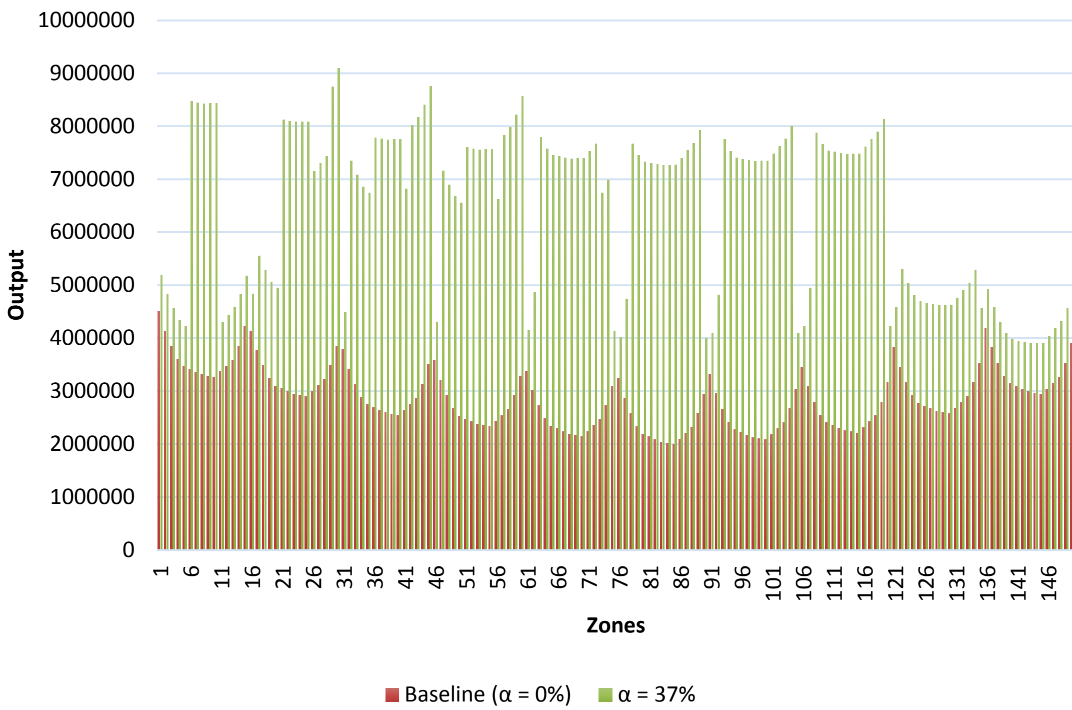

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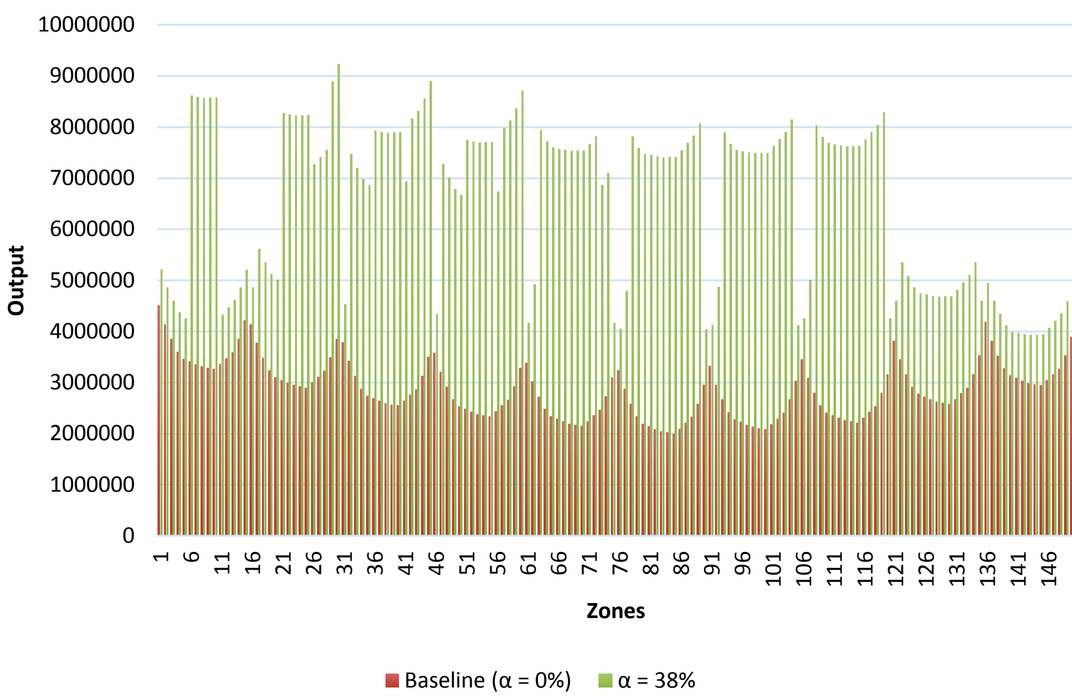


Fig. 38. Baseline Energy vs. Energy with User Preference α = 0.38 (Model #1)

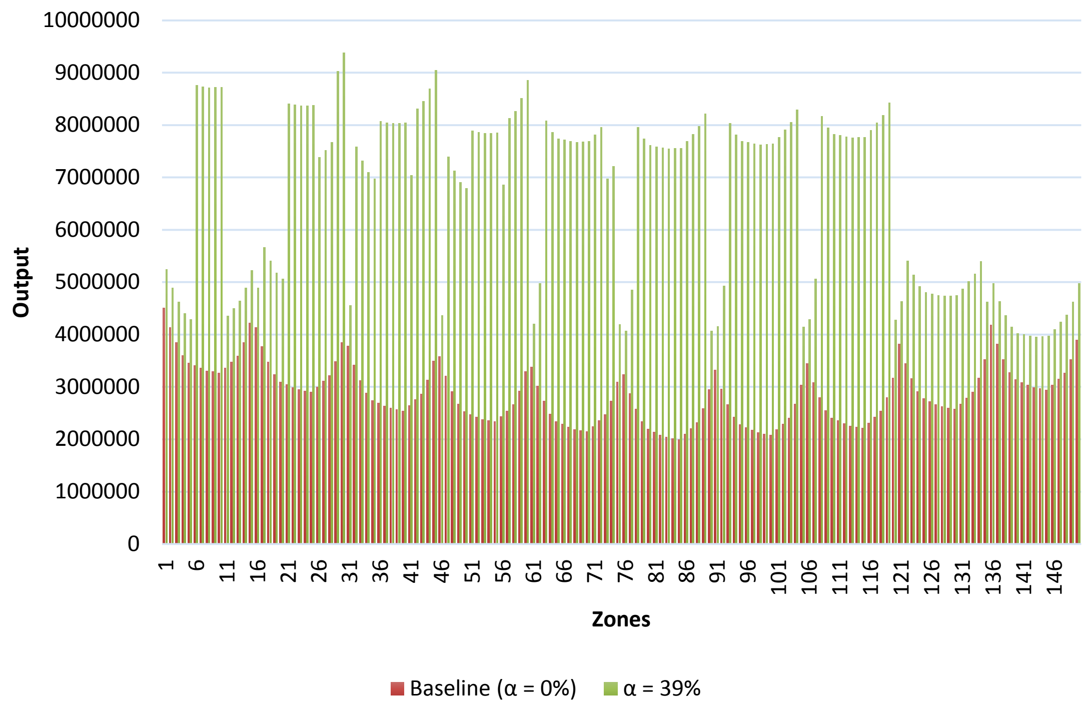


Fig. 39. Baseline Energy vs. Energy with User Preference α = 0.39 (Model #1)

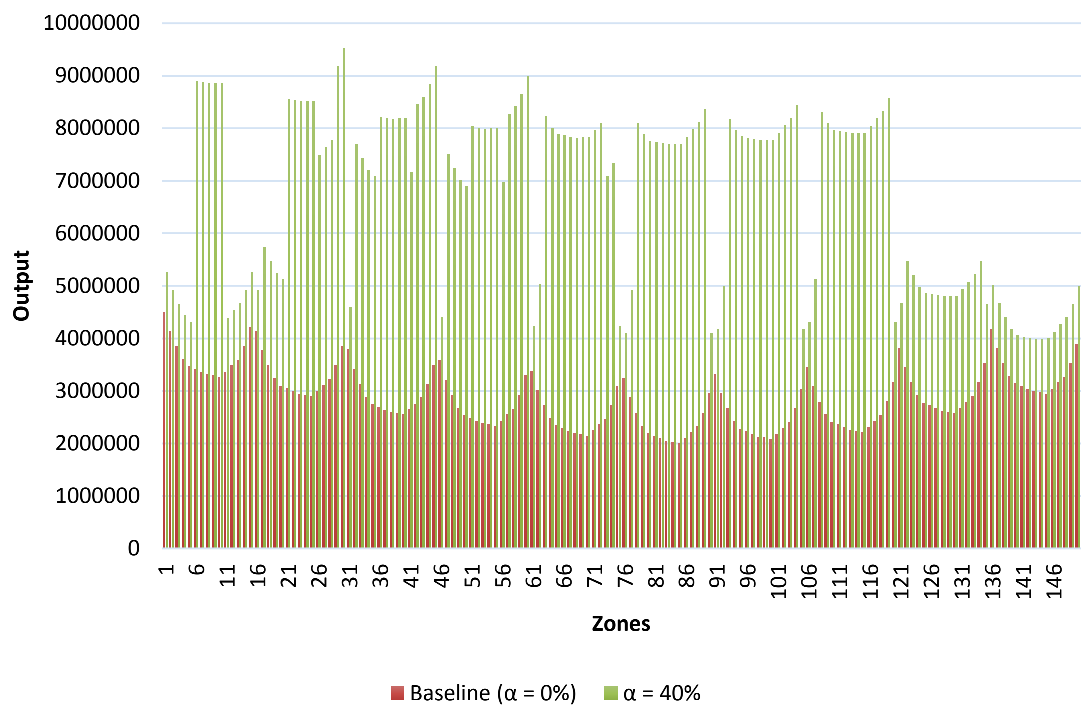


Fig. 40. Baseline Energy vs. Energy with User Preference α = 0.40 (Model #1)

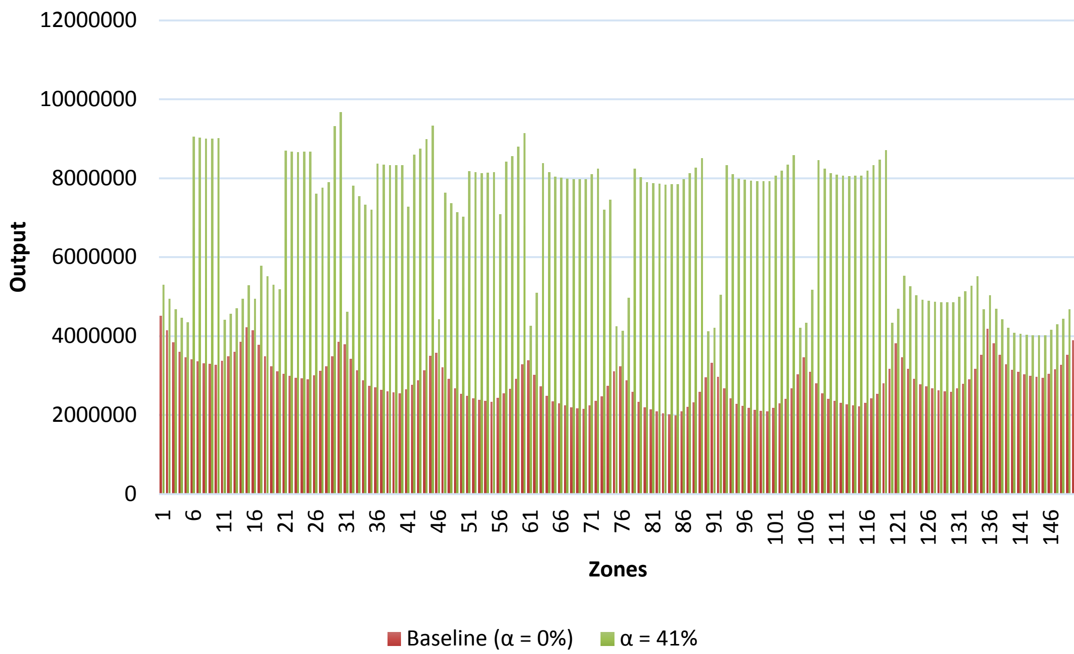


Fig. 41. Baseline Energy vs. Energy with User Preference α = 0.41 (Model #1)

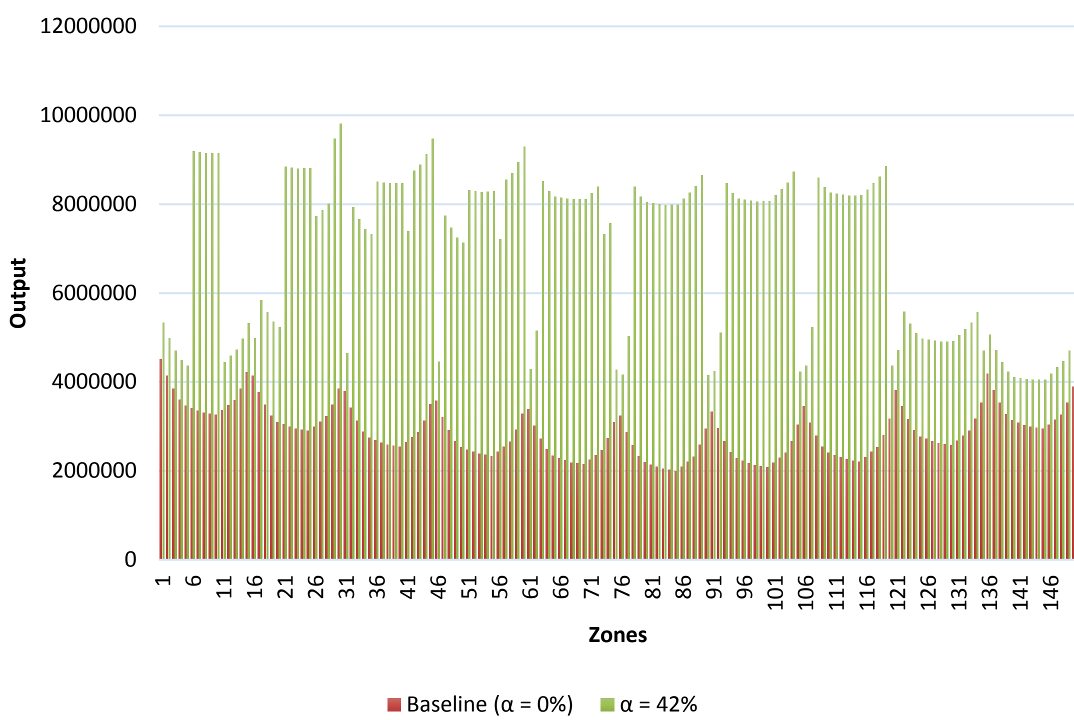


Fig. 42. Baseline Energy vs. Energy with User Preference α = 0.42 (Model #1)

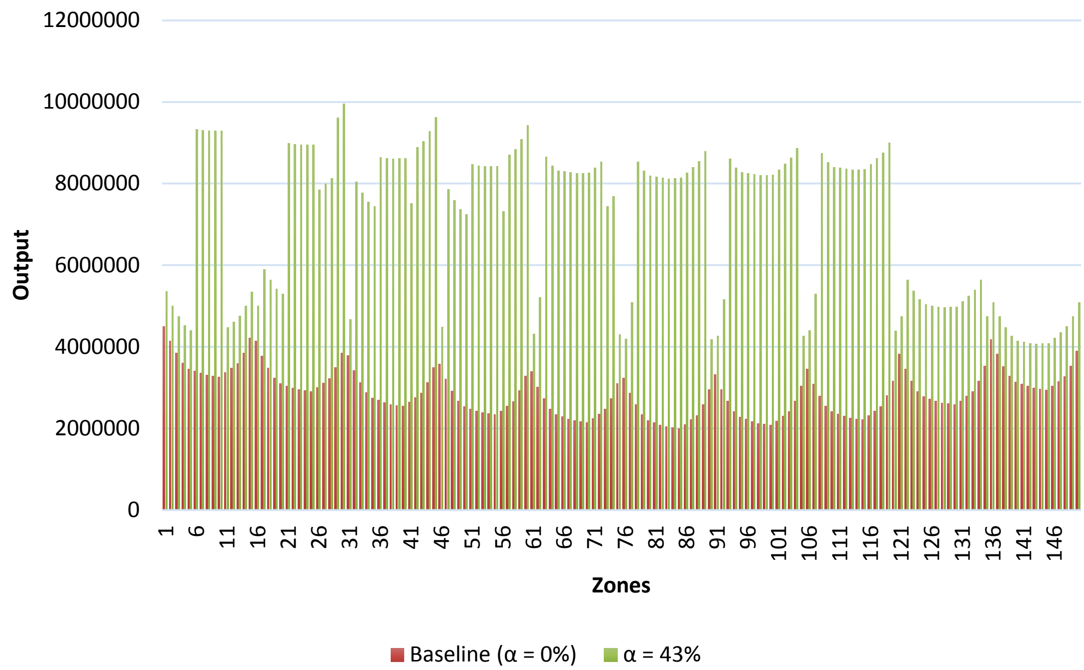


Fig. 43. Baseline Energy vs. Energy with User Preference α = 0.43 (Model #1)

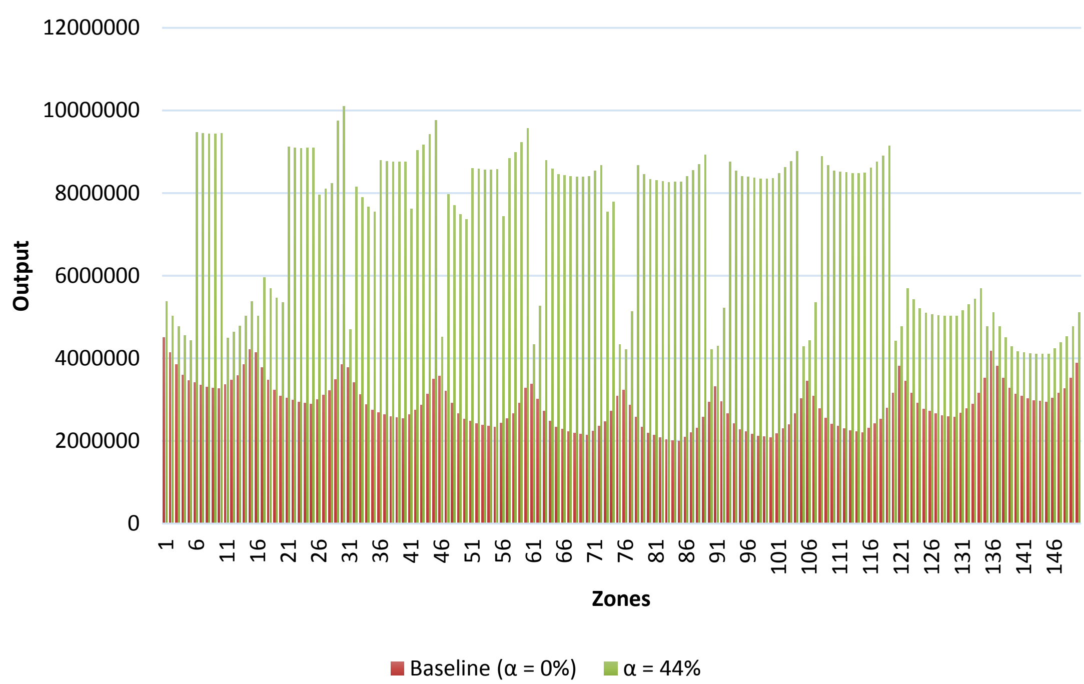


Fig. 44. Baseline Energy vs. Energy with User Preference α = 0.44 (Model #1)

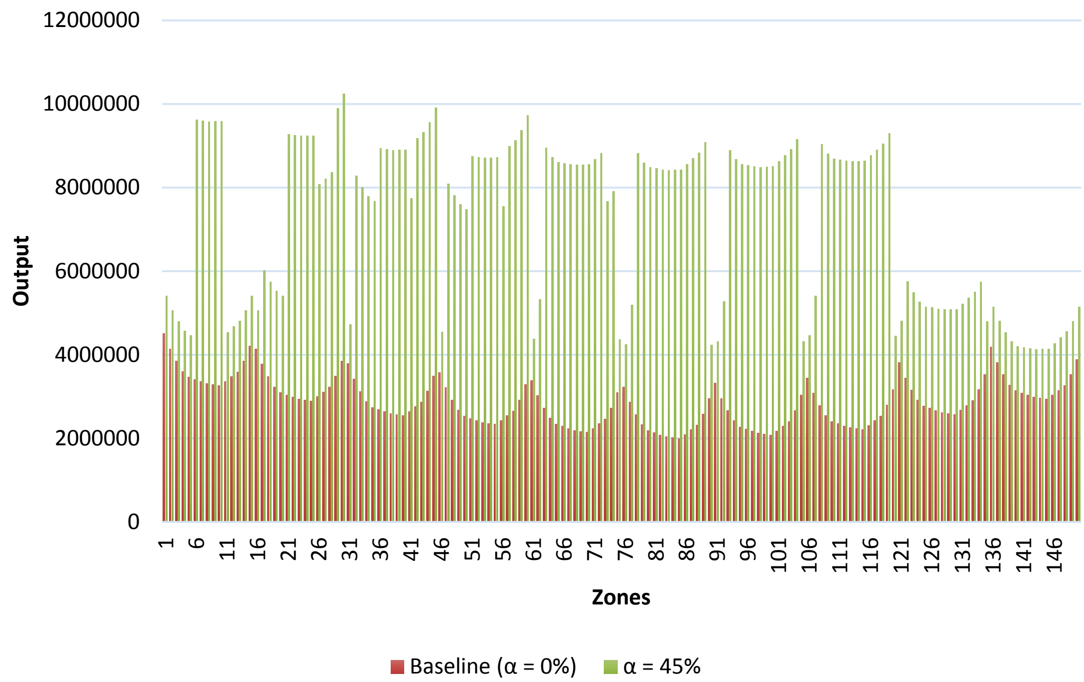


Fig. 45. Baseline Energy vs. Energy with User Preference α = 0.45 (Model #1)

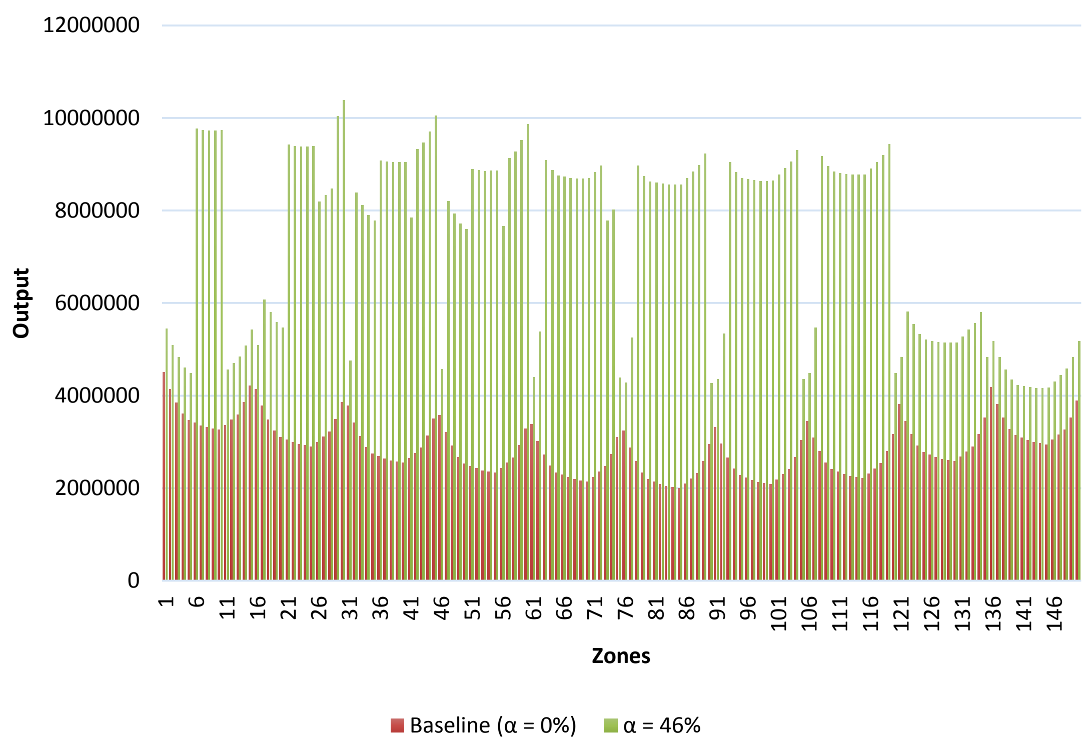


Fig. 46. Baseline Energy vs. Energy with User Preference α = 0.46 (Model #1)

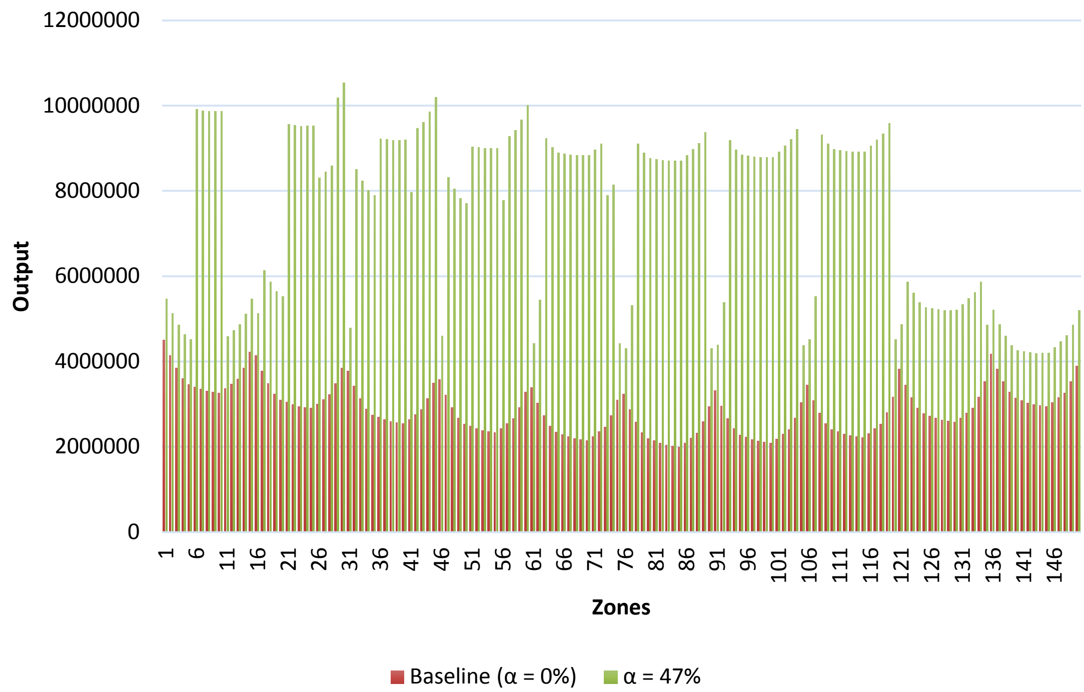


Fig. 47. Baseline Energy vs. Energy with User Preference α = 0.47 (Model #1)

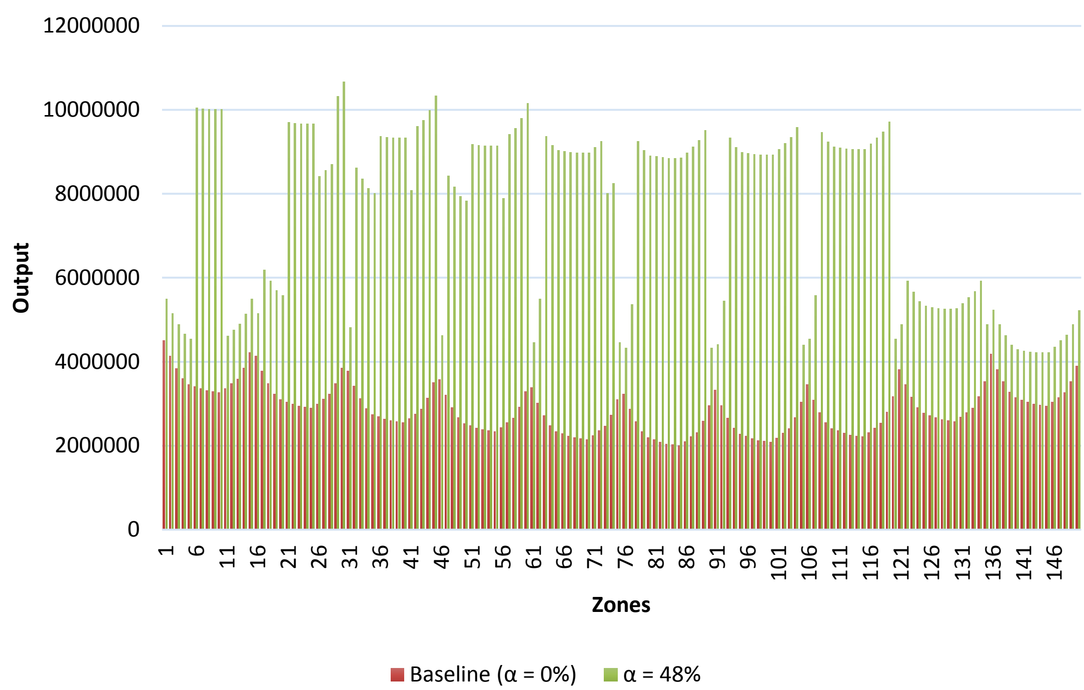


Fig. 48. Baseline Energy vs. Energy with User Preference α = 0.48 (Model #1)

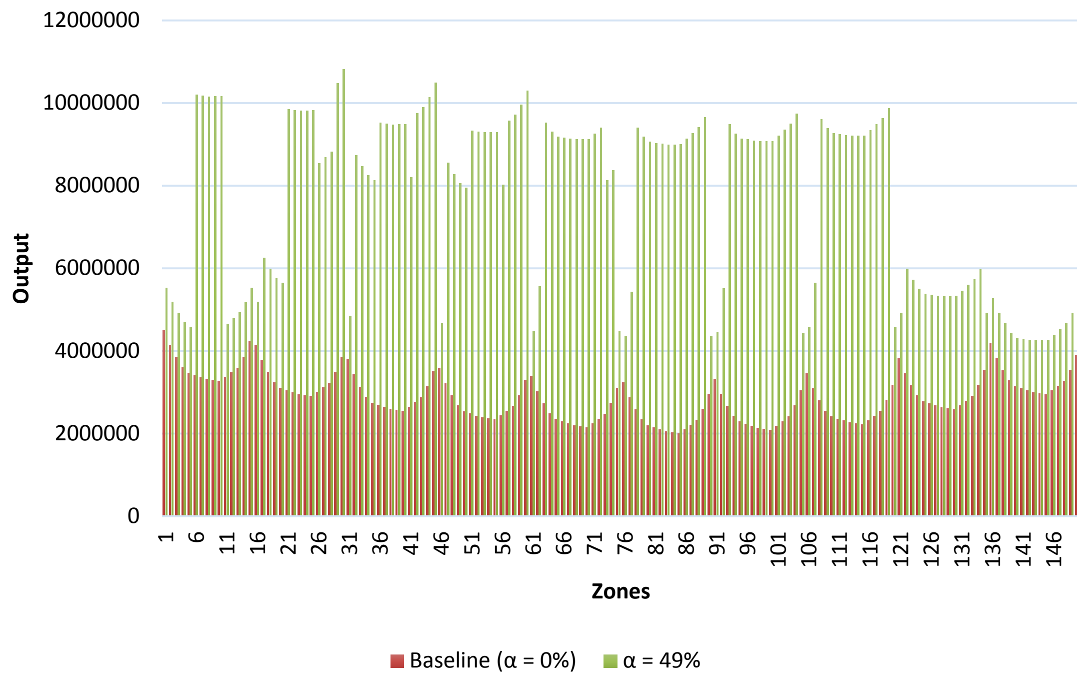


Fig. 49. Baseline Energy vs. Energy with User Preference α = 0.49 (Model #1)

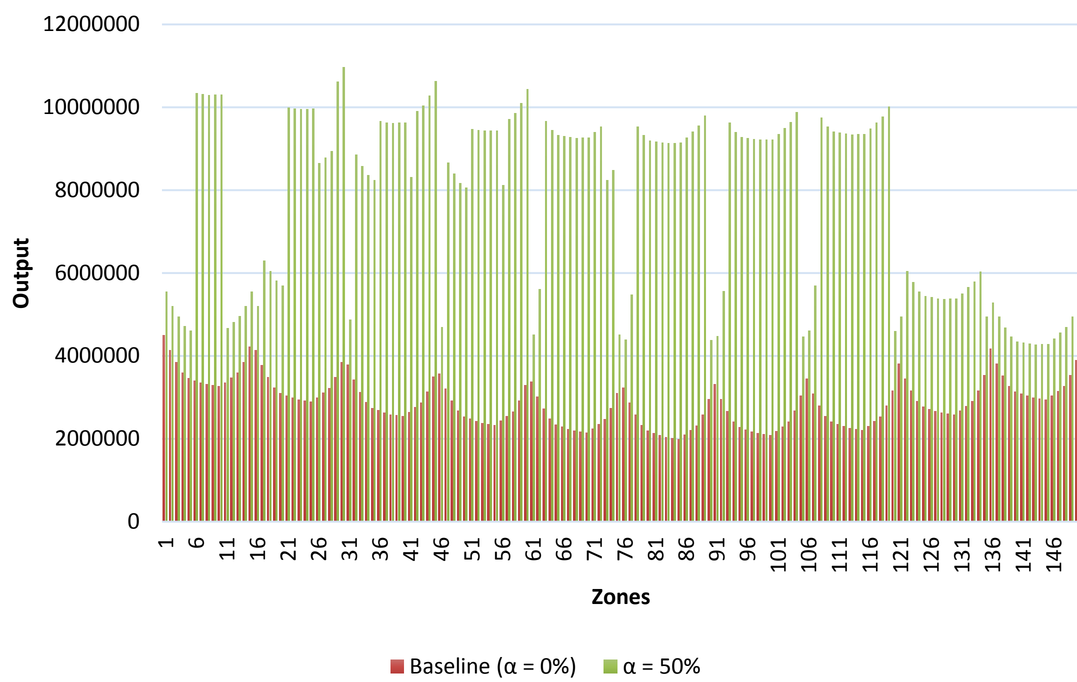


Fig. 50. Baseline Energy vs. Energy with User Preference α = 0.50 (Model #1)

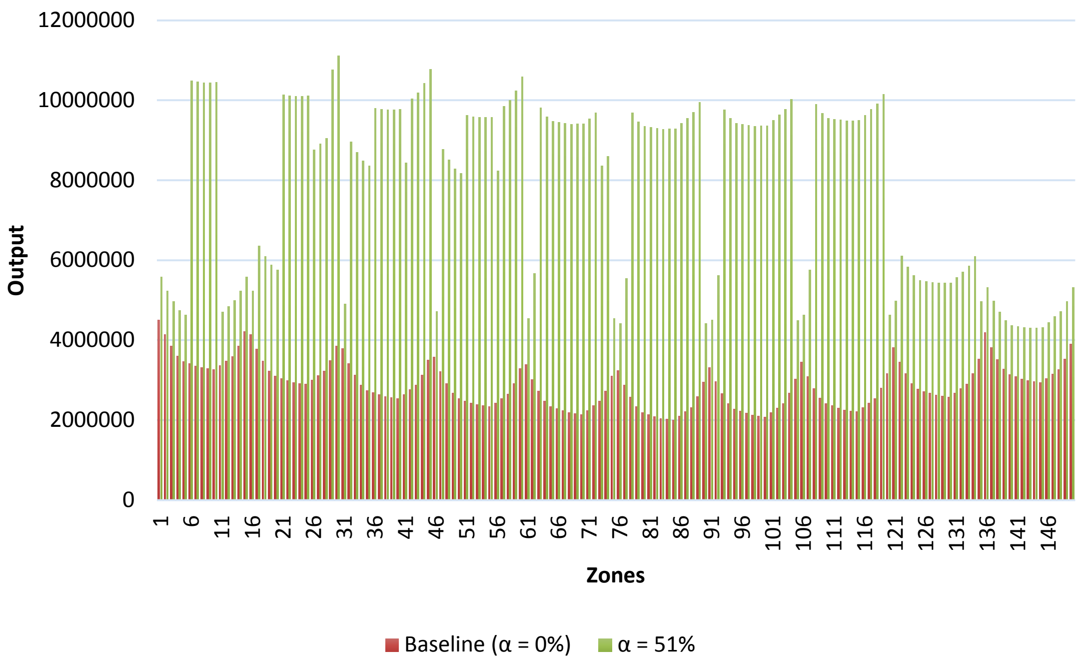


Fig. 51. Baseline Energy vs. Energy with User Preference α = 0.51 (Model #1)

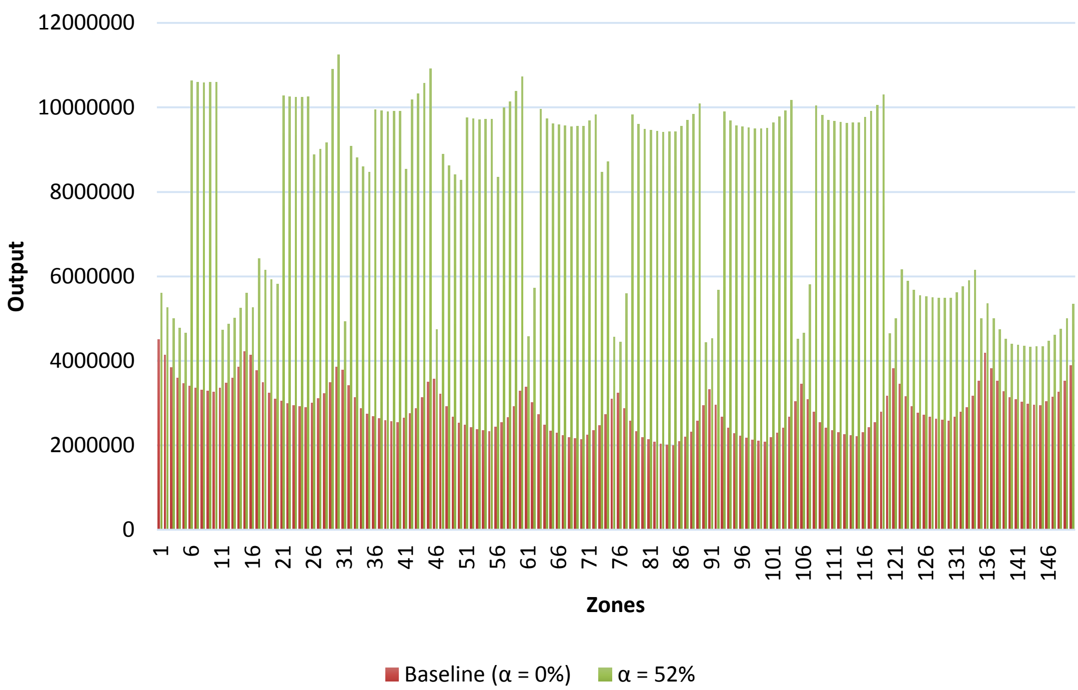
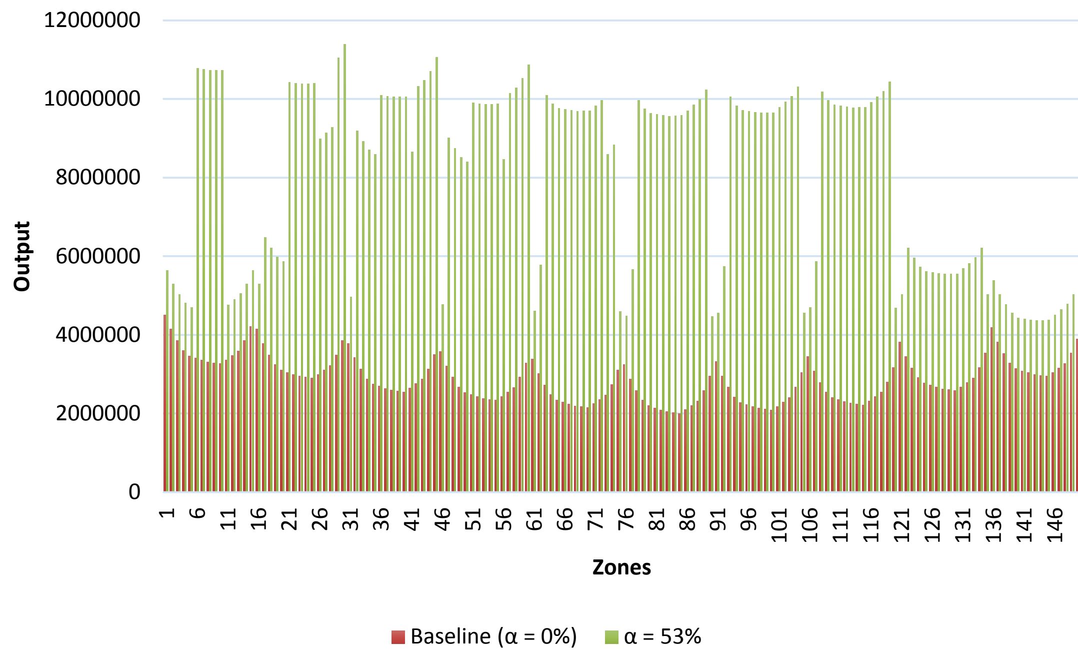


Fig. 52. Baseline Energy vs. Energy with User Preference α = 0.52 (Model #1)



v

Fig. 53. Baseline Energy vs. Energy with User Preference α = 0.53 (Model #1)

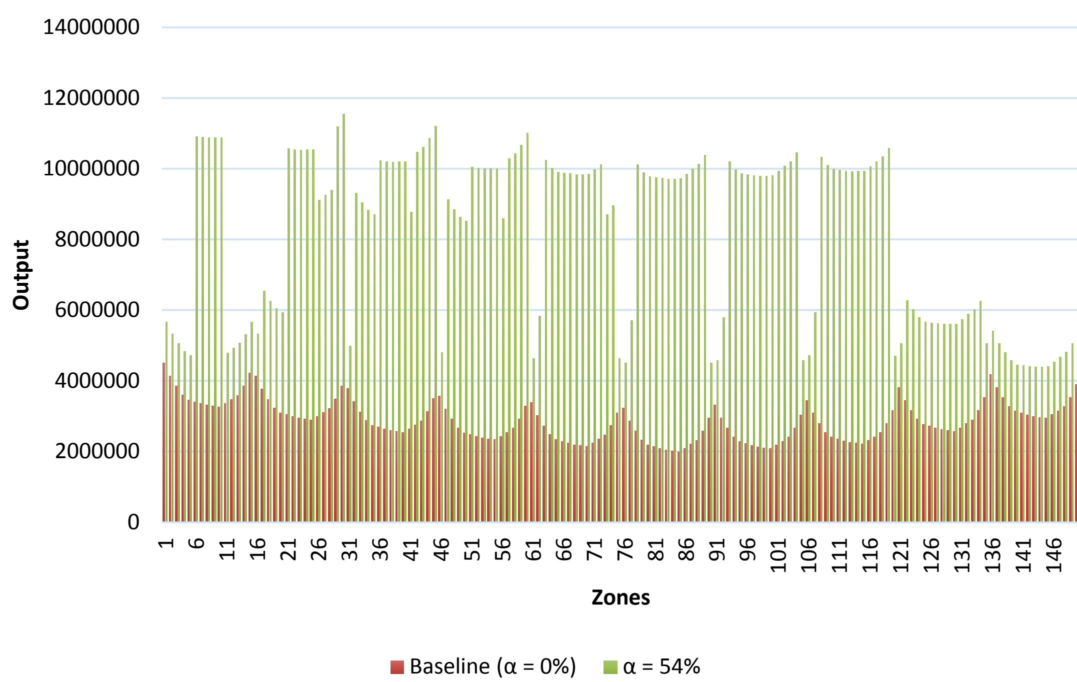


Fig. 54. Baseline Energy vs. Energy with User Preference α = 0.54 (Model #1)

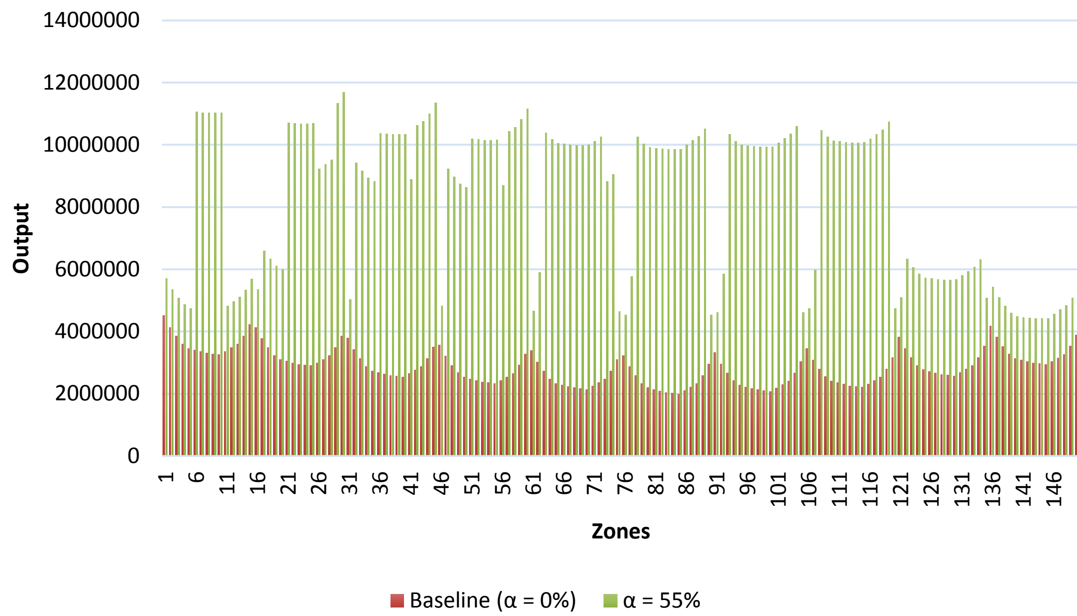


Fig. 55. Baseline Energy vs. Energy with User Preference α = 0.55 (Model #1)

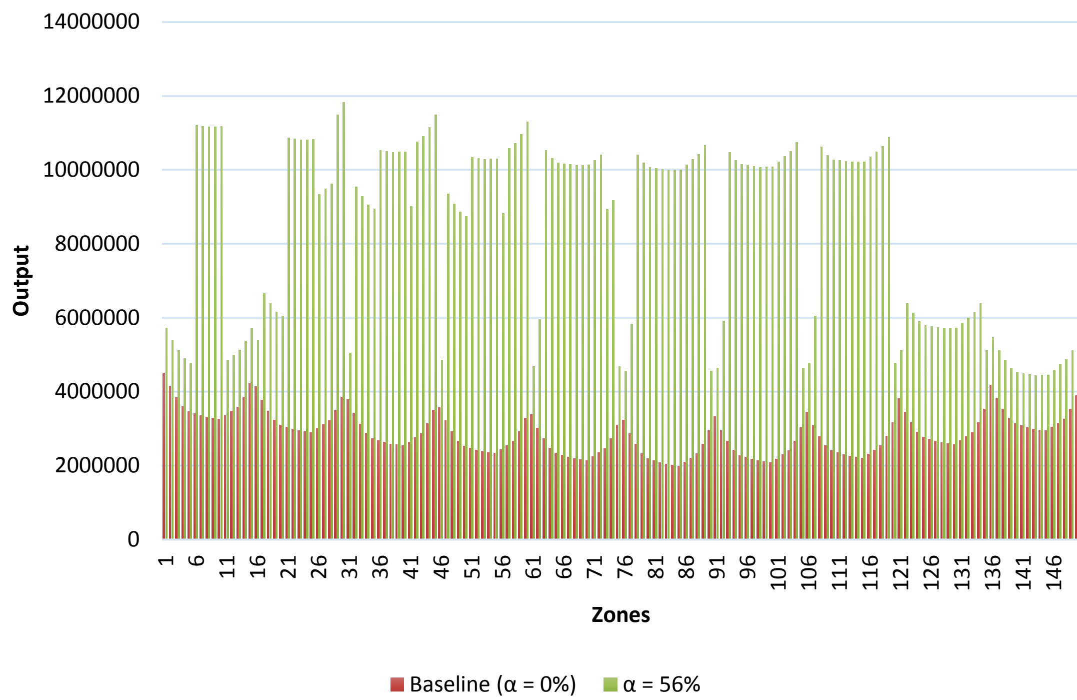


Fig. 56. Baseline Energy vs. Energy with User Preference α = 0.56 (Model #1)

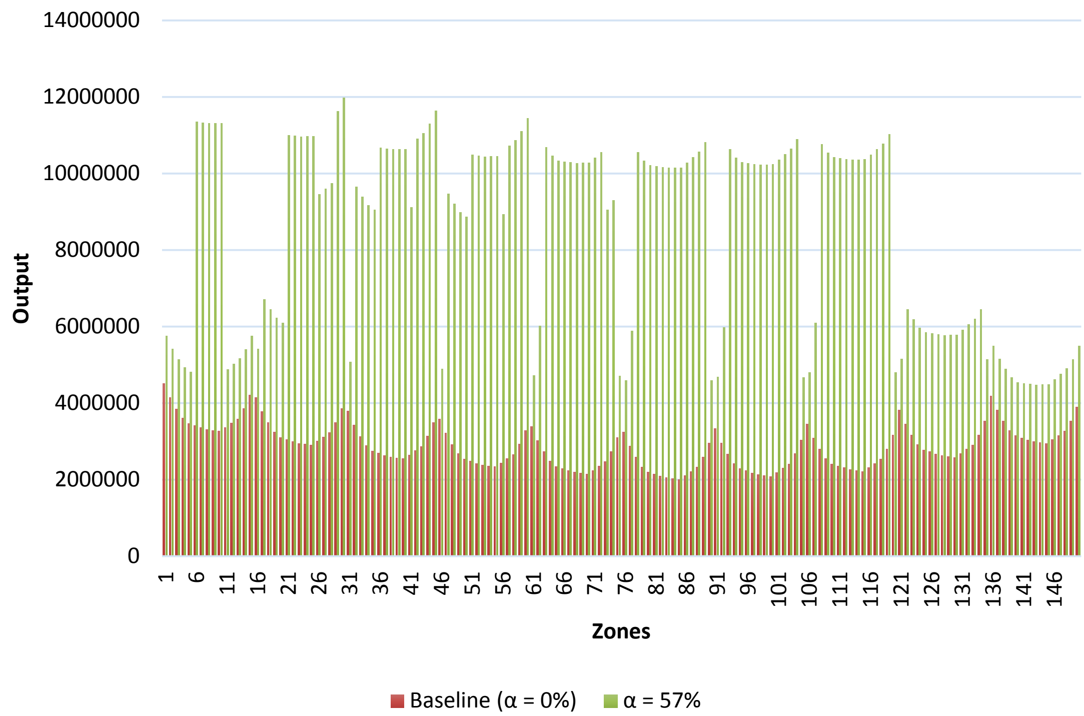


Fig. 57. Baseline Energy vs. Energy with User Preference α = 0.57 (Model #1)

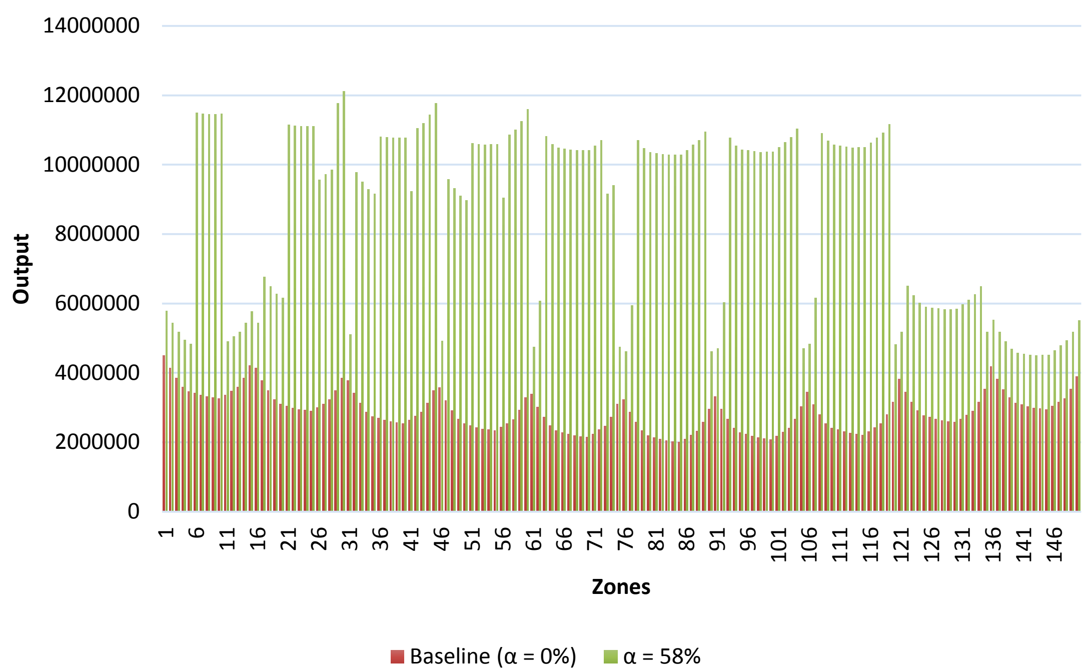


Fig. 58. Baseline Energy vs. Energy with User Preference α = 0.58 (Model #1)

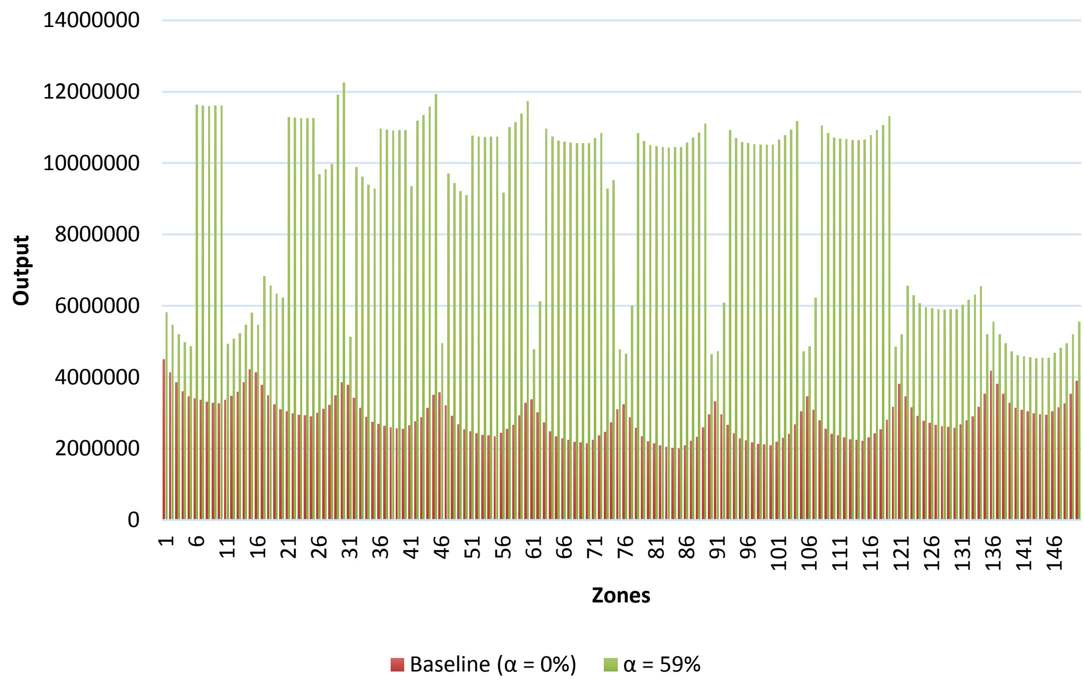


Fig. 59. Baseline Energy vs. Energy with User Preference α = 0.59 (Model #1)

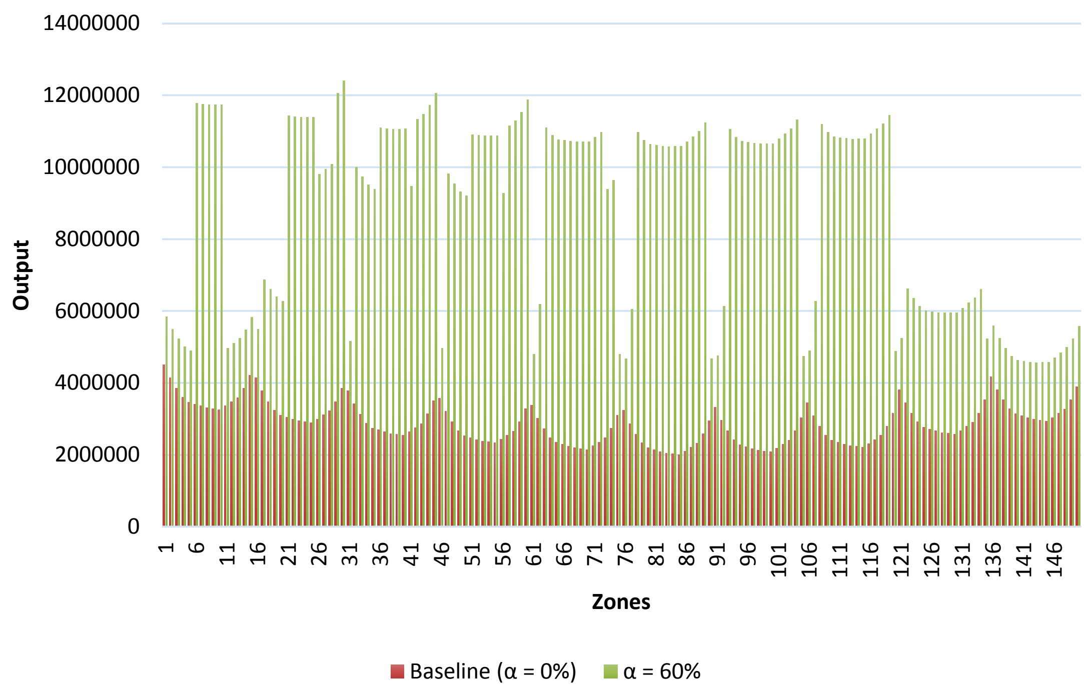


Fig. 60. Baseline Energy vs. Energy with User Preference α = 0.60 (Model #1)

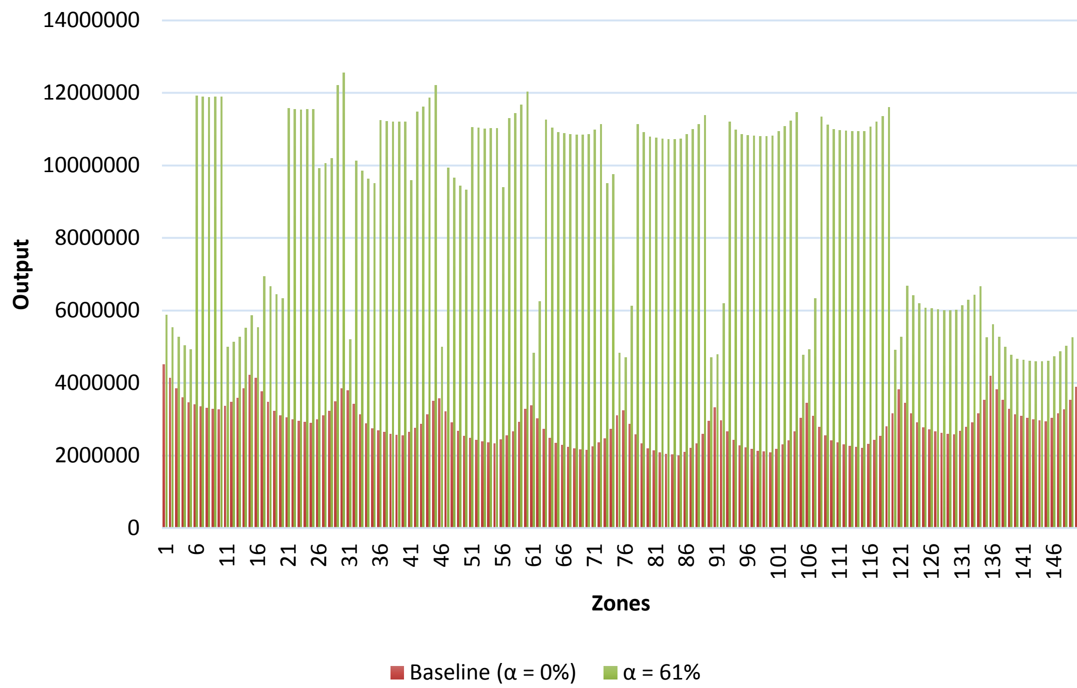


Fig. 61. Baseline Energy vs. Energy with User Preference α = 0.61 (Model #1)

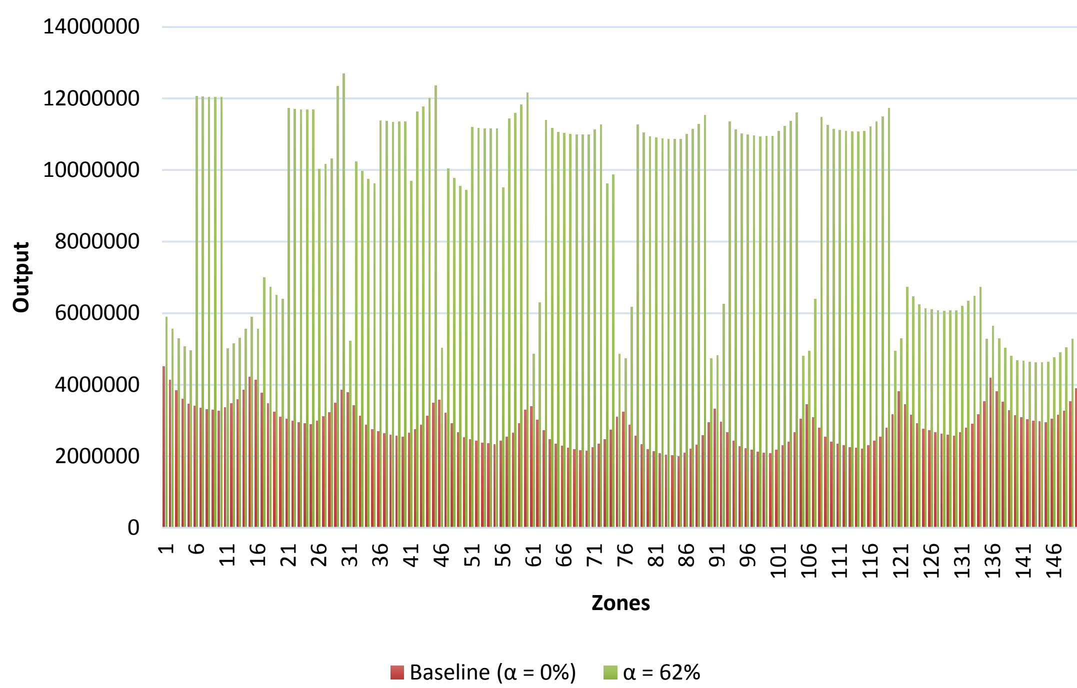


Fig. 62. Baseline Energy vs. Energy with User Preference α = 0.62 (Model #1)

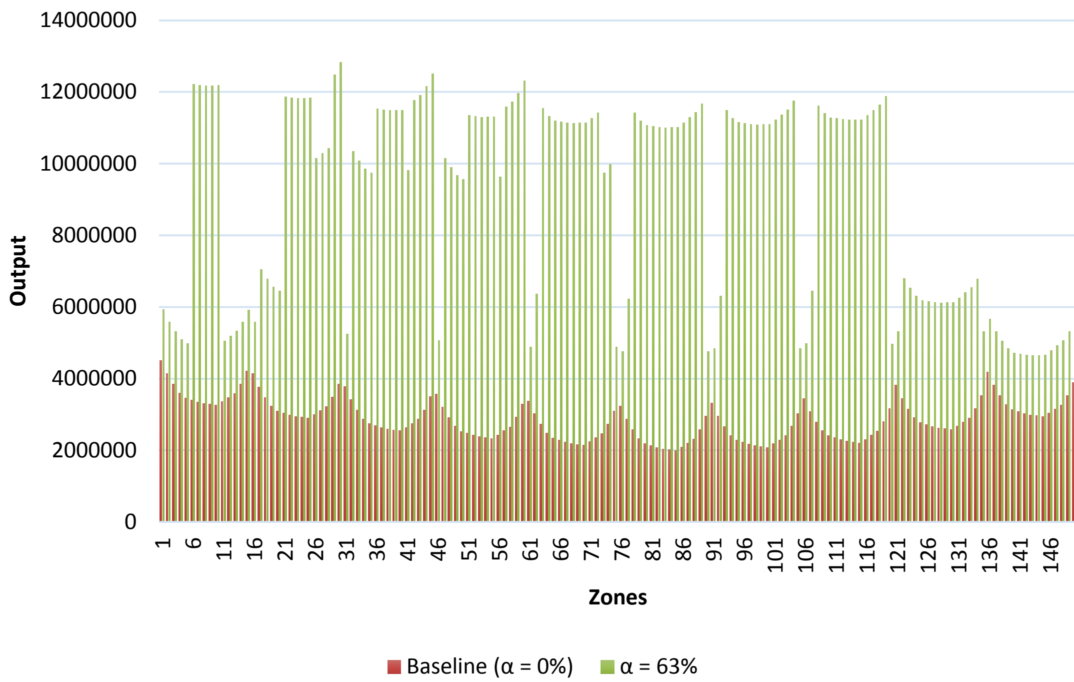


Fig. 63. Baseline Energy vs. Energy with User Preference α = 0.63 (Model #1)

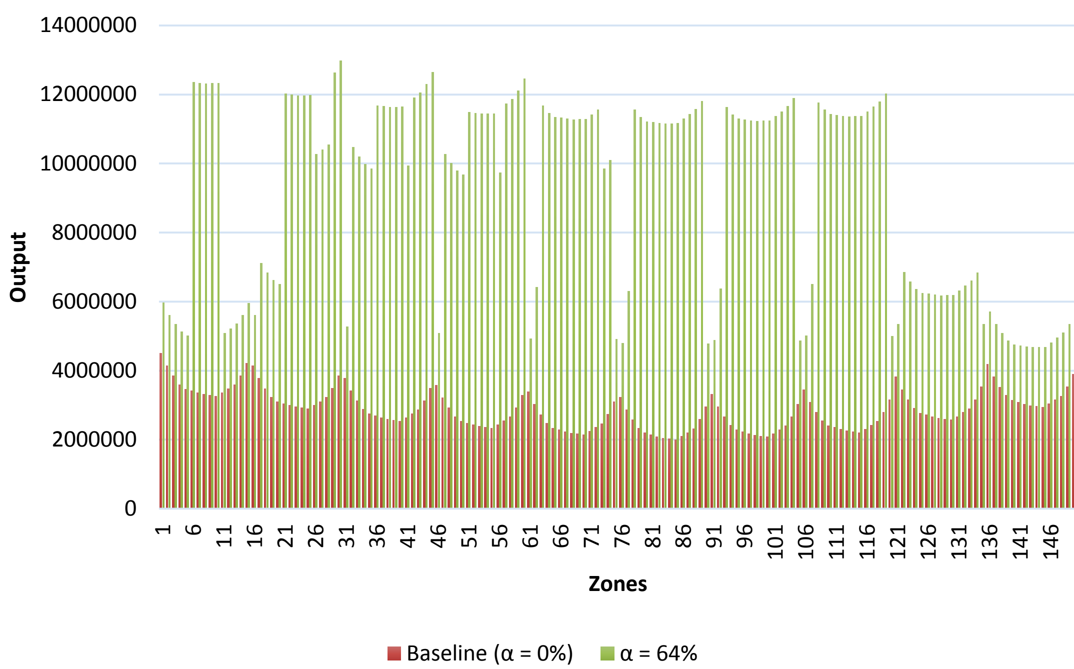


Fig. 64. Baseline Energy vs. Energy with User Preference α = 0.64 (Model #1)

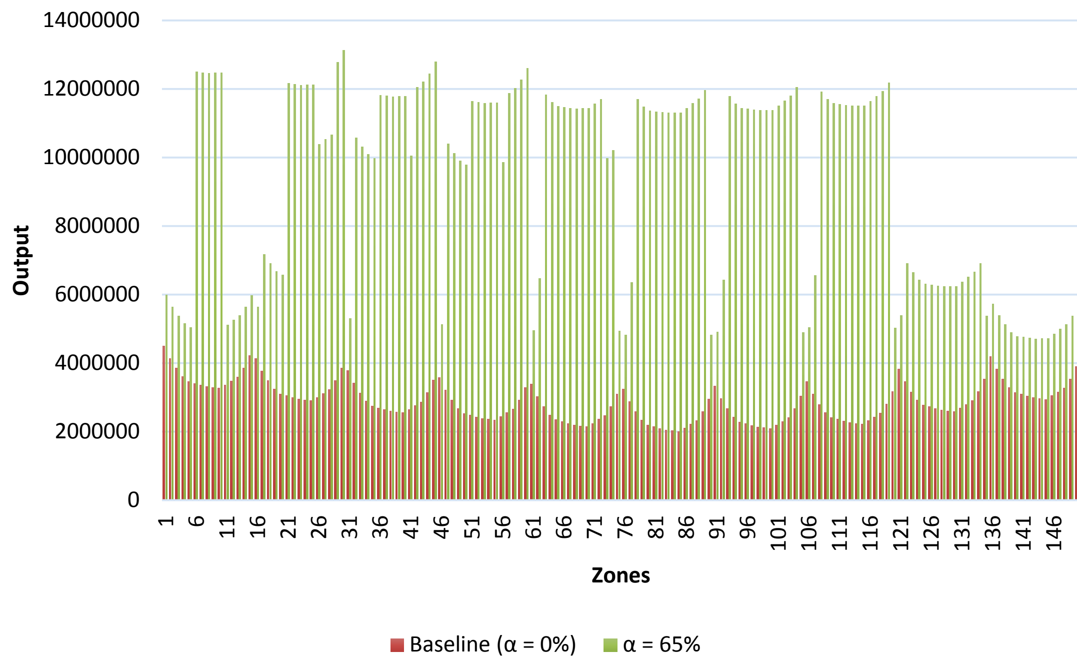


Fig. 65. Baseline Energy vs. Energy with User Preference α = 0.65 (Model #1)

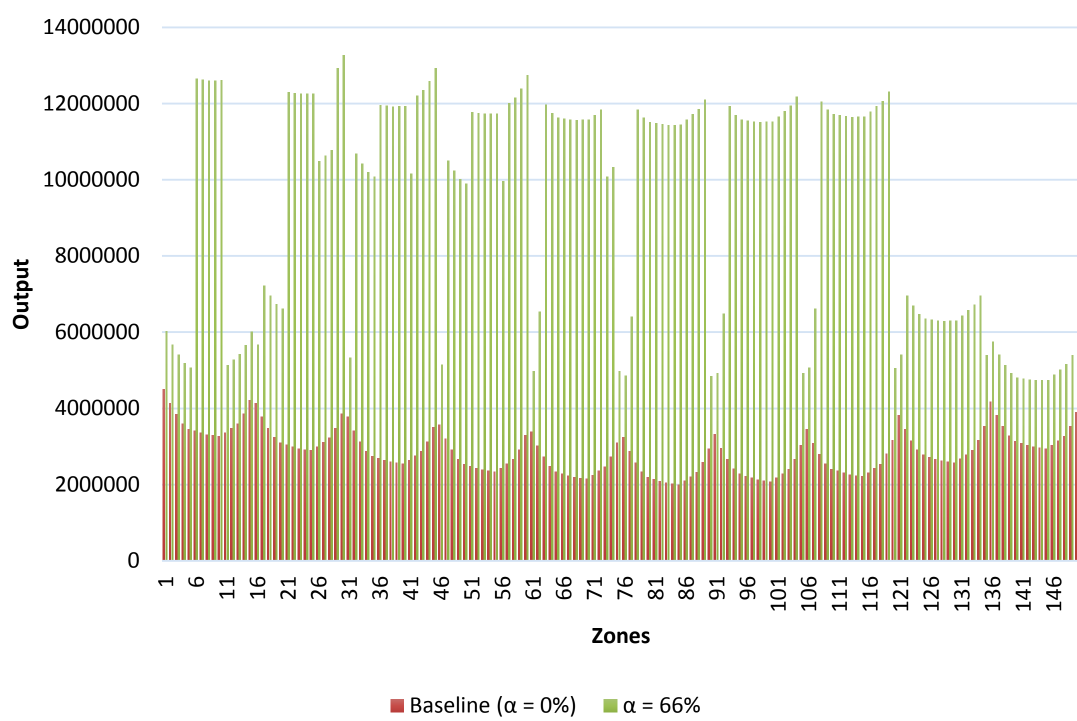


Fig. 66. Baseline Energy vs. Energy with User Preference α = 0.66 (Model #1)

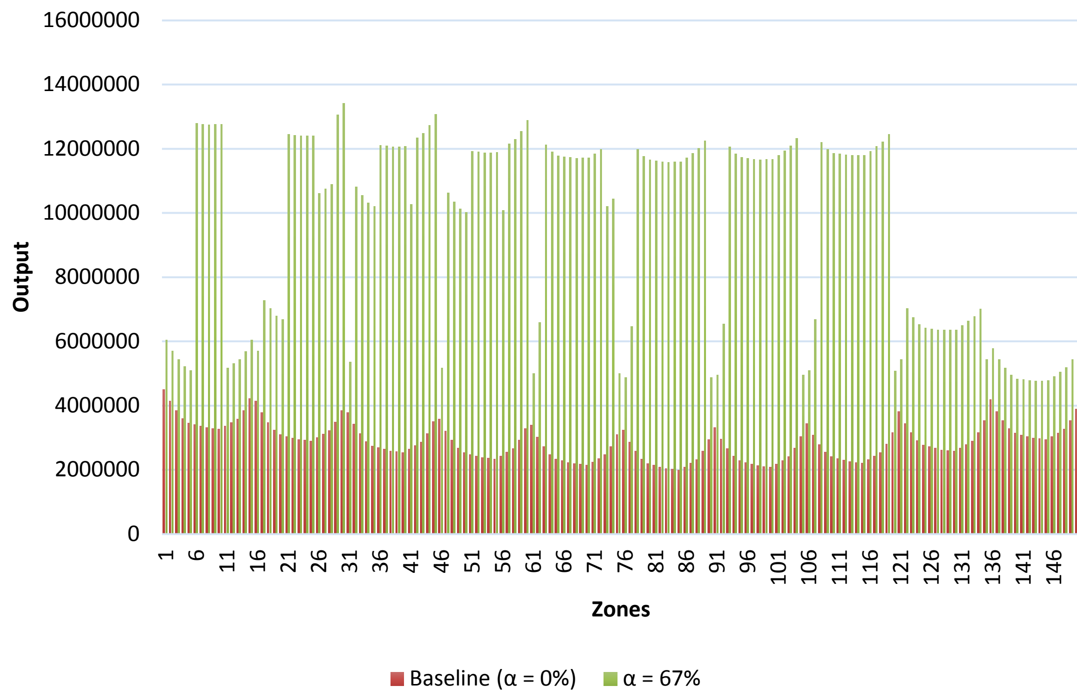


Fig. 67. Baseline Energy vs. Energy with User Preference α = 0.67 (Model #1)

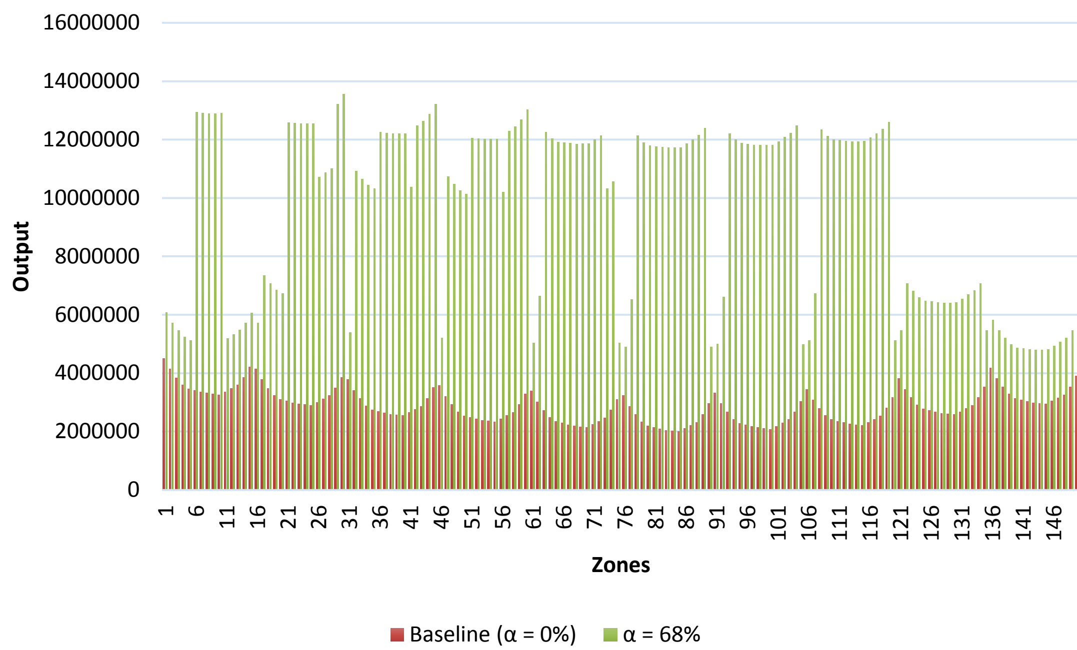


Fig. 68. Baseline Energy vs. Energy with User Preference α = 0.68 (Model #1)

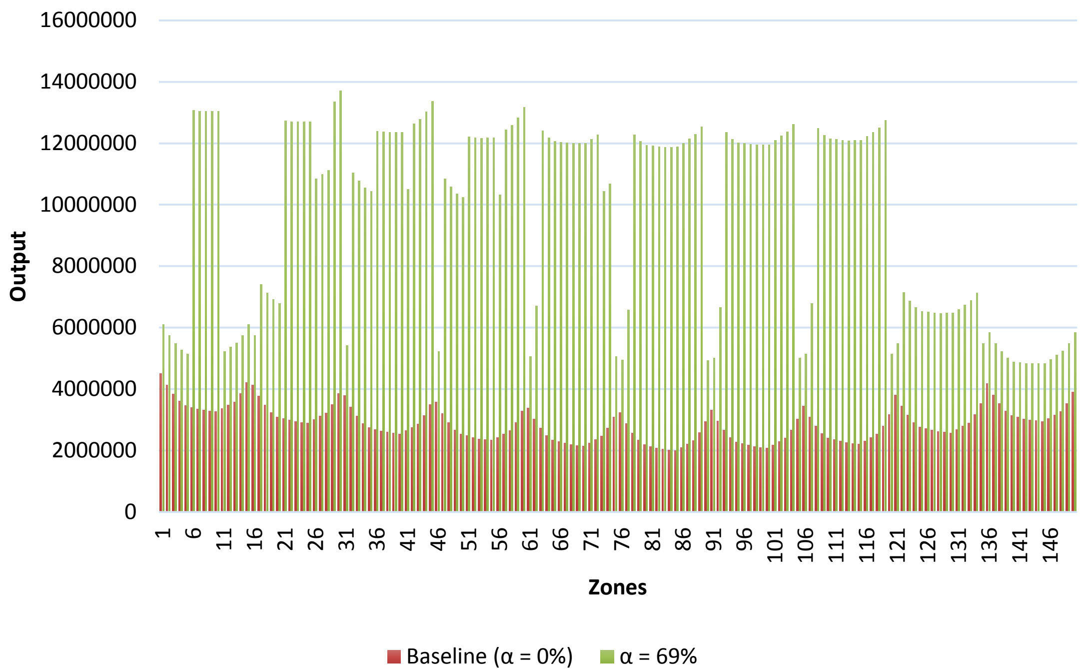


Fig. 69. Baseline Energy vs. Energy with User Preference α = 0.69 (Model #1)

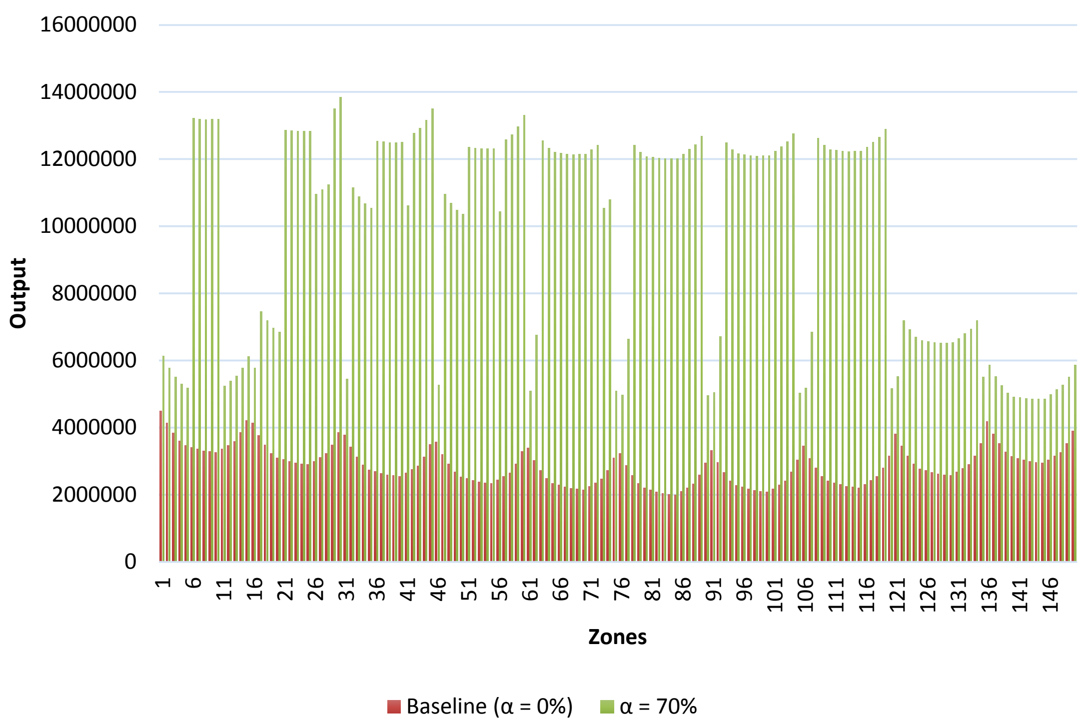


Fig. 70. Baseline Energy vs. Energy with User Preference α = 0.70 (Model #1)

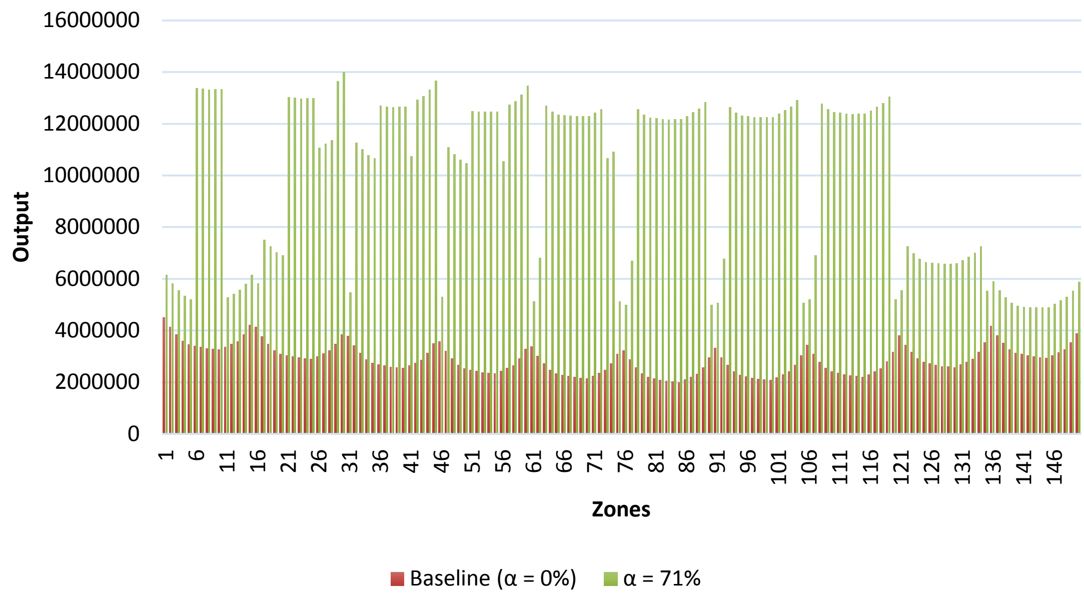


Fig. 71. Baseline Energy vs. Energy with User Preference α = 0.71 (Model #1)

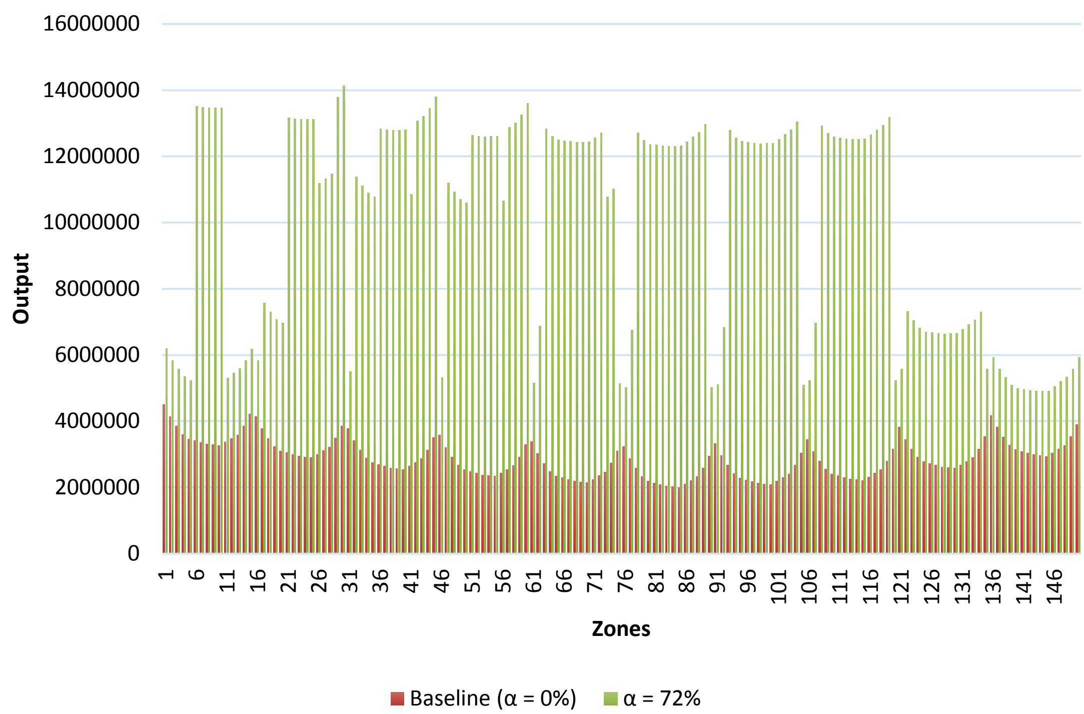


Fig. 72. Baseline Energy vs. Energy with User Preference α = 0.72 (Model #2)

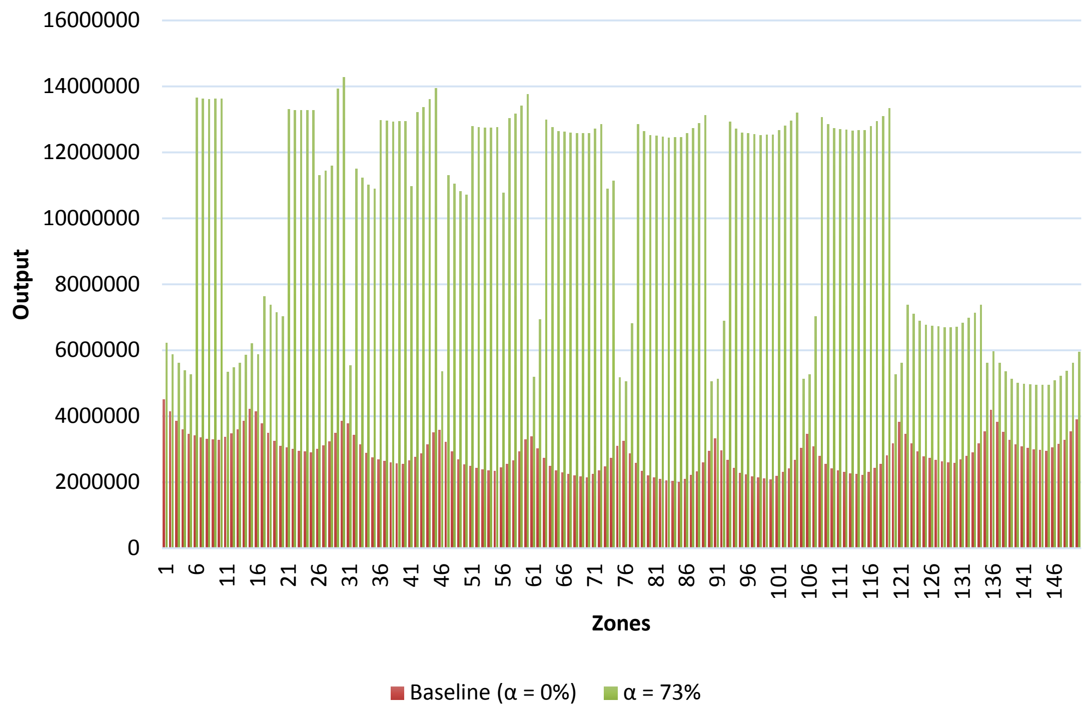


Fig. 73. Baseline Energy vs. Energy with User Preference α = 0.73 (Model #1)

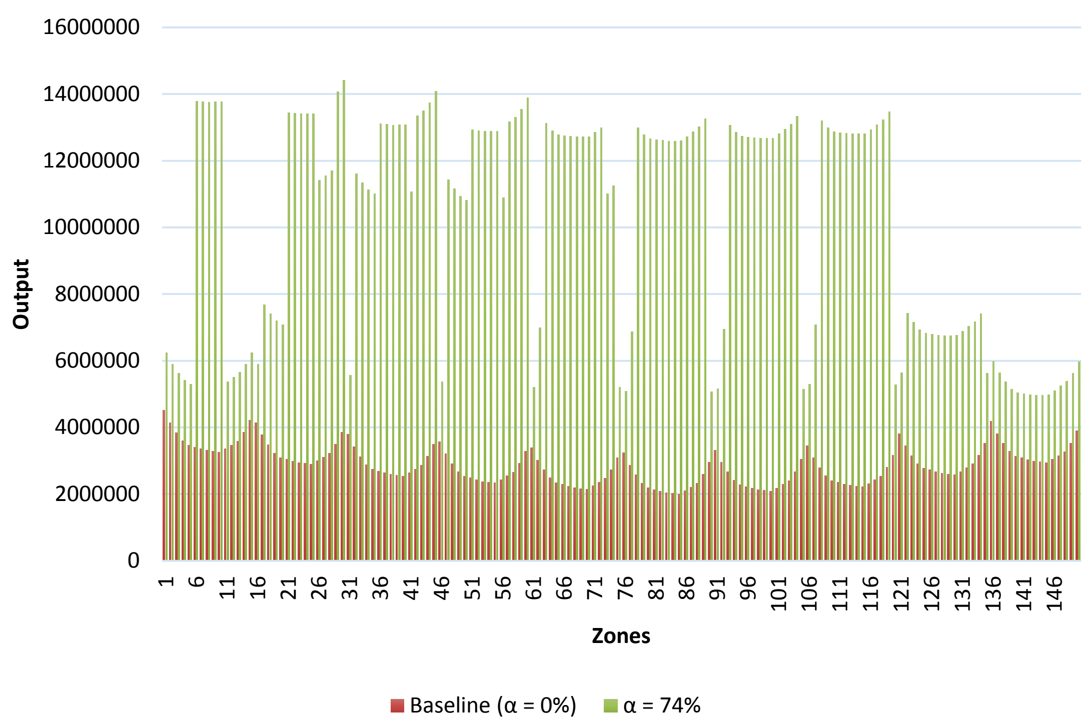


Fig. 74. Baseline Energy vs. Energy with User Preference α = 0.74 (Model #1)

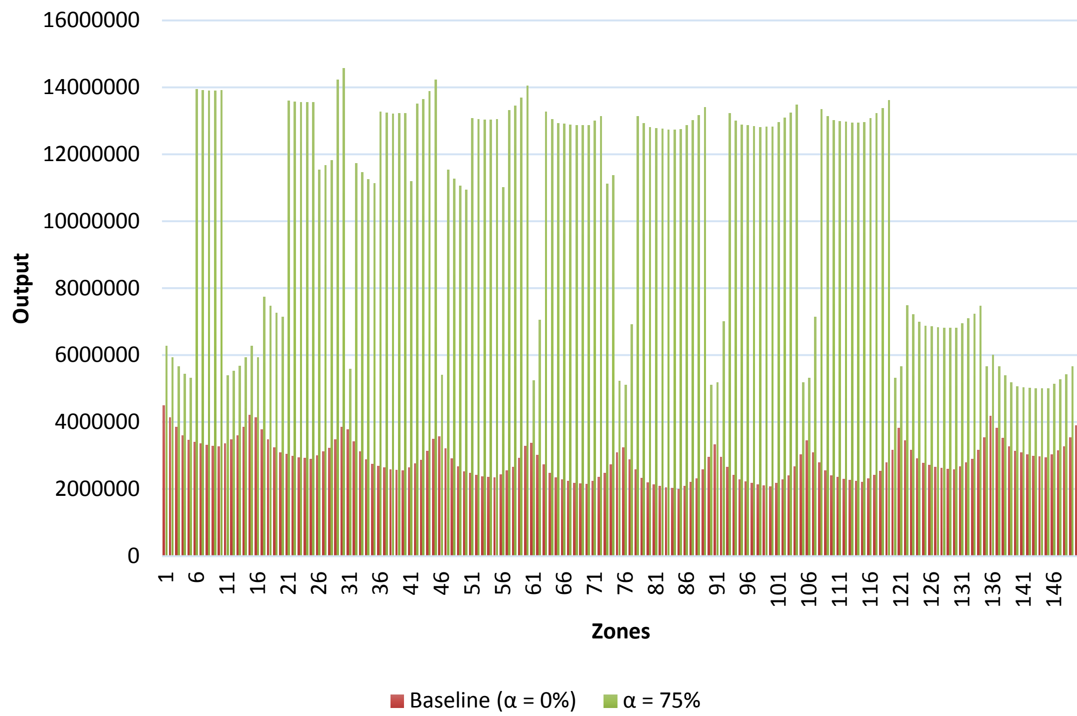


Fig. 75. Baseline Energy vs. Energy with User Preference α = 0.75 (Model #1)

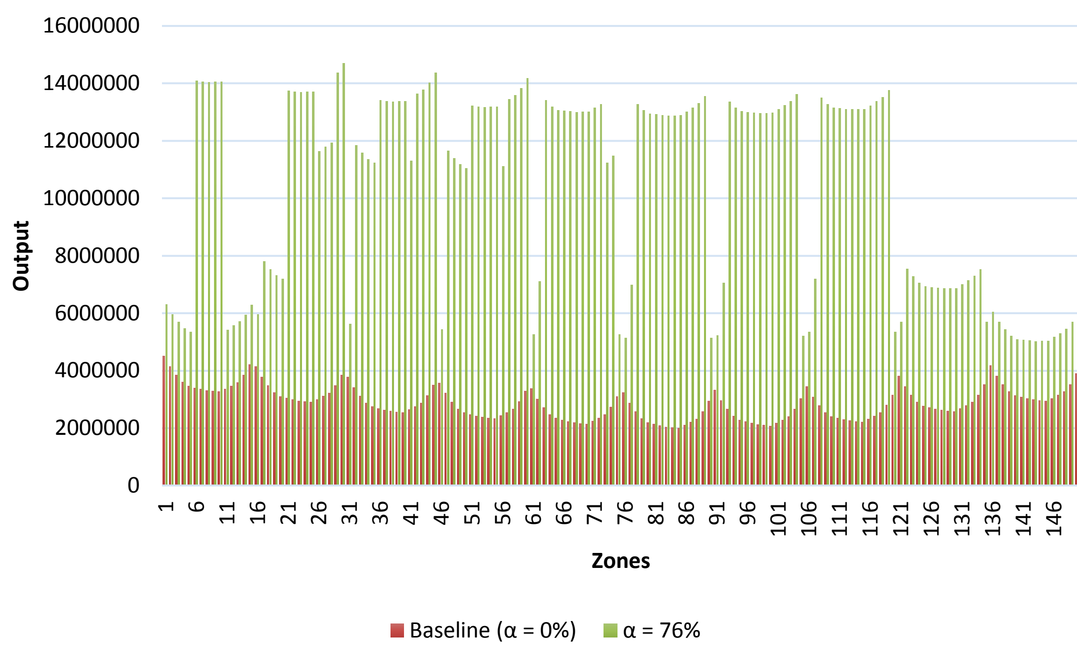


Fig. 76. Baseline Energy vs. Energy with User Preference α = 0.76 (Model #1)

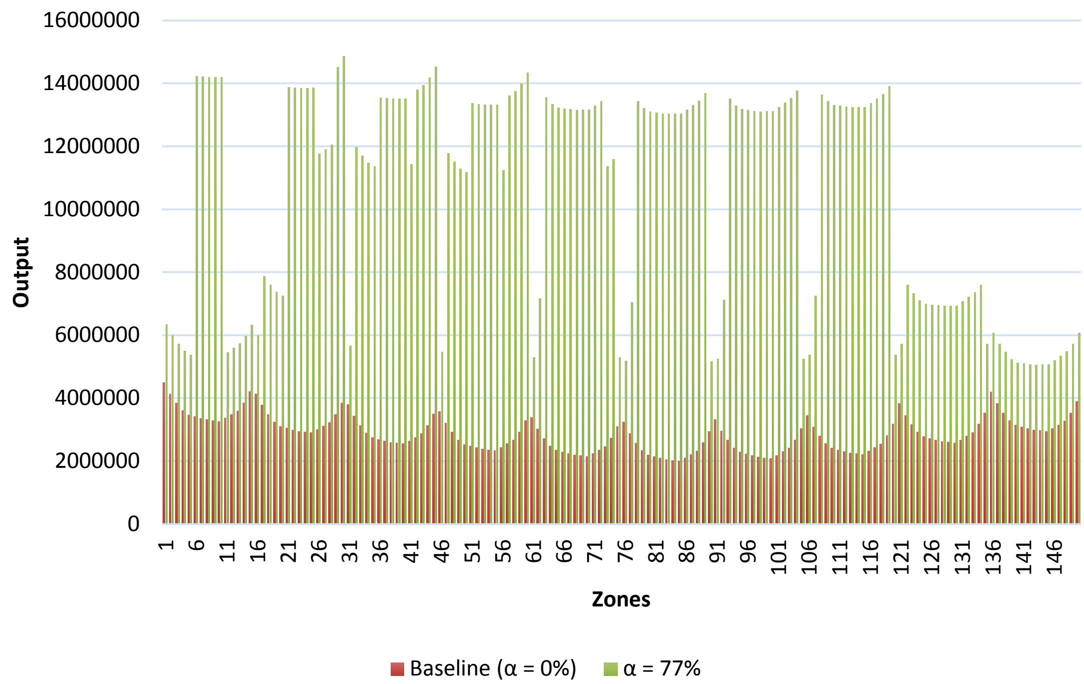


Fig. 77. Baseline Energy vs. Energy with User Preference α = 0.77 (Model #1)

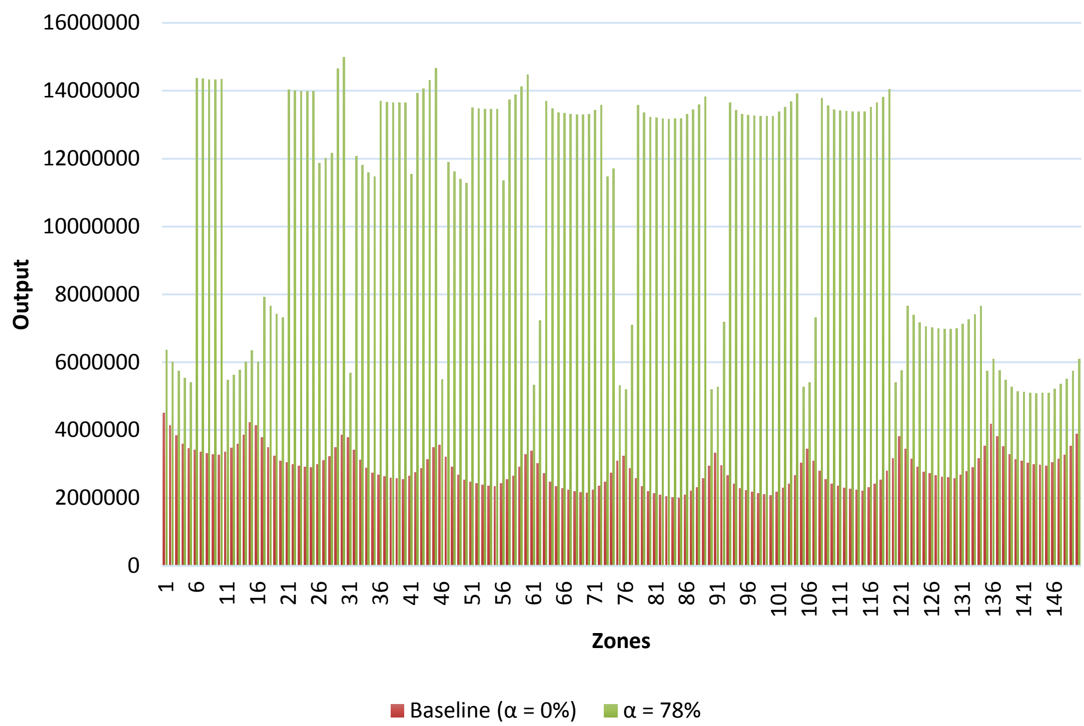


Fig. 78. Baseline Energy vs. Energy with User Preference α = 0.78 (Model #1)

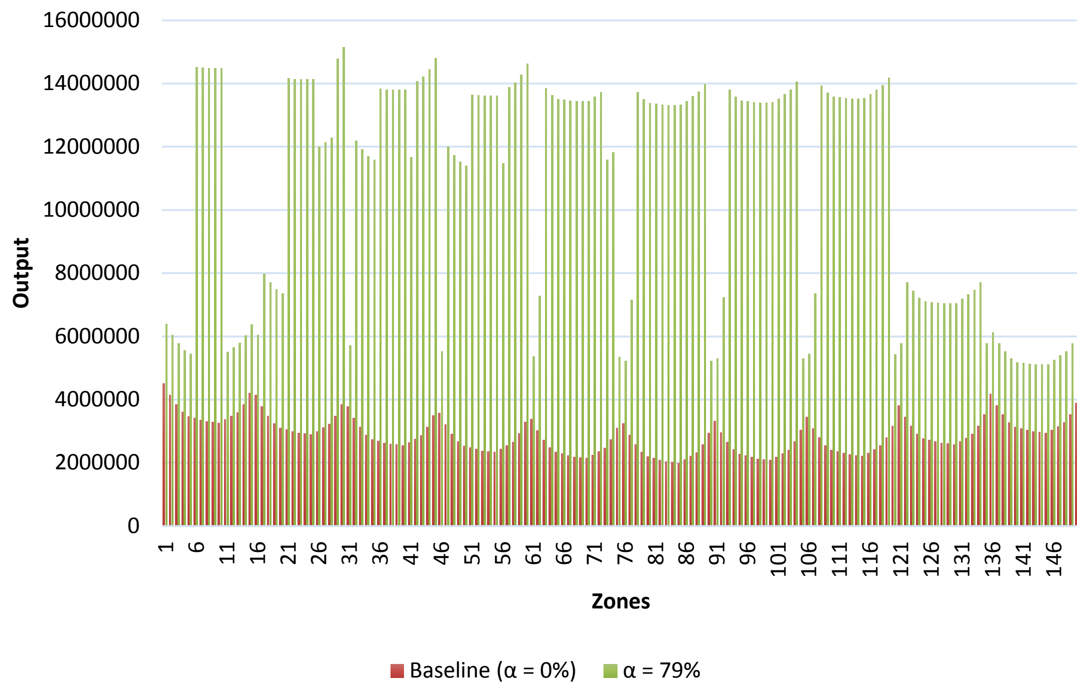


Fig. 79. Baseline Energy vs. Energy with User Preference α = 0.79 (Model #1)

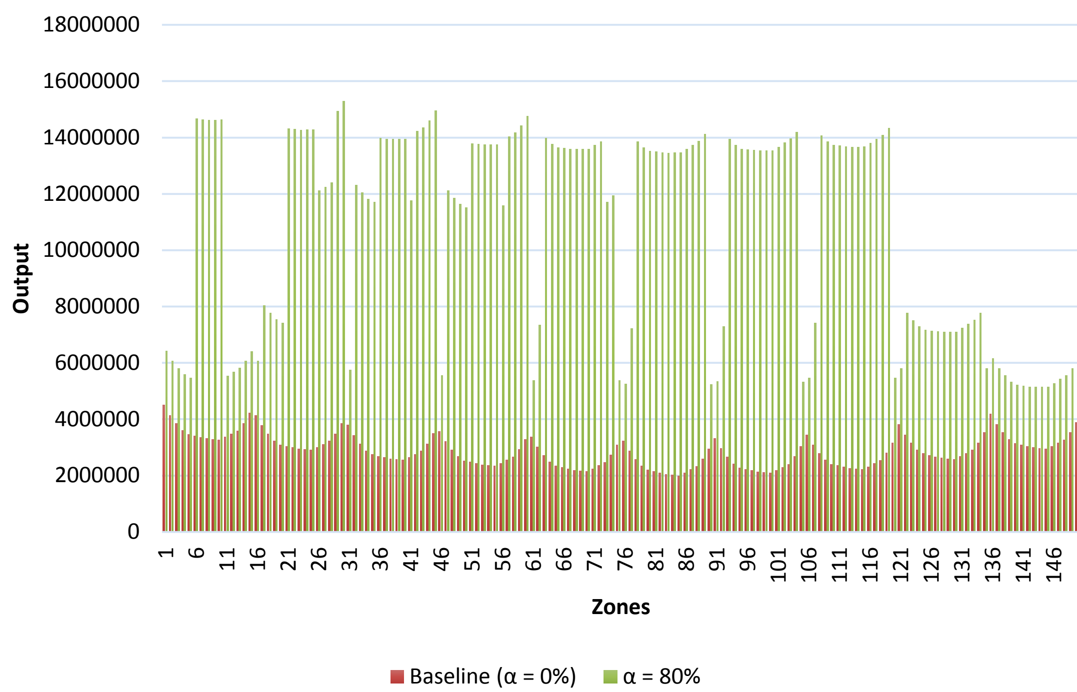


Fig. 80. Baseline Energy vs. Energy with User Preference α = 0.80 (Model #1)

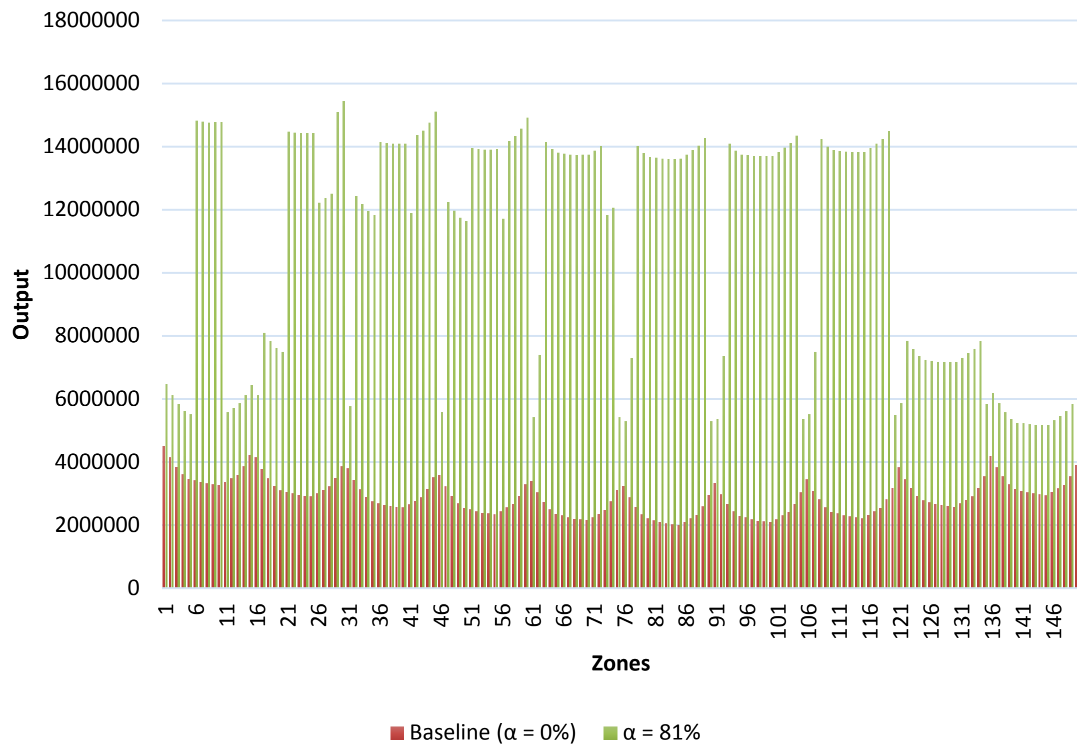


Fig. 81. Baseline Energy vs. Energy with User Preference α = 0.81 (Model #1)

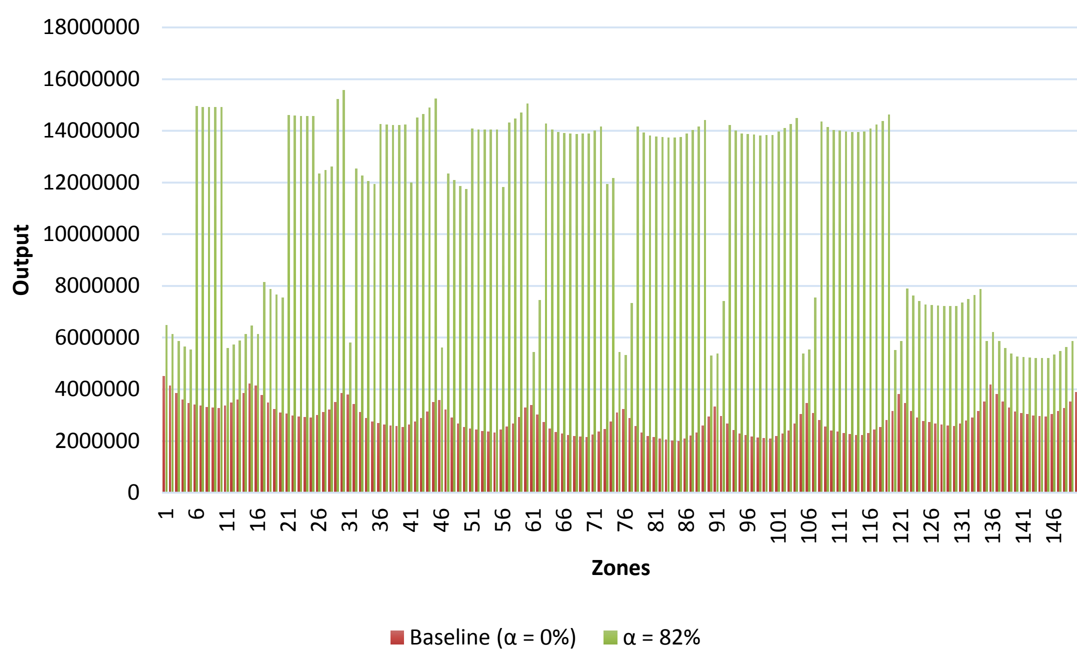


Fig. 82. Baseline Energy vs. Energy with User Preference α = 0.82 (Model #1)

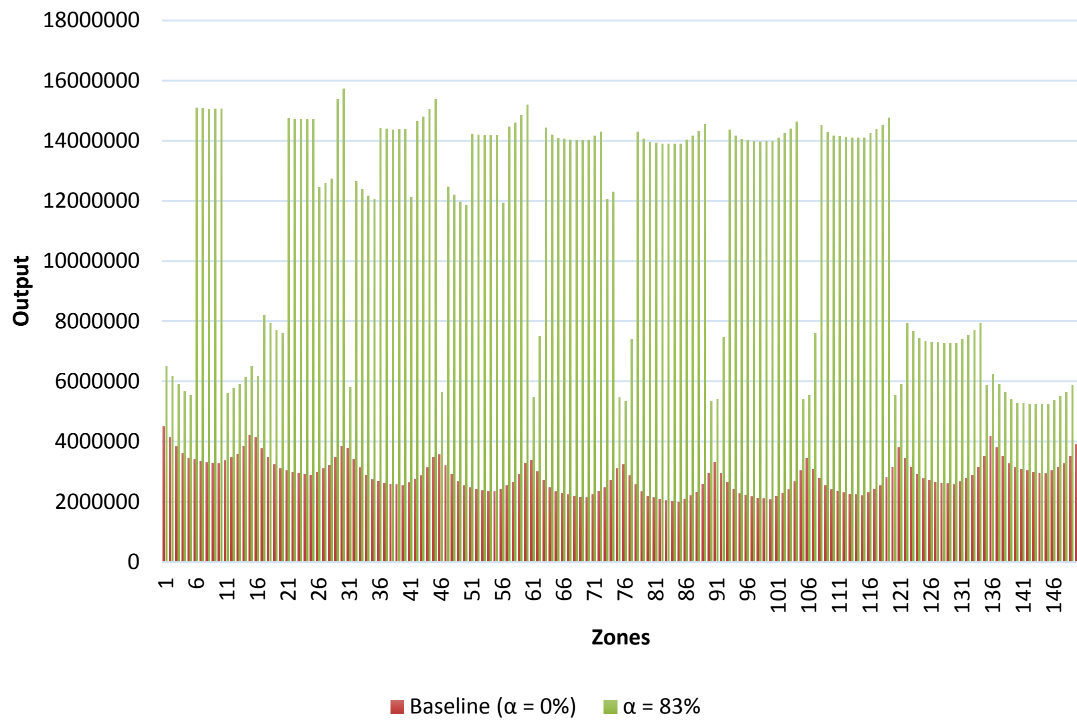


Fig. 83. Baseline Energy vs. Energy with User Preference α = 0.83 (Model #1)

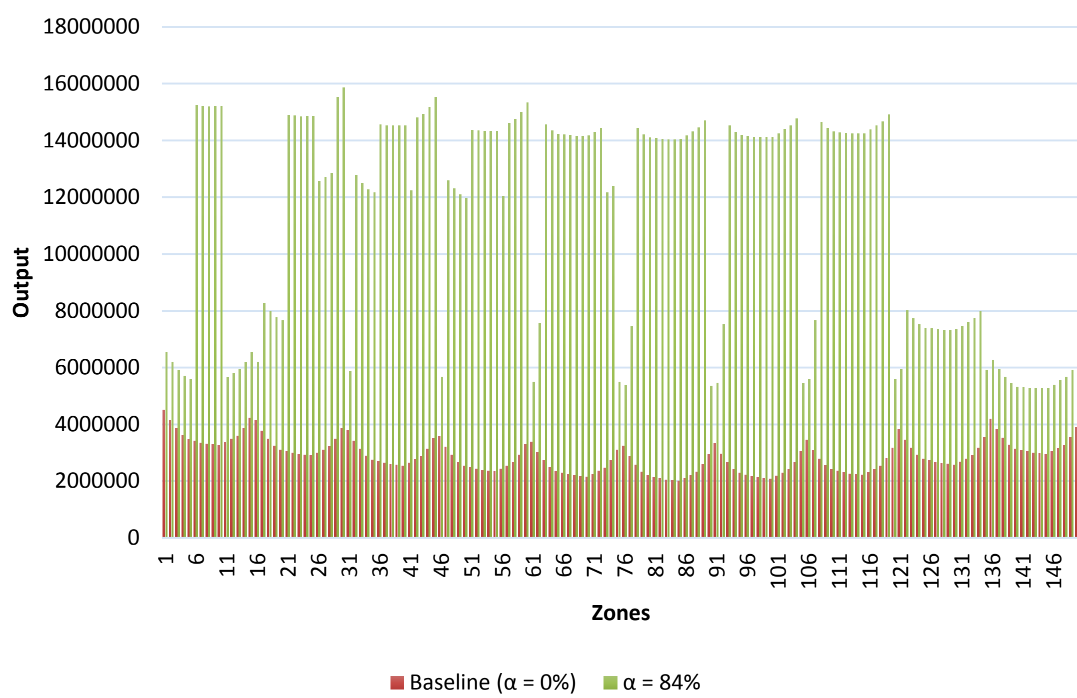


Fig. 84. Baseline Energy vs. Energy with User Preference α = 0.84 (Model #1)

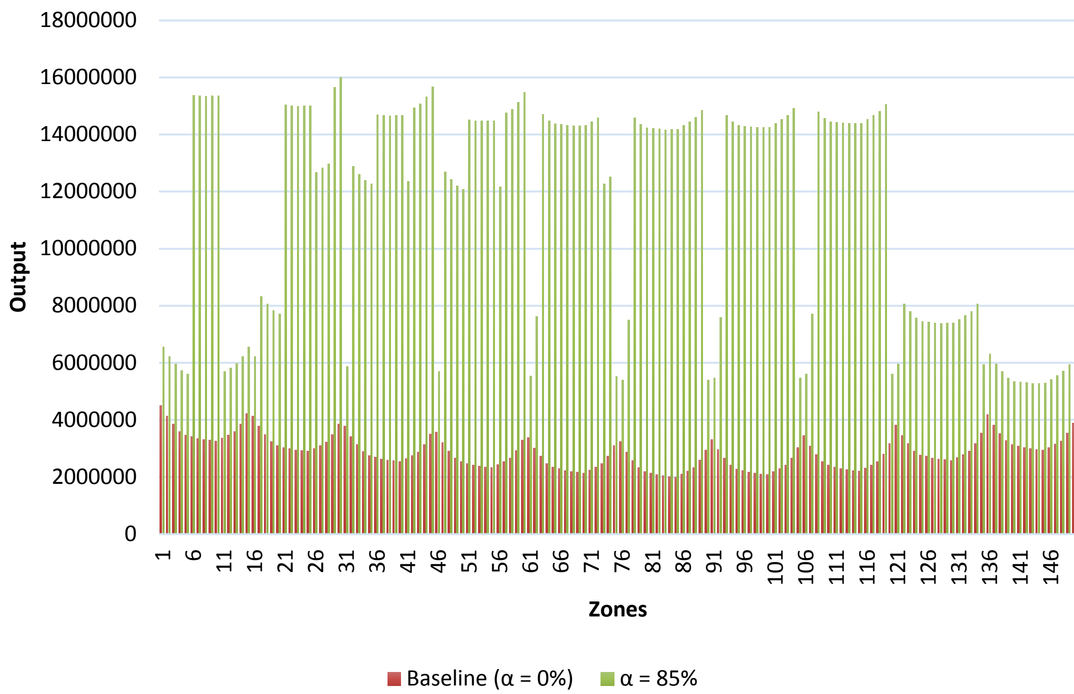


Fig. 85. Baseline Energy vs. Energy with User Preference α = 0.85 (Model #2)

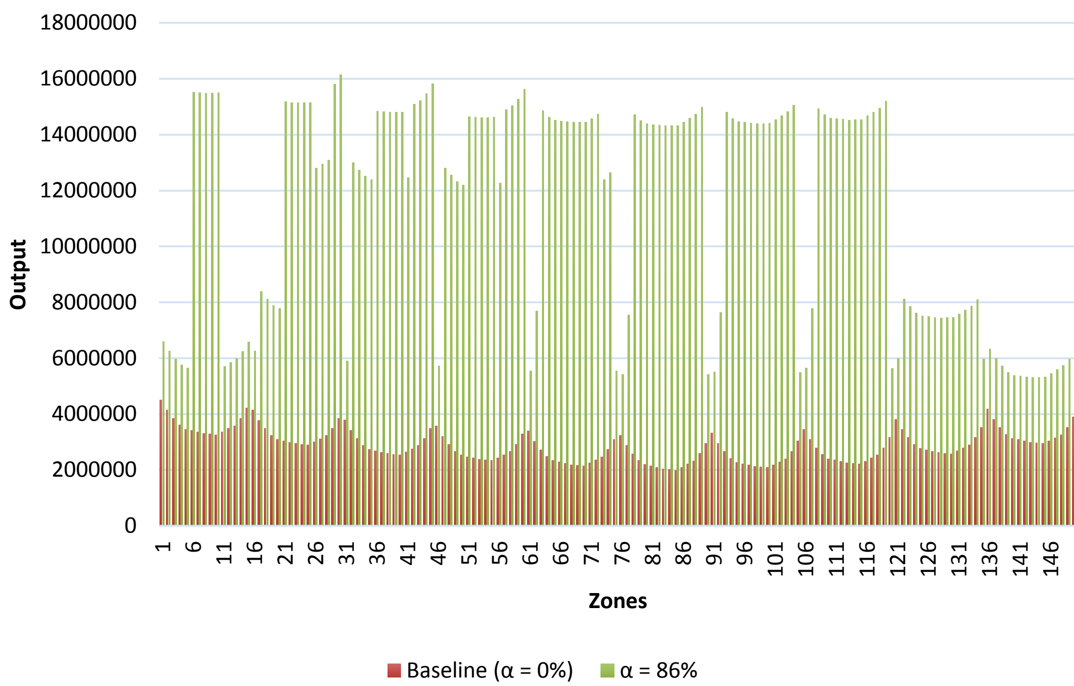


Fig. 86. Baseline Energy vs. Energy with User Preference α = 0.86 (Model #1)

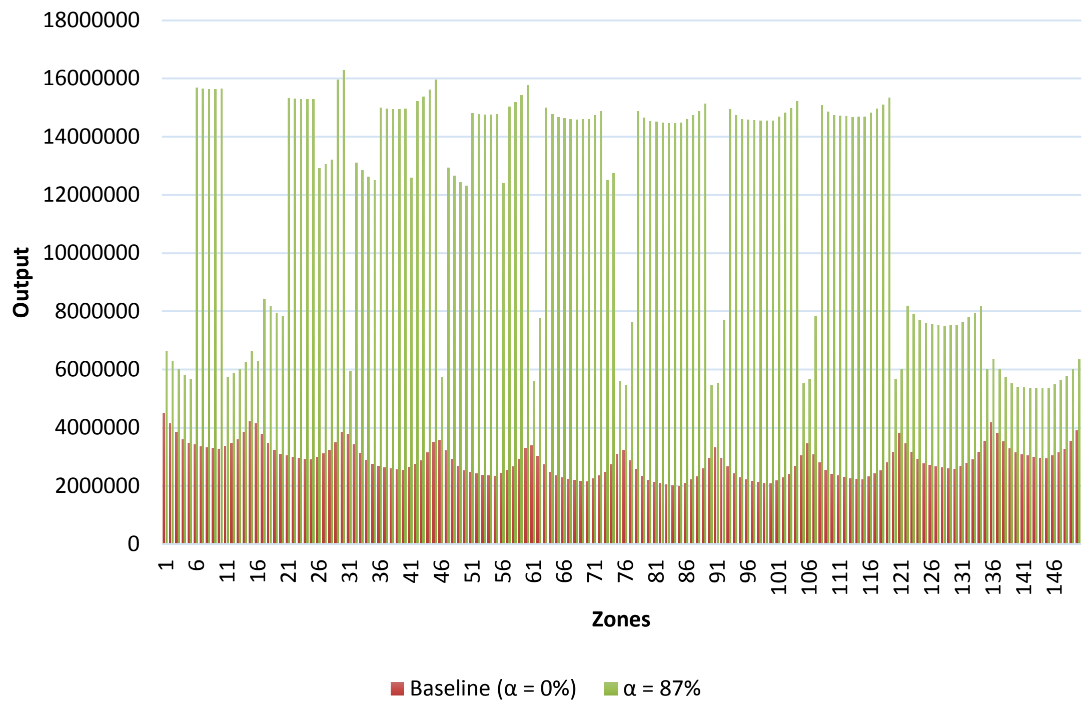


Fig. 87. Baseline Energy vs. Energy with User Preference α = 0.87 (Model #1)

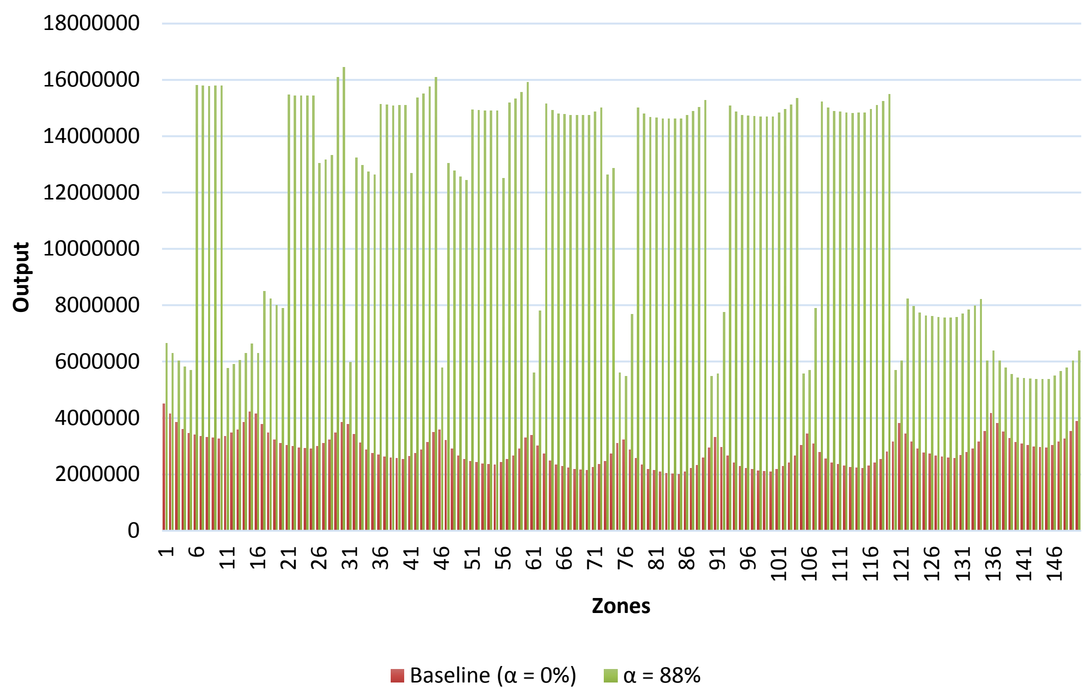


Fig. 88. Baseline Energy vs. Energy with User Preference α = 0.88 (Model #1)

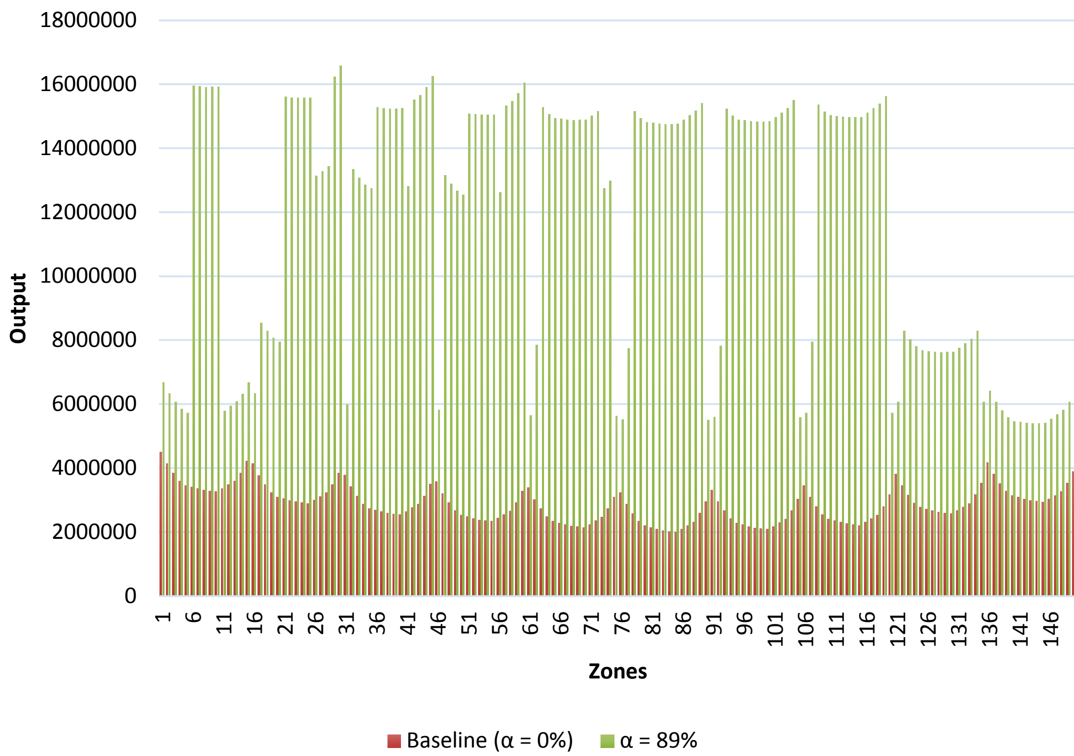


Fig. 89. Baseline Energy vs. Energy with User Preference α = 0.89 (Model #1)

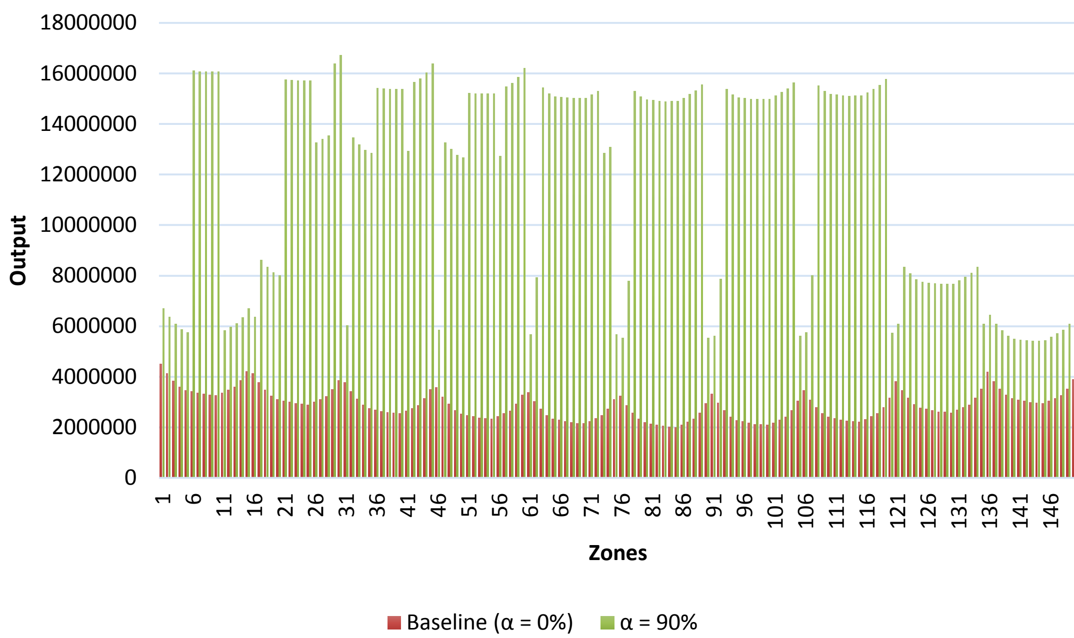


Fig. 90. Baseline Energy vs. Energy with User Preference α = 0.90 (Model #1)

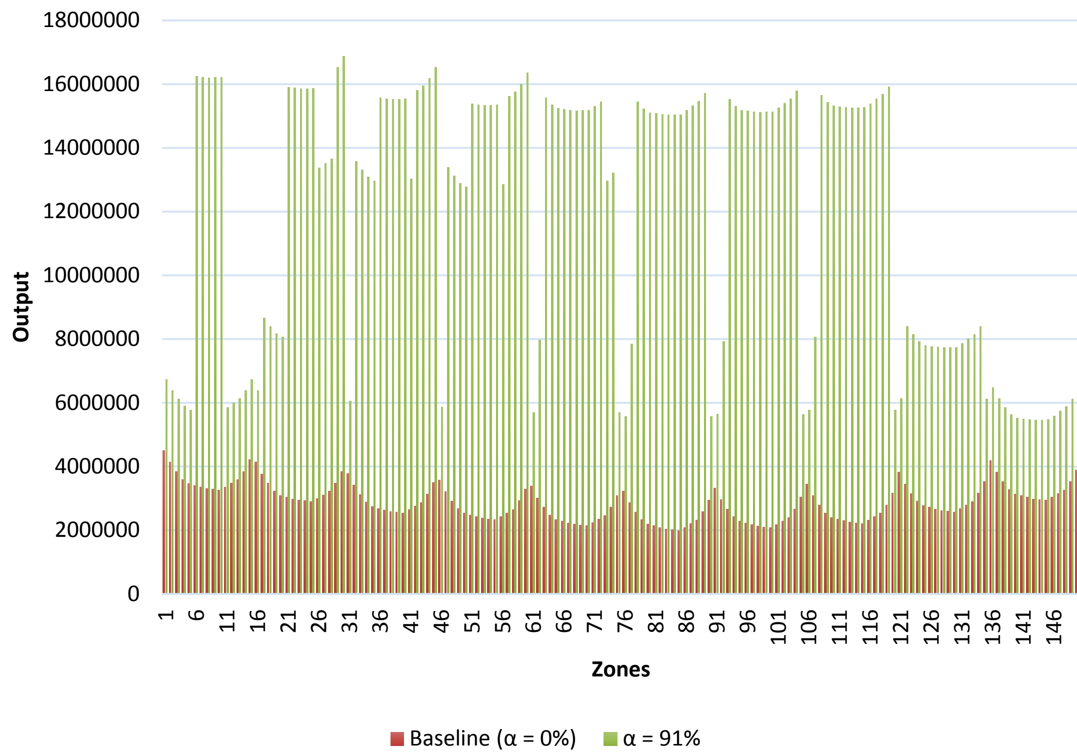


Fig. 91. Baseline Energy vs. Energy with User Preference α = 0.91 (Model #1)

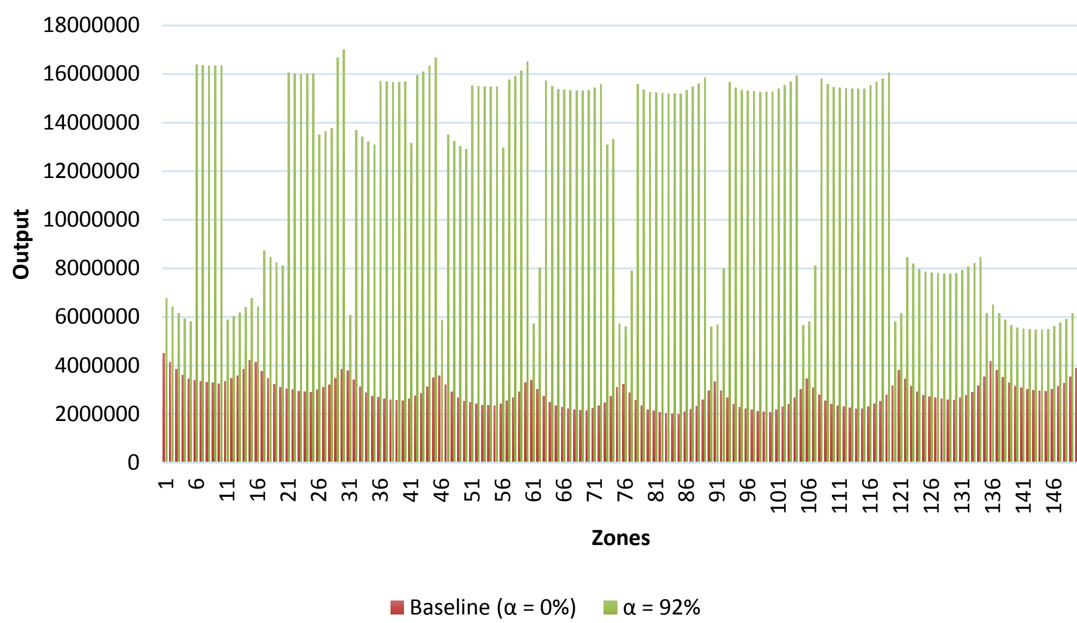


Fig. 92. Baseline Energy vs. Energy with User Preference α = 0.92 (Model #1)

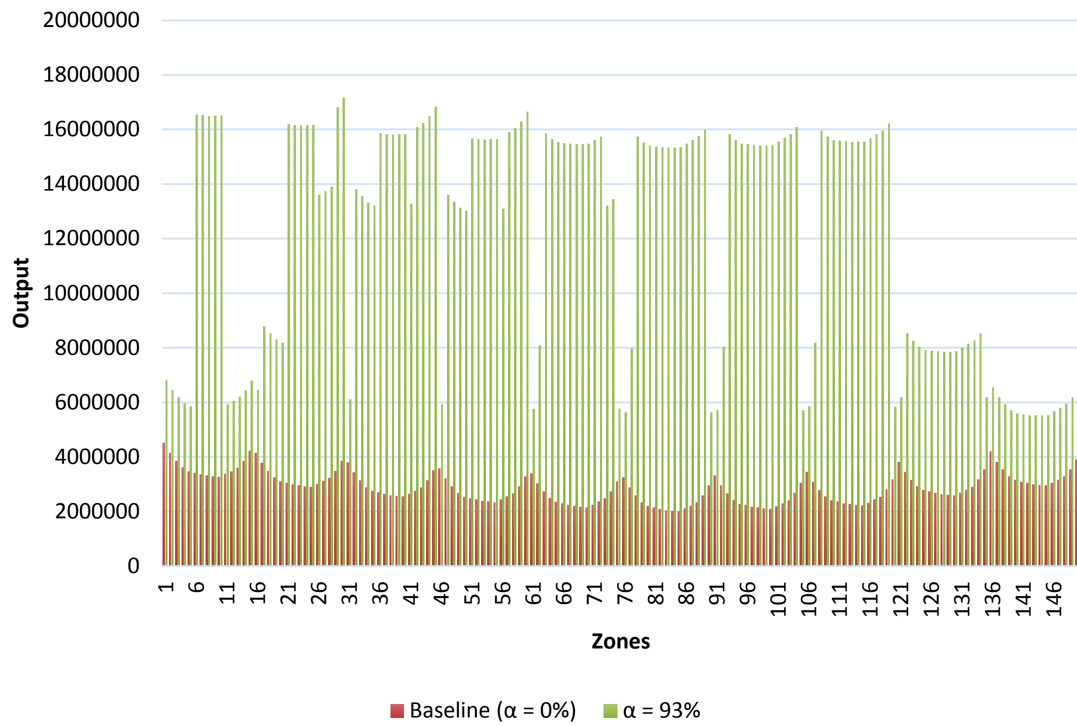


Fig. 93. Baseline Energy vs. Energy with User Preference α = 0.93 (Model #2)

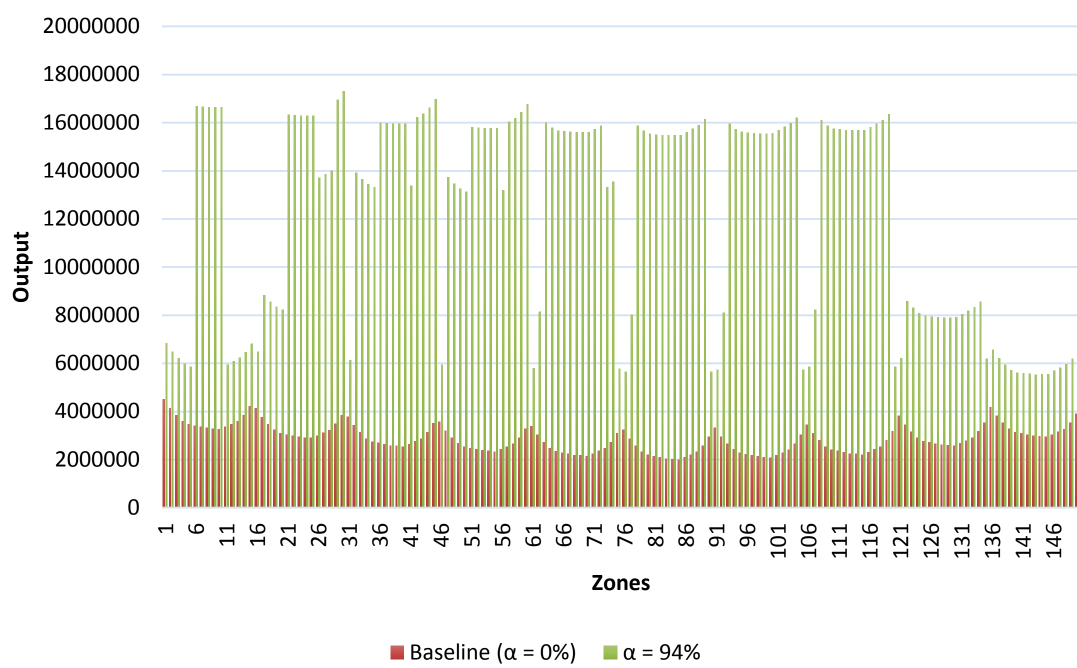


Fig. 94. Baseline Energy vs. Energy with User Preference α = 0.94 (Model #1)

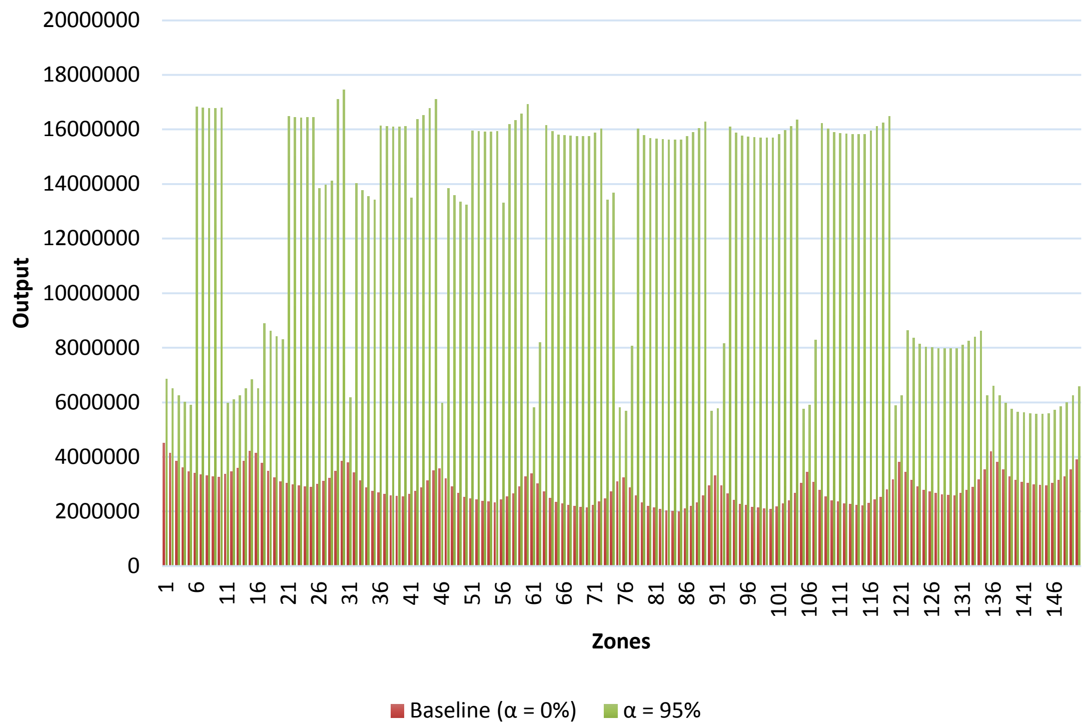


Fig. 95. Baseline Energy vs. Energy with User Preference α = 0.95 (Model #1)

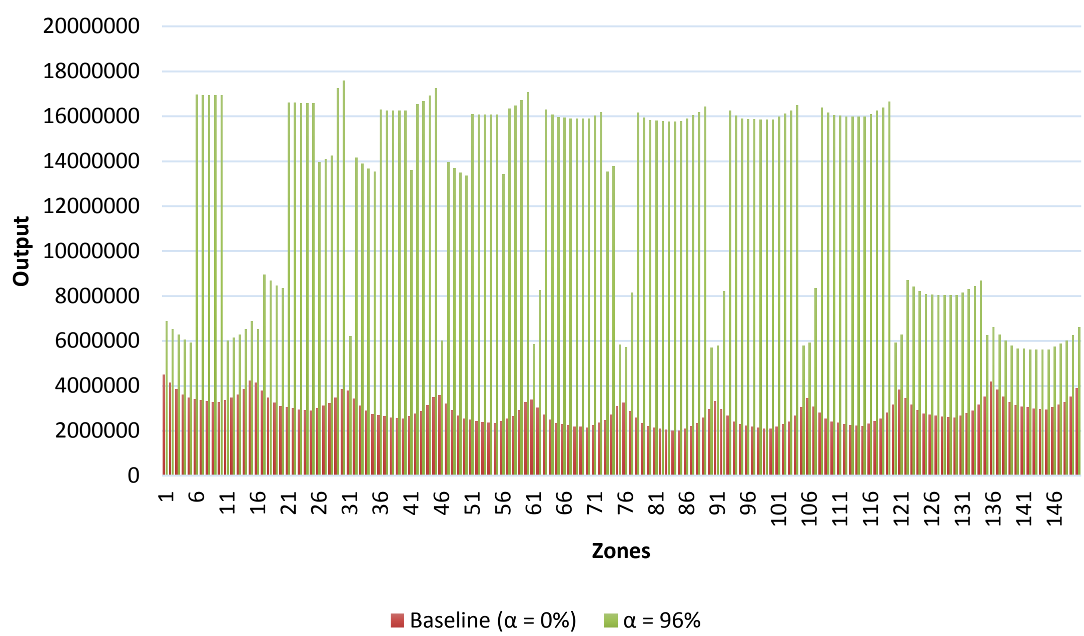


Fig. 96. Baseline Energy vs. Energy with User Preference α = 0.96 (Model #1)

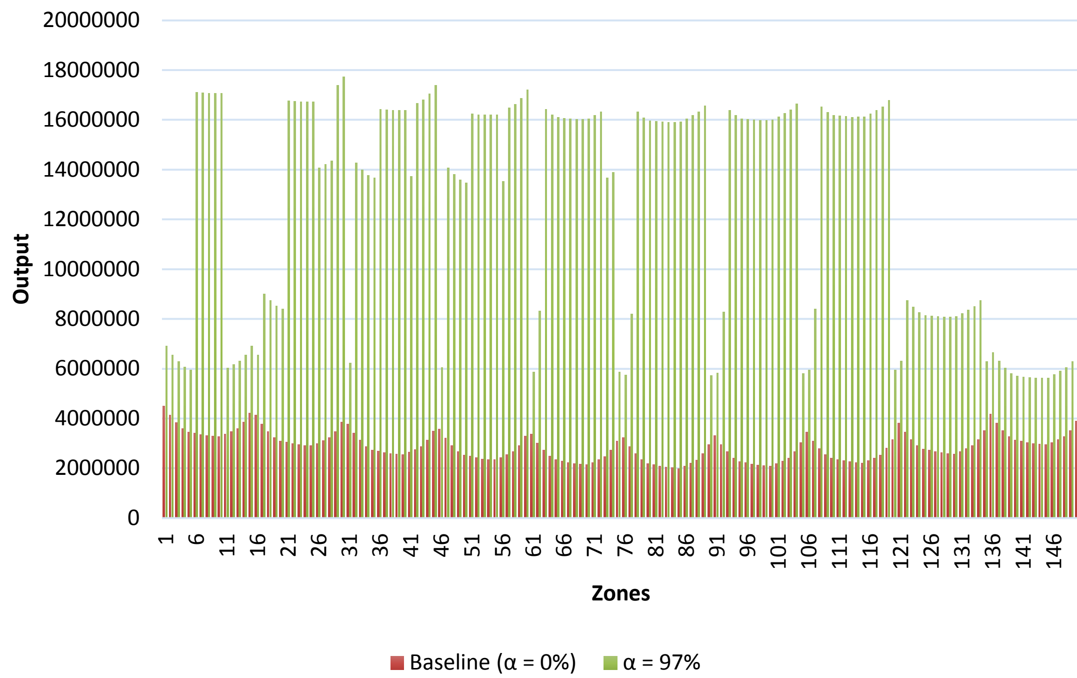


Fig. 97. Baseline Energy vs. Energy with User Preference α = 0.97 (Model #1)

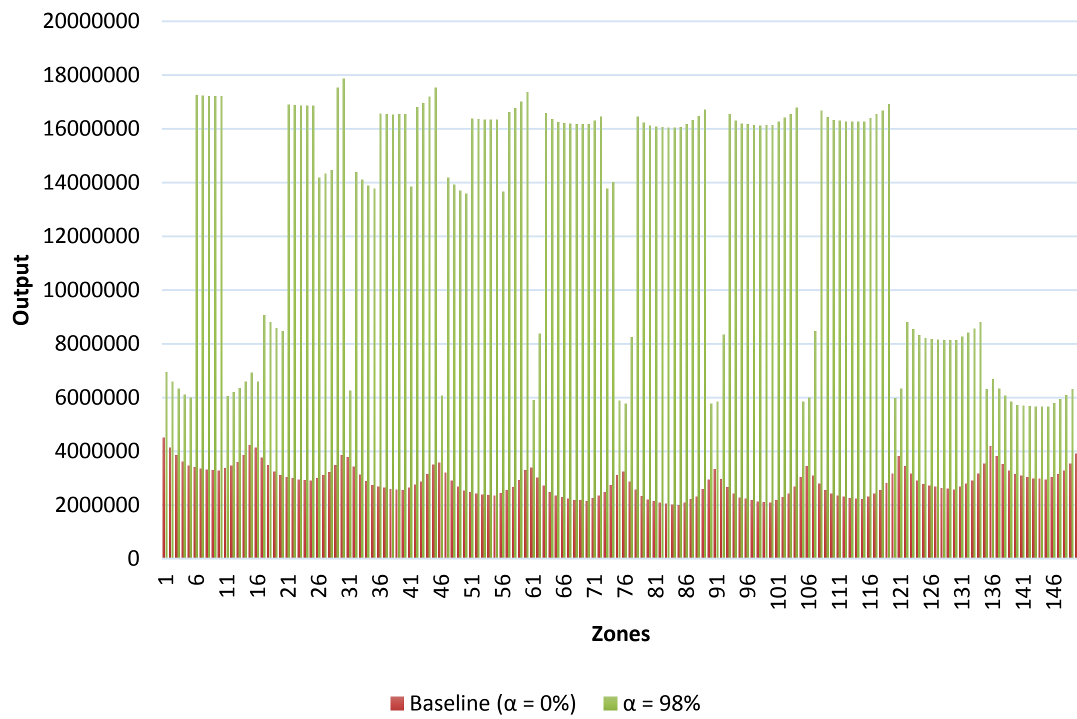


Fig. 98. Baseline Energy vs. Energy with User Preference α = 0.98 (Model #1)

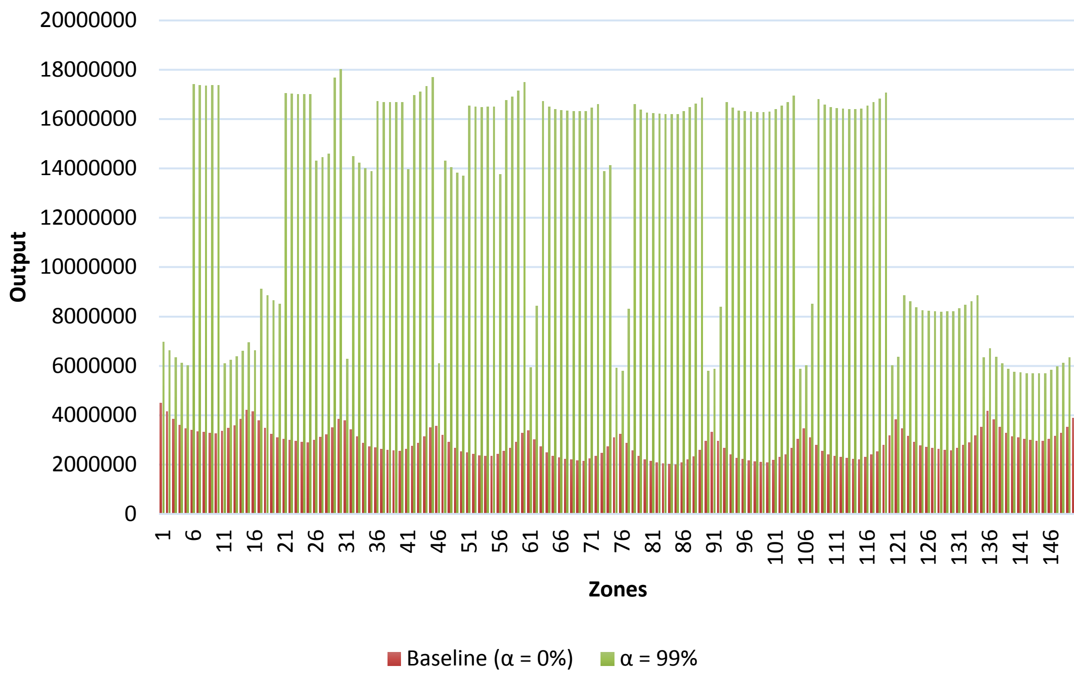


Fig. 99. Baseline Energy vs. Energy with User Preference α = 0.99 (Model #1)

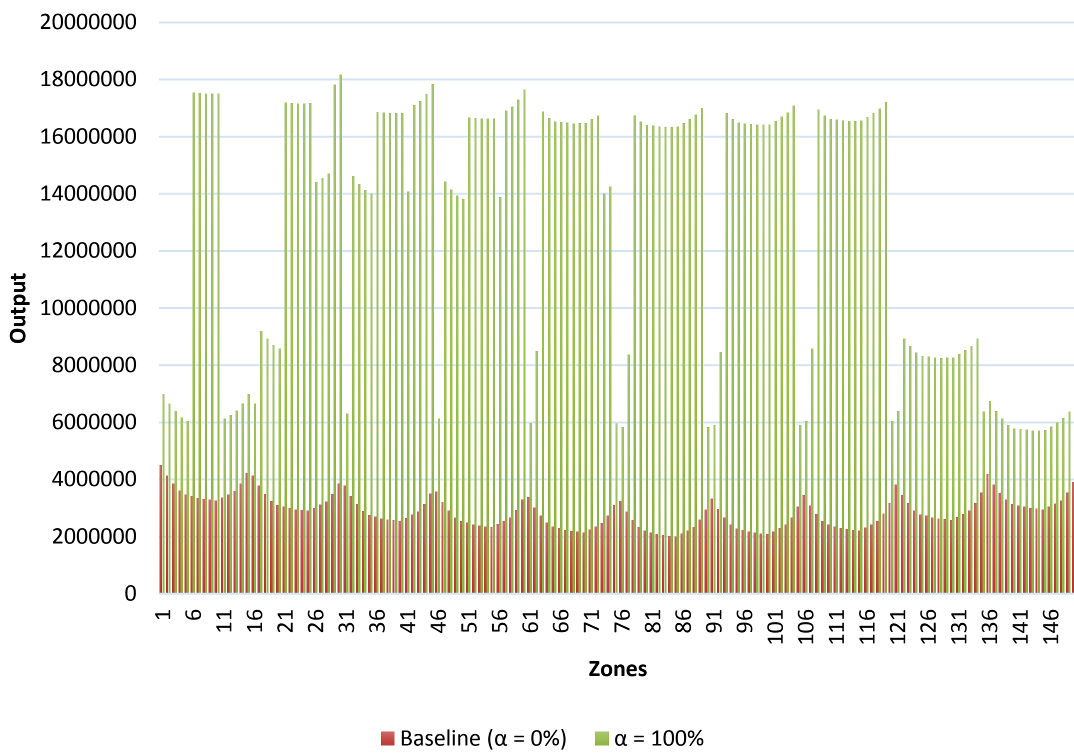


Fig. 100. Baseline Energy vs. Energy with User Preference α = 1 (Model #1)

APPENDIX B

RESULTS OF SENSITIVITY ANALYSIS ON OPTIMAL LOCATION (MODEL #2)

Supplementary Data File

Description:

The accompanying graphs show the results of the sensitivity analysis performed on the **second** model (Model #2) introduced. A sensitivity analysis is conducted to see the relationship between the energy consumption of the air compressor and the user's location preference. The sensitivity analysis is conducted on all possible weights of the user preference.

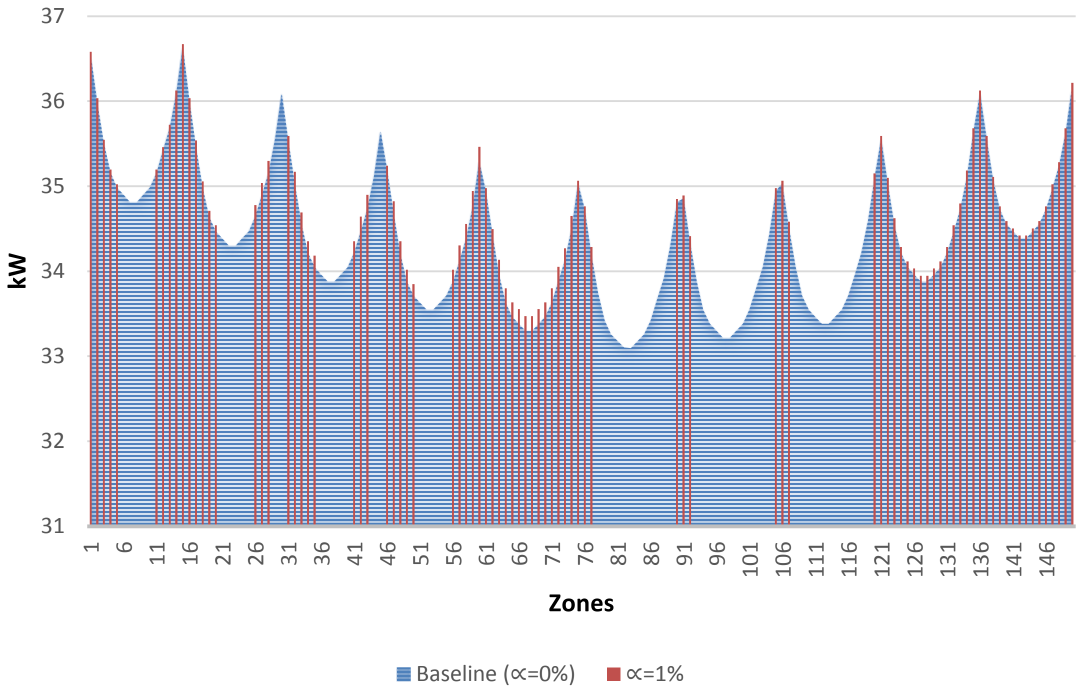


Fig. 1. Baseline Energy vs. Energy with User Preference α = 0.01 (Model #2)

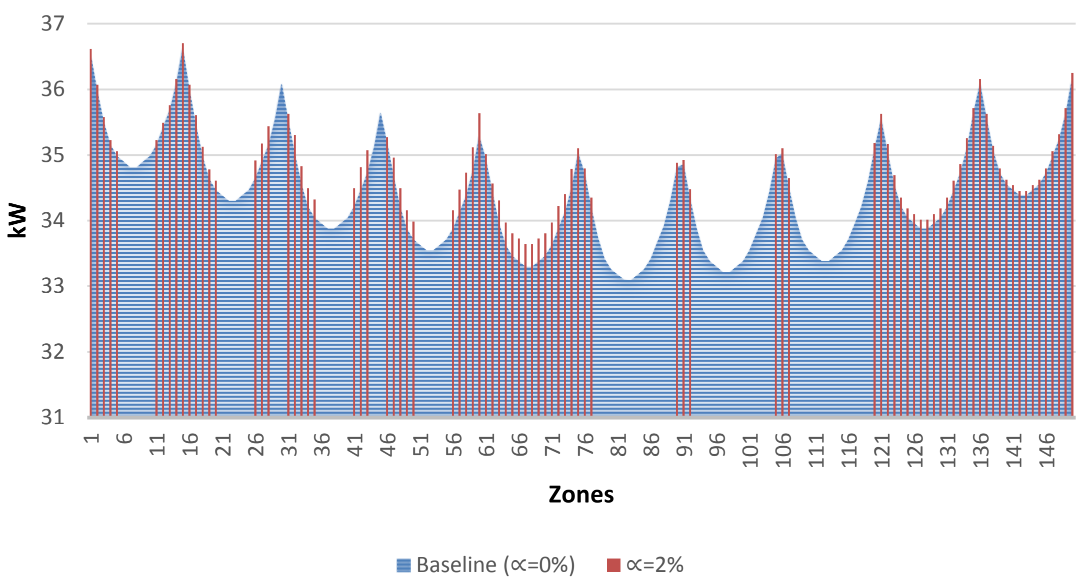


Fig. 2. Baseline Energy vs. Energy with User Preference α = 0.02 (Model #2)

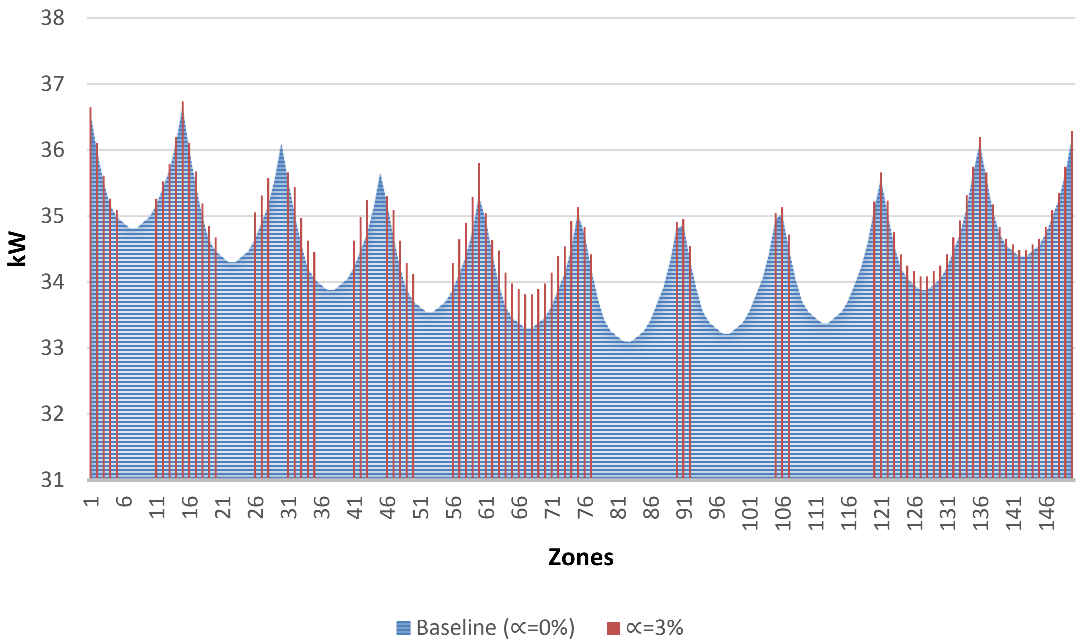


Fig. 3. Baseline Energy vs. Energy with User Preference α = 0.03 (Model #2)

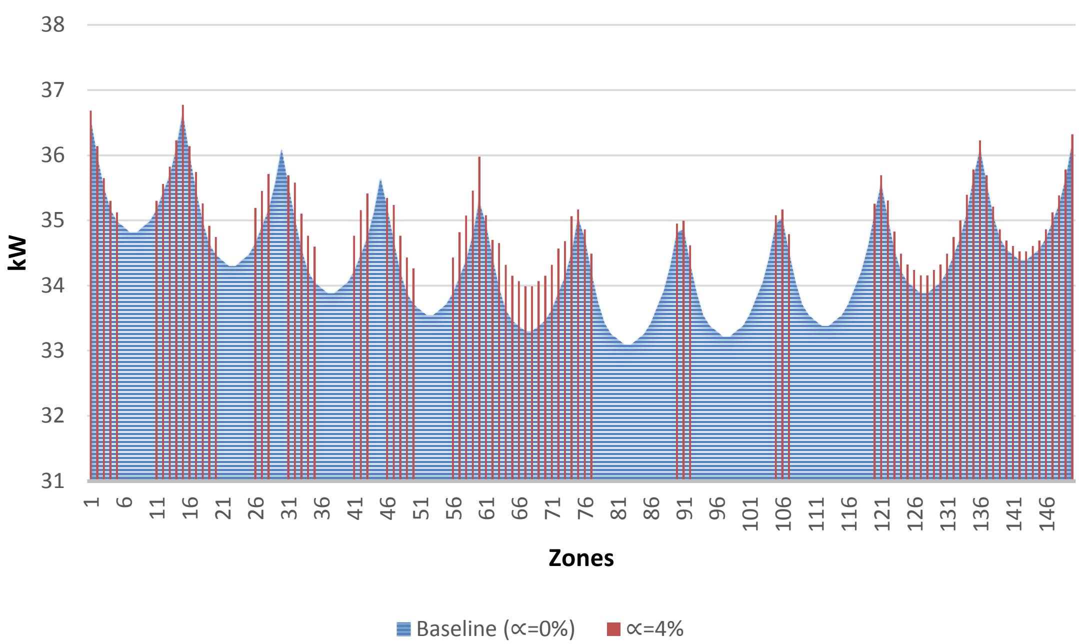


Fig. 4. Baseline Energy vs. Energy with User Preference α = 0.04 (Model #2)

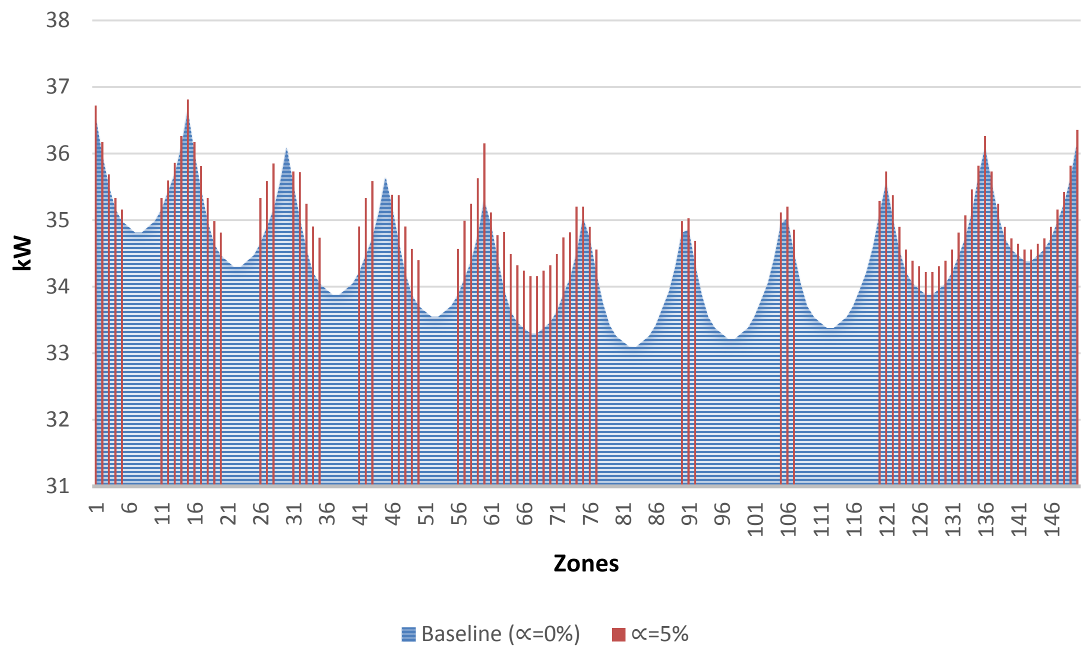


Fig. 5. Baseline Energy vs. Energy with User Preference α = 0.05 (Model #2)

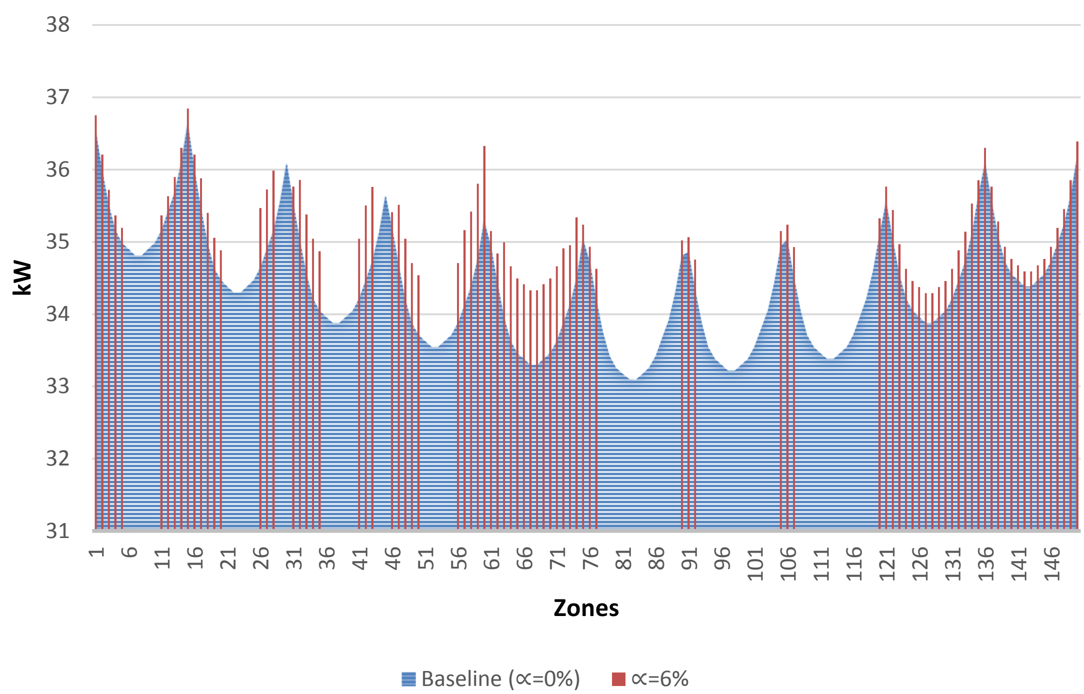


Fig. 6. Baseline Energy vs. Energy with User Preference α = 0.06 (Model #2)

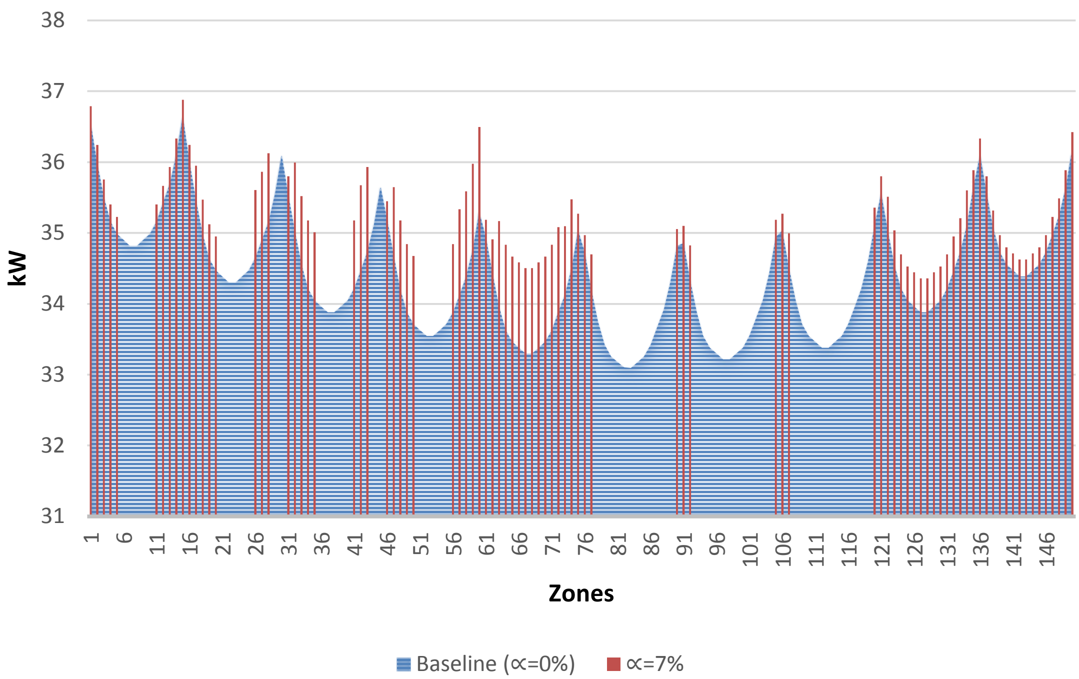


Fig. 7. Baseline Energy vs. Energy with User Preference α = 0.07 (Model #2)

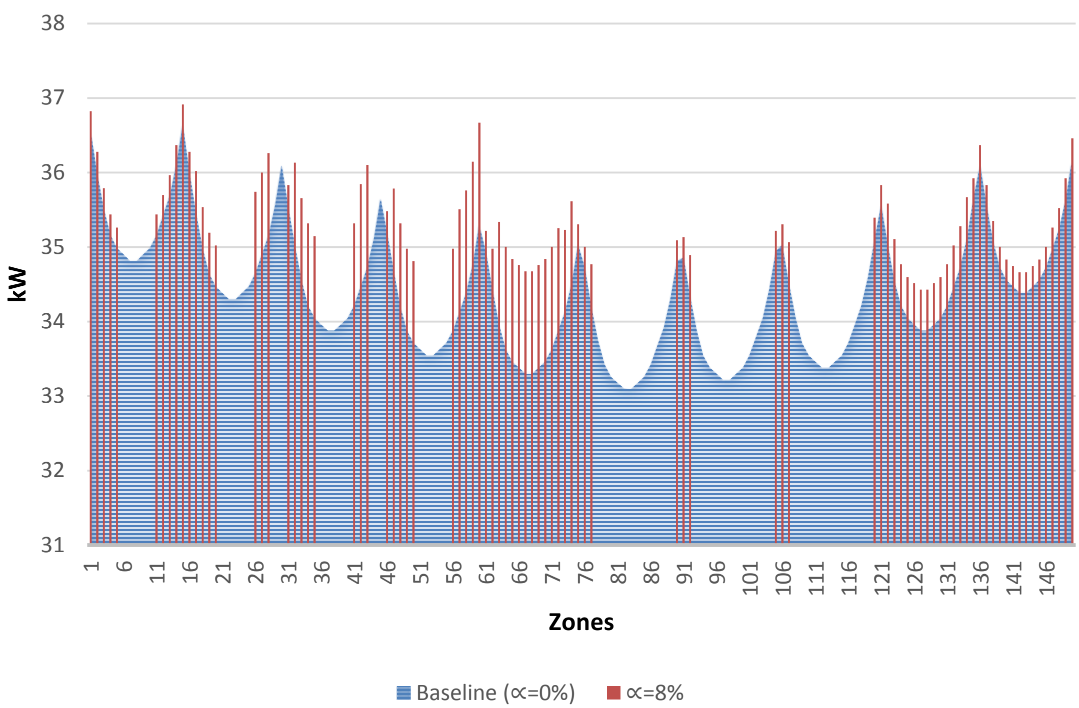


Fig. 8. Baseline Energy vs. Energy with User Preference α = 0.08 (Model #2)

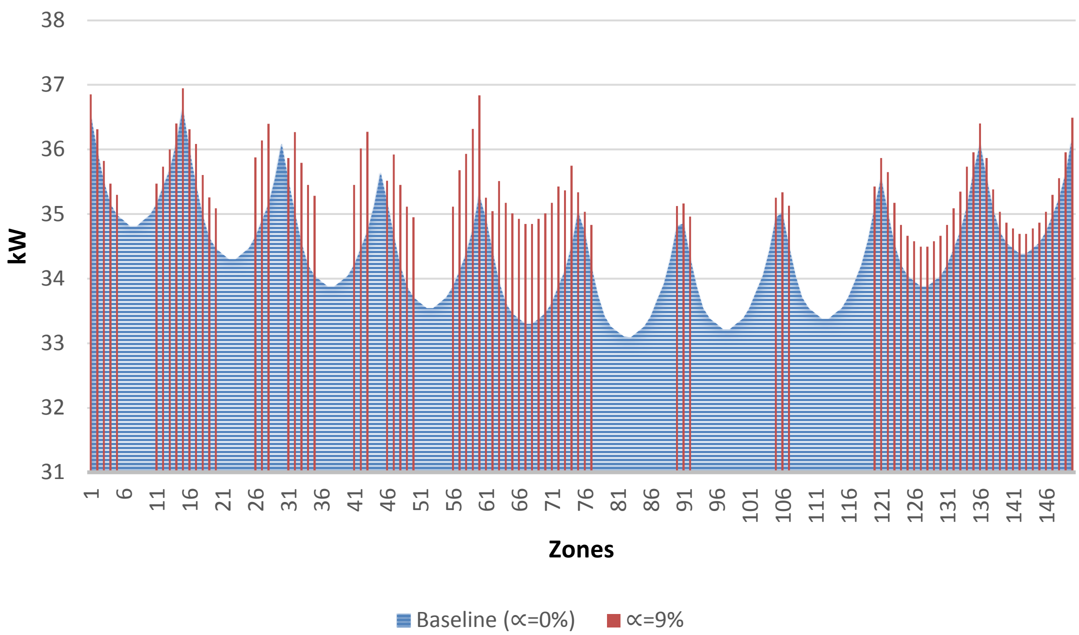


Fig. 9. Baseline Energy vs. Energy with User Preference α = 0.09 (Model #2)

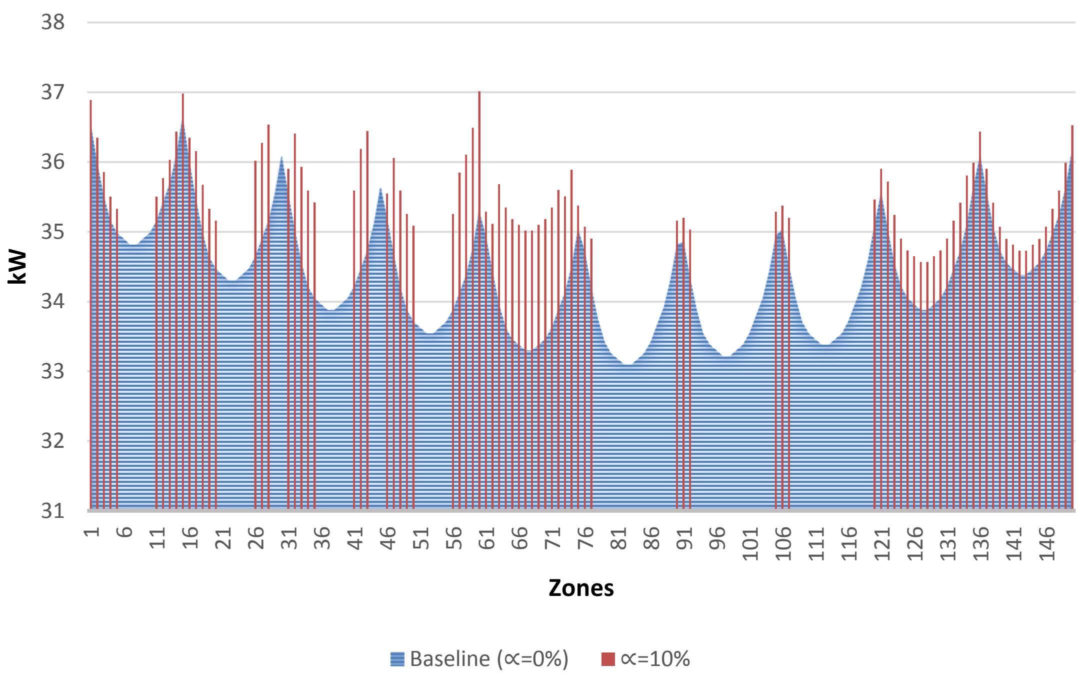


Fig. 10. Baseline Energy vs. Energy with User Preference α = 0.1 (Model #2)

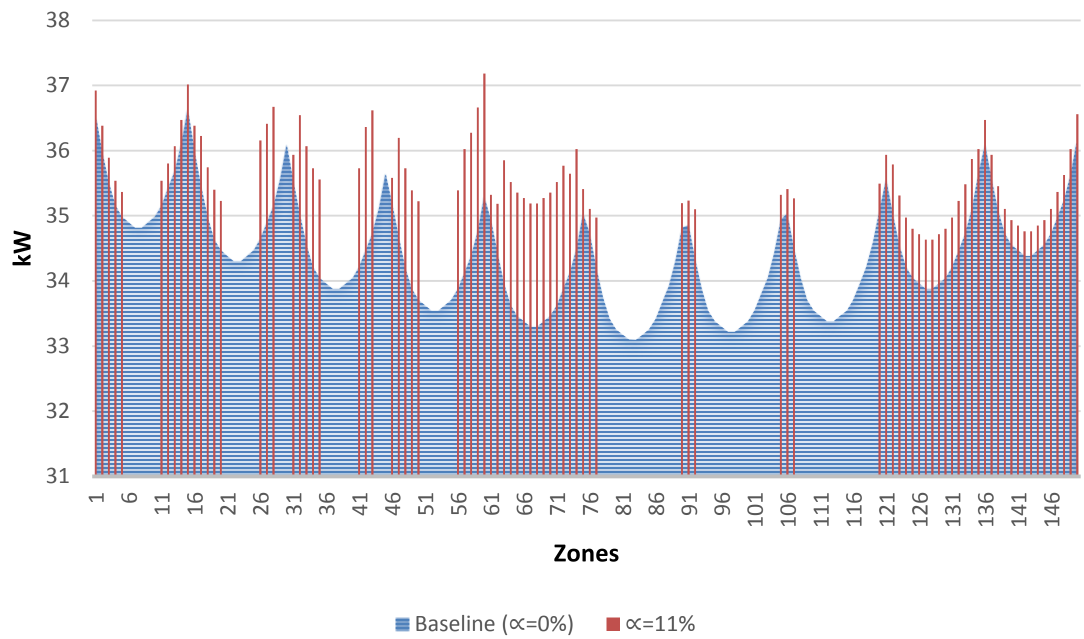


Fig. 11. Baseline Energy vs. Energy with User Preference α = 0.11 (Model #2)

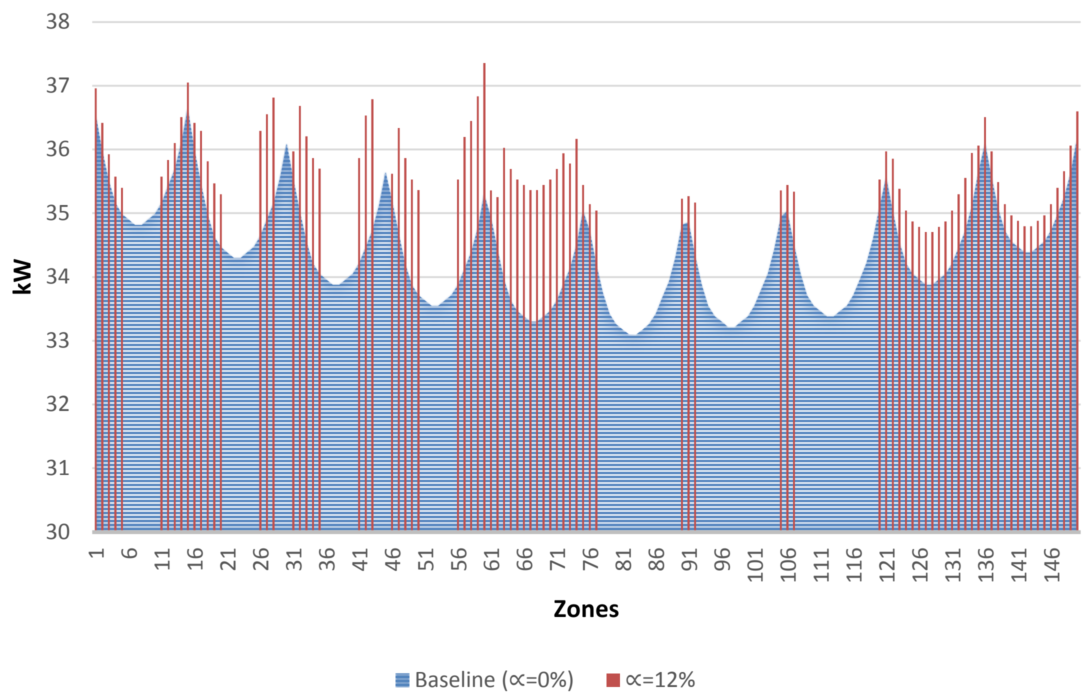


Fig. 12. Baseline Energy vs. Energy with User Preference α = 0.12 (Model #2)

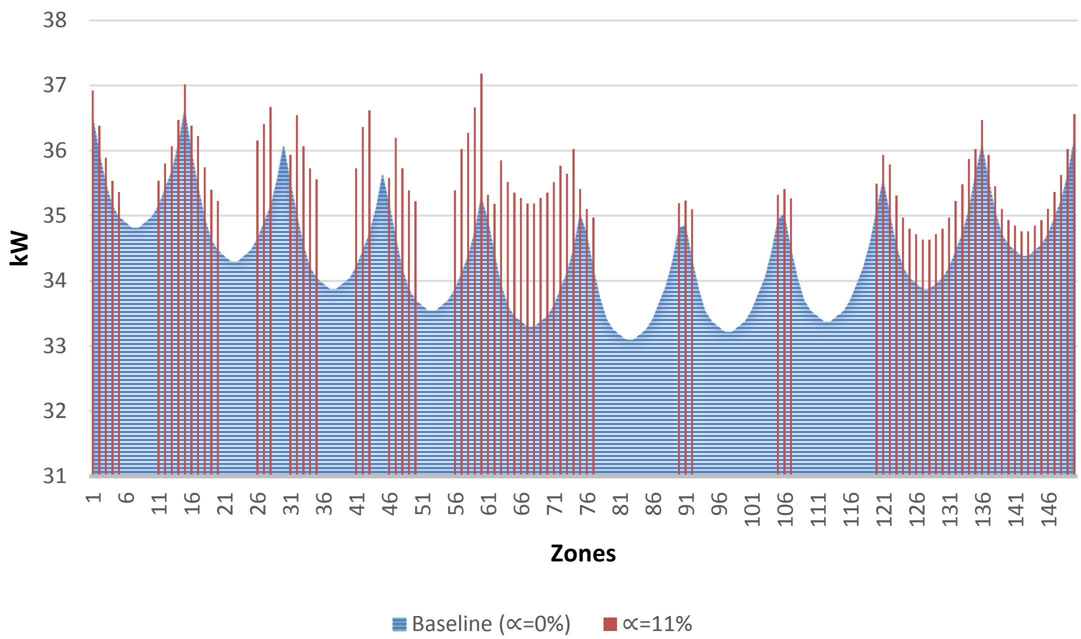


Fig. 13. Baseline Energy vs. Energy with User Preference α = 0.13 (Model #2)

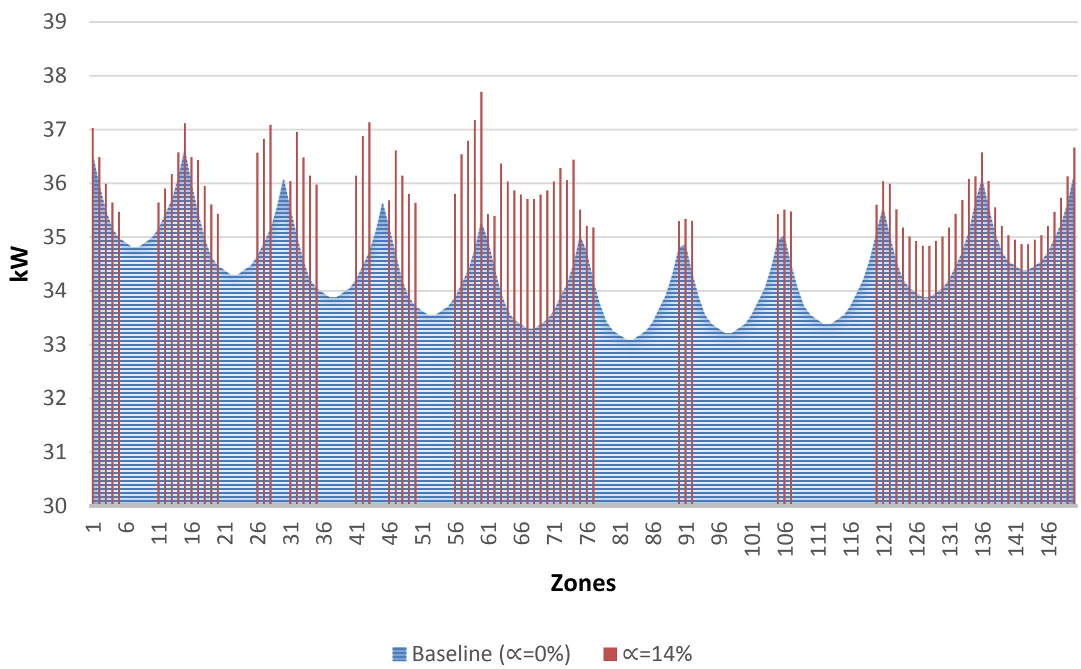


Fig. 14. Baseline Energy vs. Energy with User Preference α = 0.14 (Model #2)

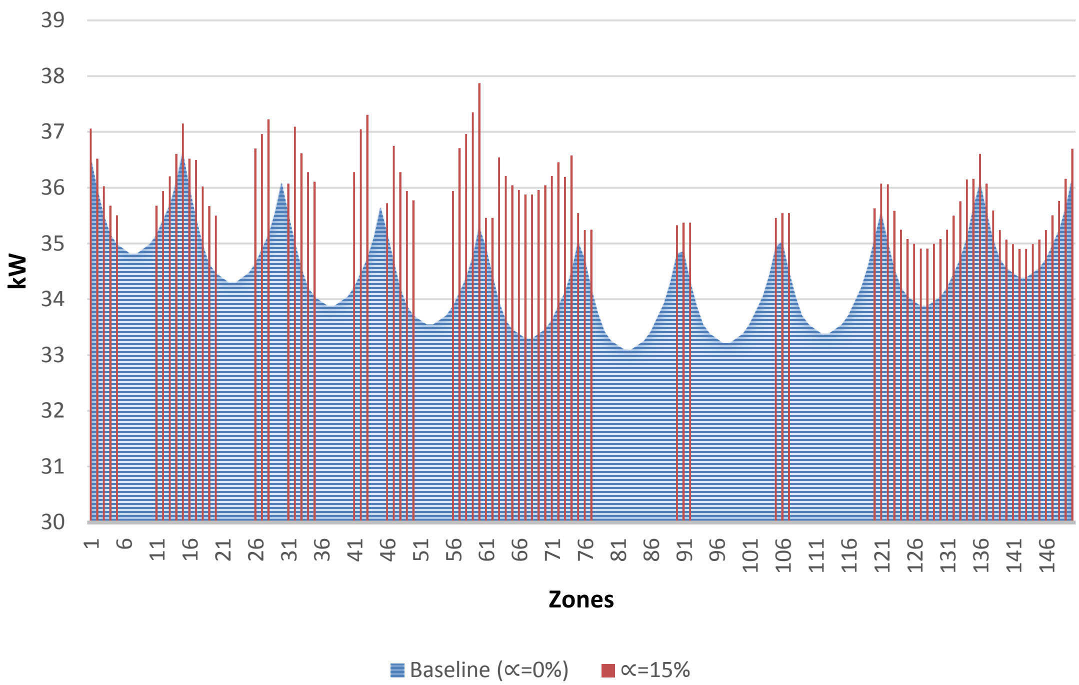


Fig. 15. Baseline Energy vs. Energy with User Preference α = 0.15 (Model #2)

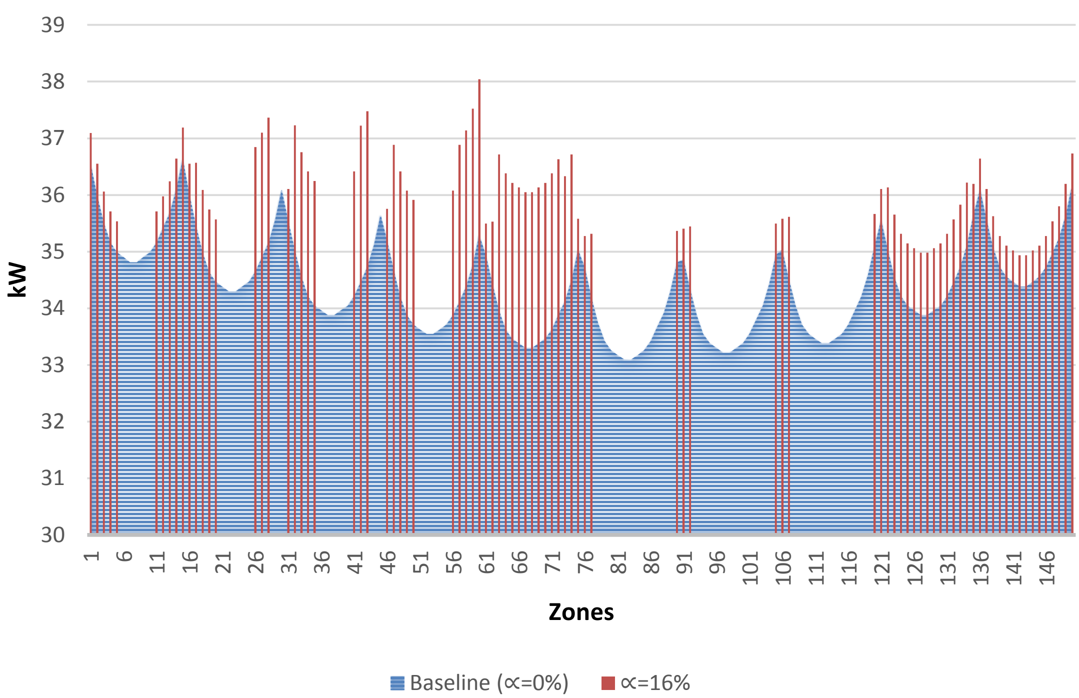


Fig. 16. Baseline Energy vs. Energy with User Preference α = 0.16 (Model #2)

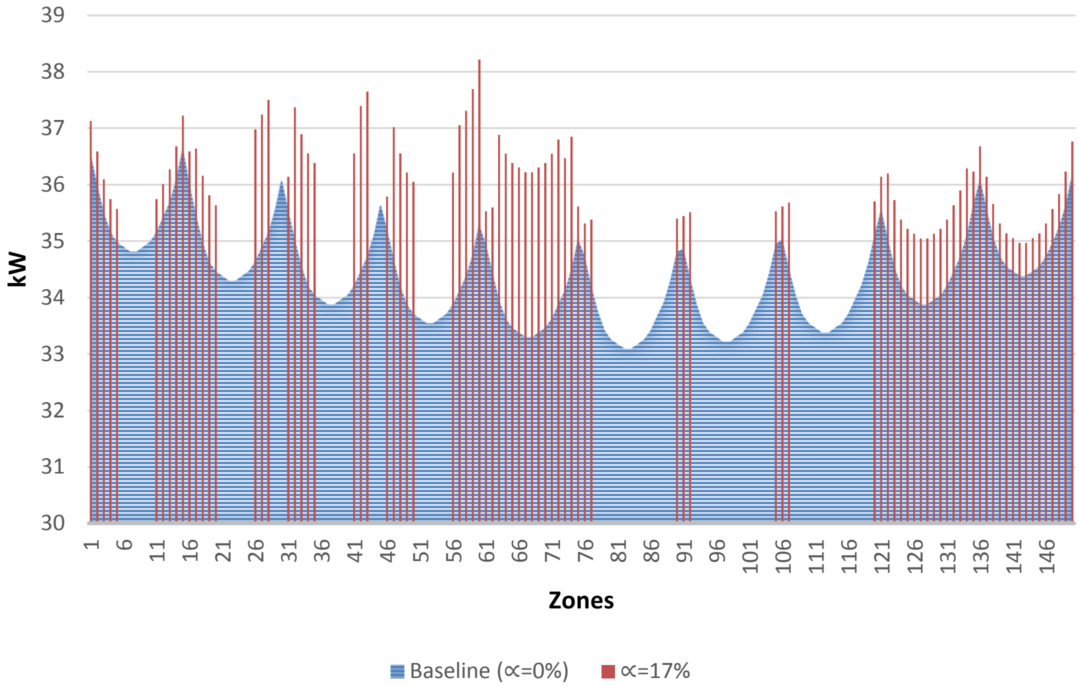


Fig. 17. Baseline Energy vs. Energy with User Preference α = 0.17 (Model #2)

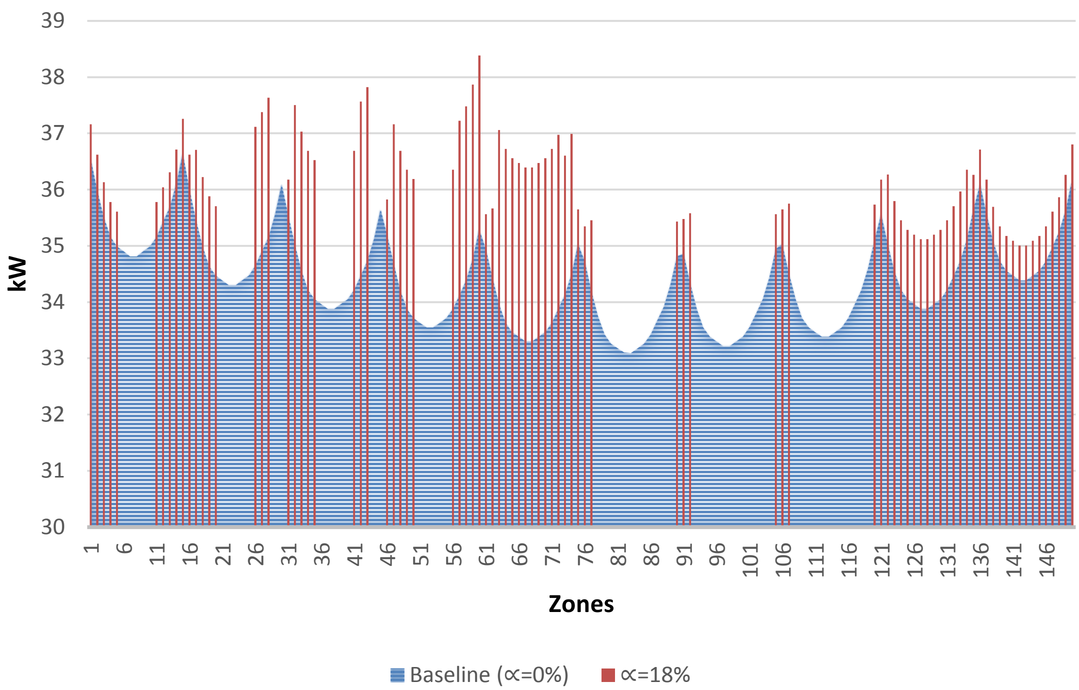


Fig. 18. Baseline Energy vs. Energy with User Preference α = 0.18 (Model #2)

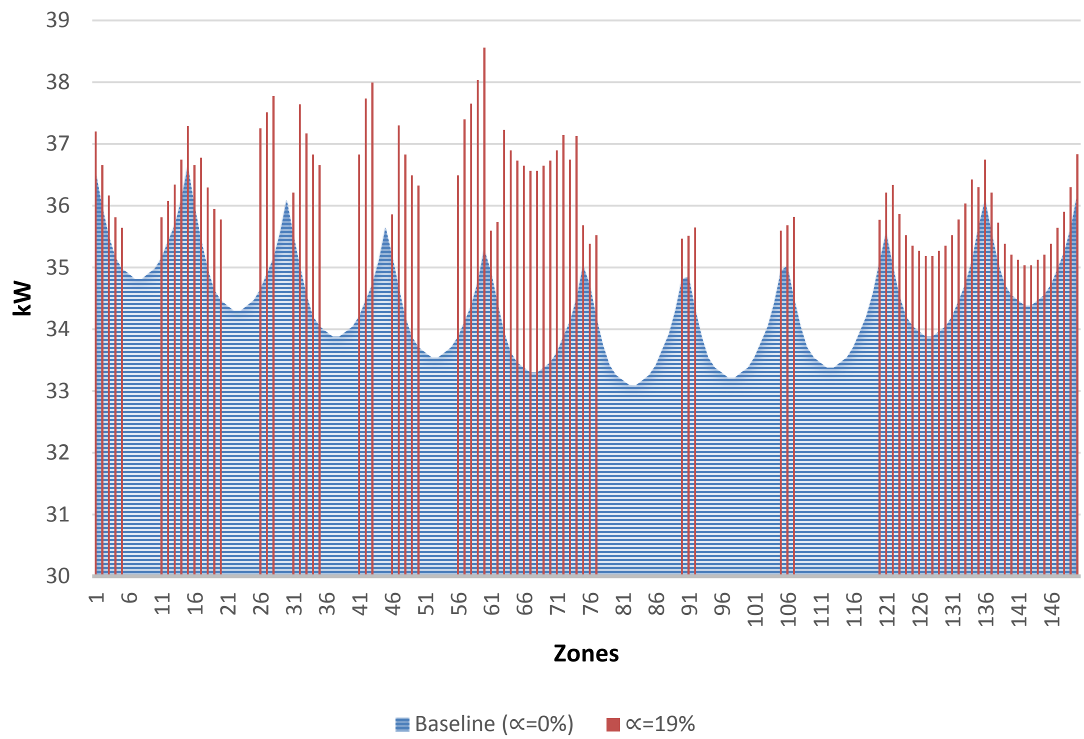


Fig. 19. Baseline Energy vs. Energy with User Preference α = 0.19 (Model #2)

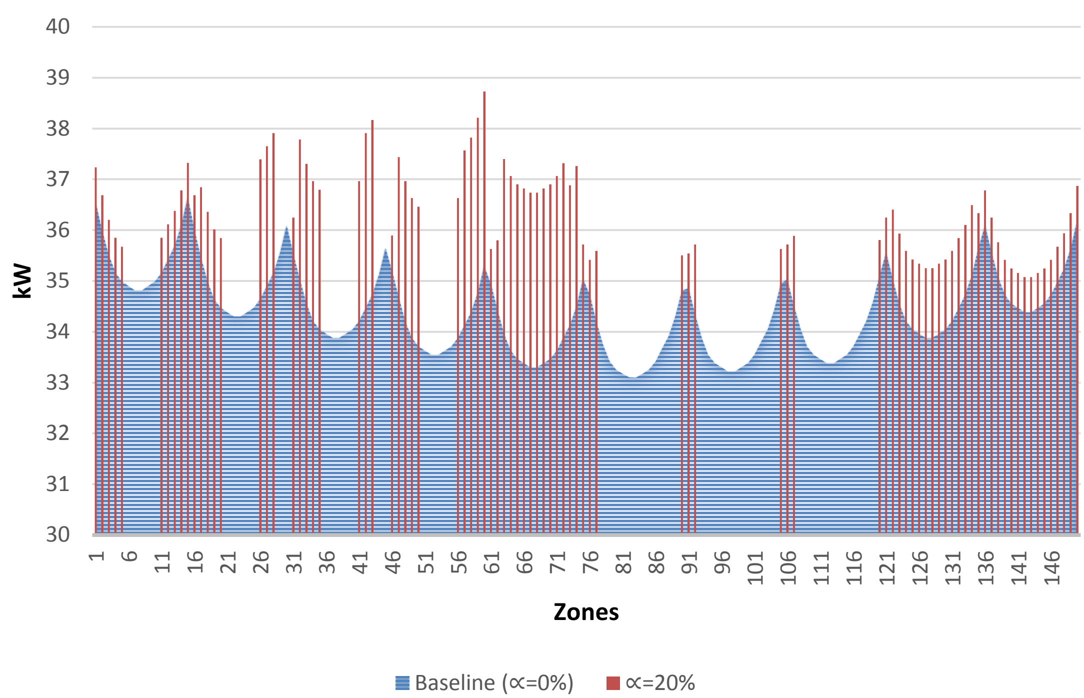


Fig. 20. Baseline Energy vs. Energy with User Preference α = 0.20 (Model #2)

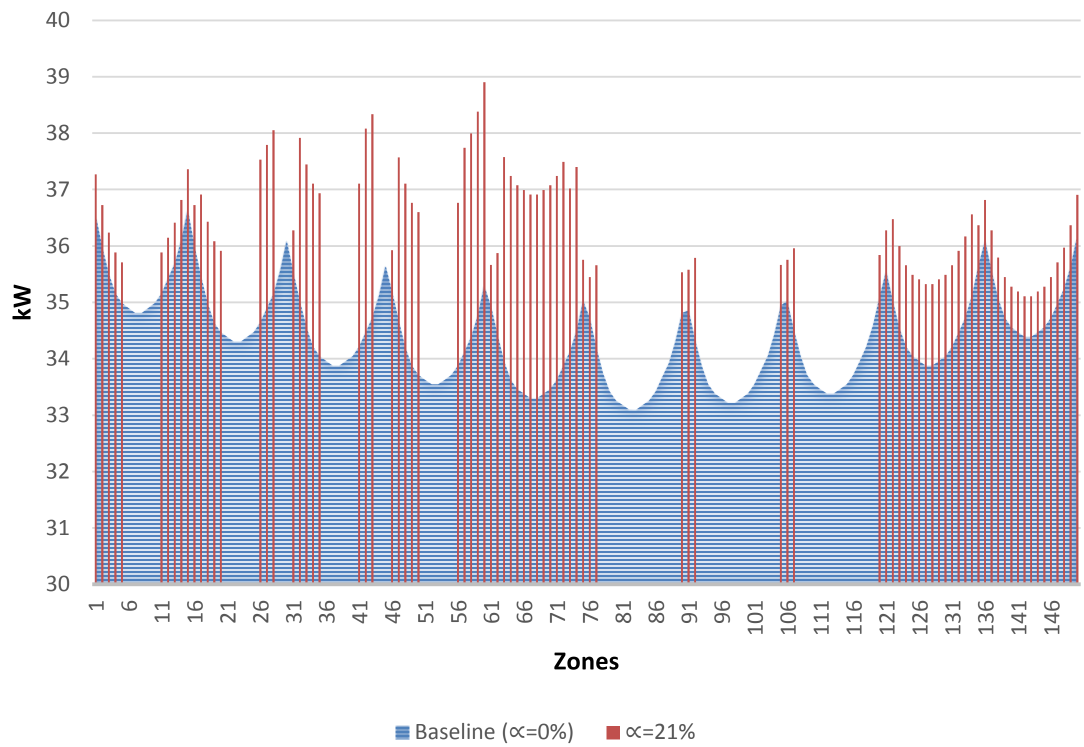


Fig. 21. Baseline Energy vs. Energy with User Preference α = 0.21 (Model #2)

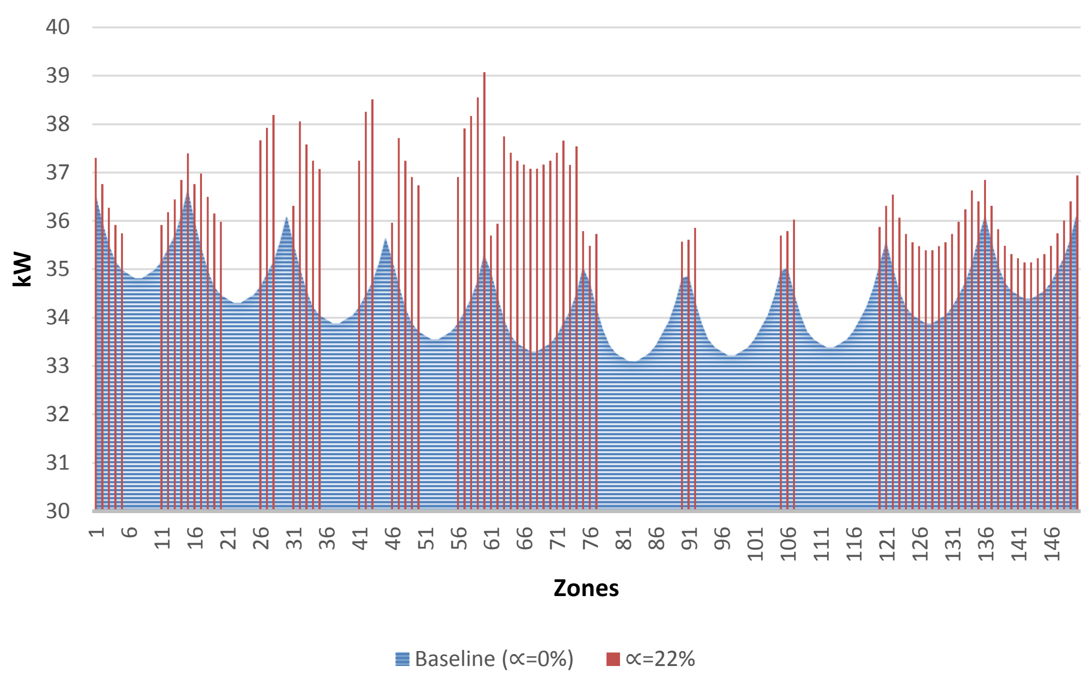


Fig. 22. Baseline Energy vs. Energy with User Preference α = 0.22 (Model #2)

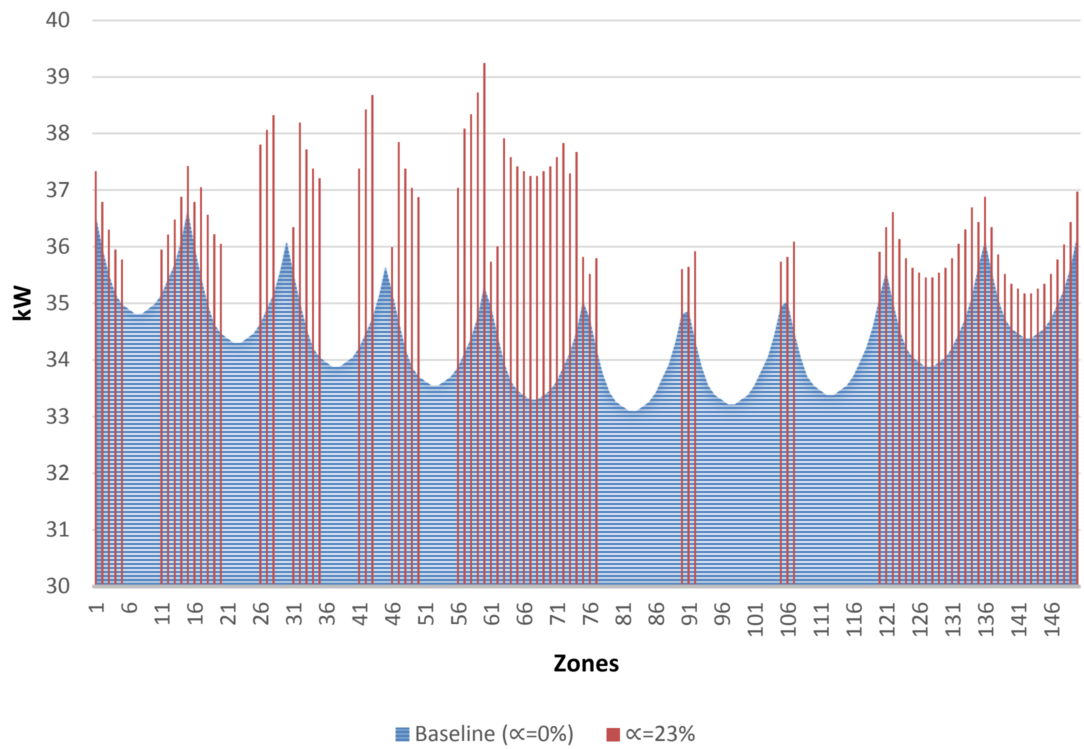


Fig. 23. Baseline Energy vs. Energy with User Preference α = 0.23 (Model #2)

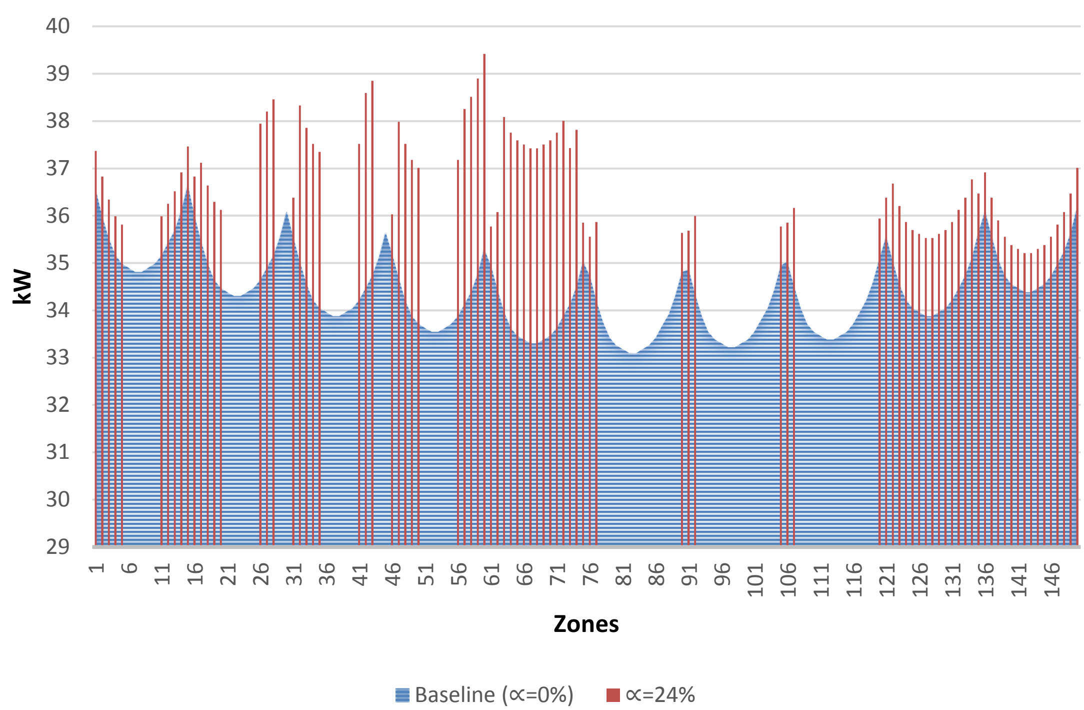


Fig. 24. Baseline Energy vs. Energy with User Preference α = 0.24 (Model #2)

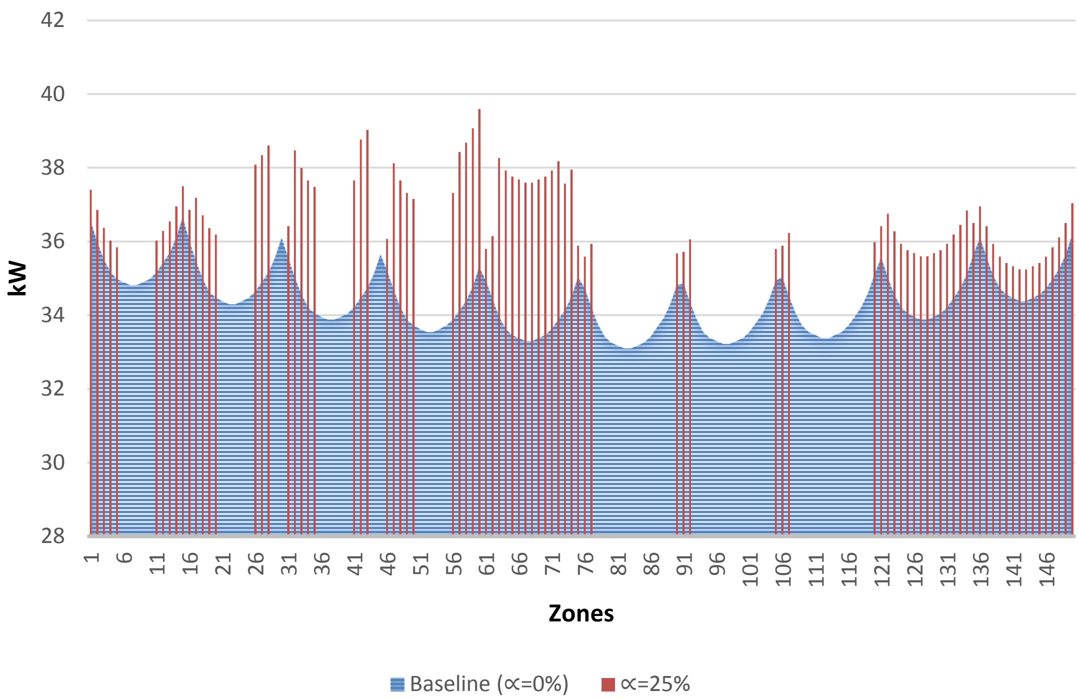


Fig. 25. Baseline Energy vs. Energy with User Preference α = 0.25 (Model #2)

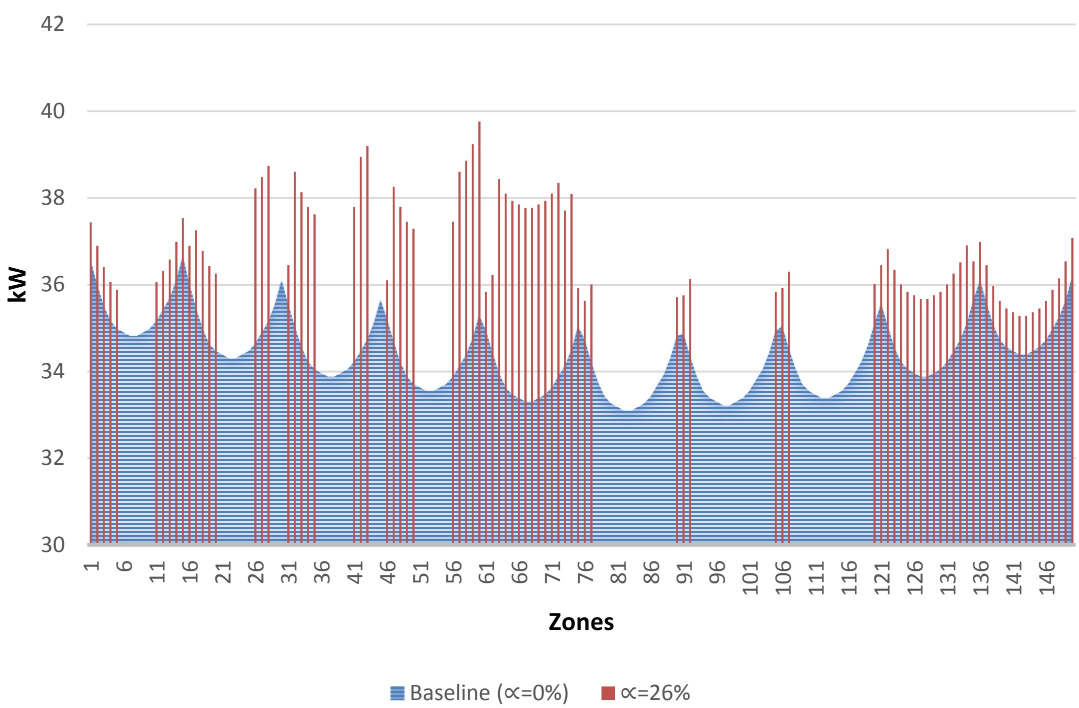


Fig. 26. Baseline Energy vs. Energy with User Preference α = 0.26 (Model #2)

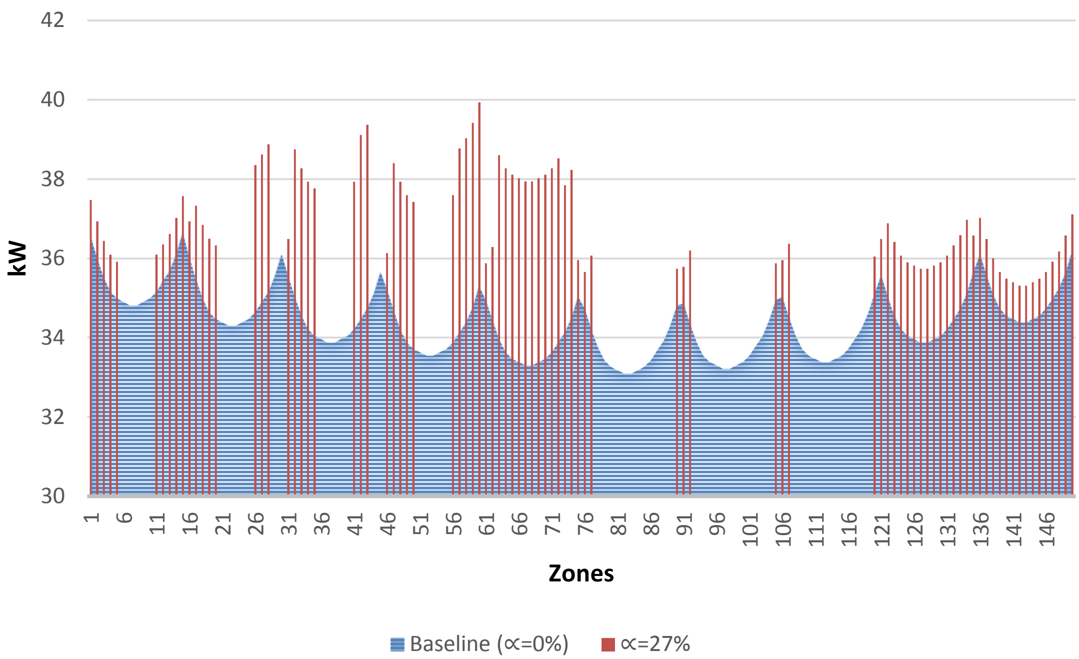


Fig. 27. Baseline Energy vs. Energy with User Preference α = 0.28 (Model #2)

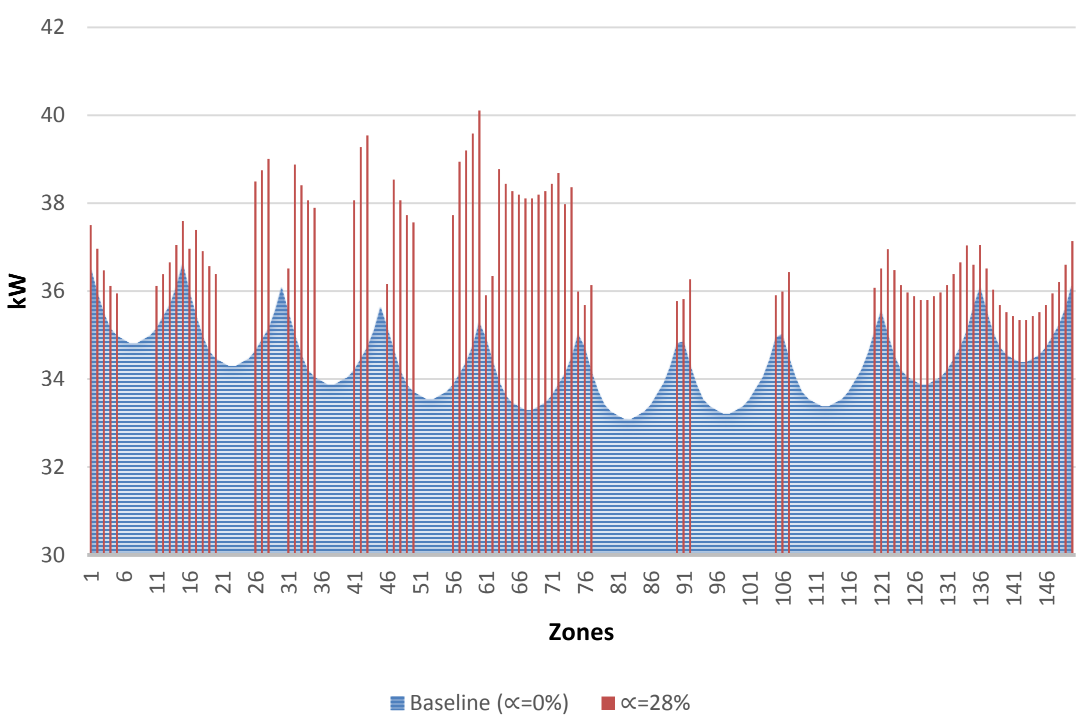


Fig. 28. Baseline Energy vs. Energy with User Preference α = 0.28 (Model #2)

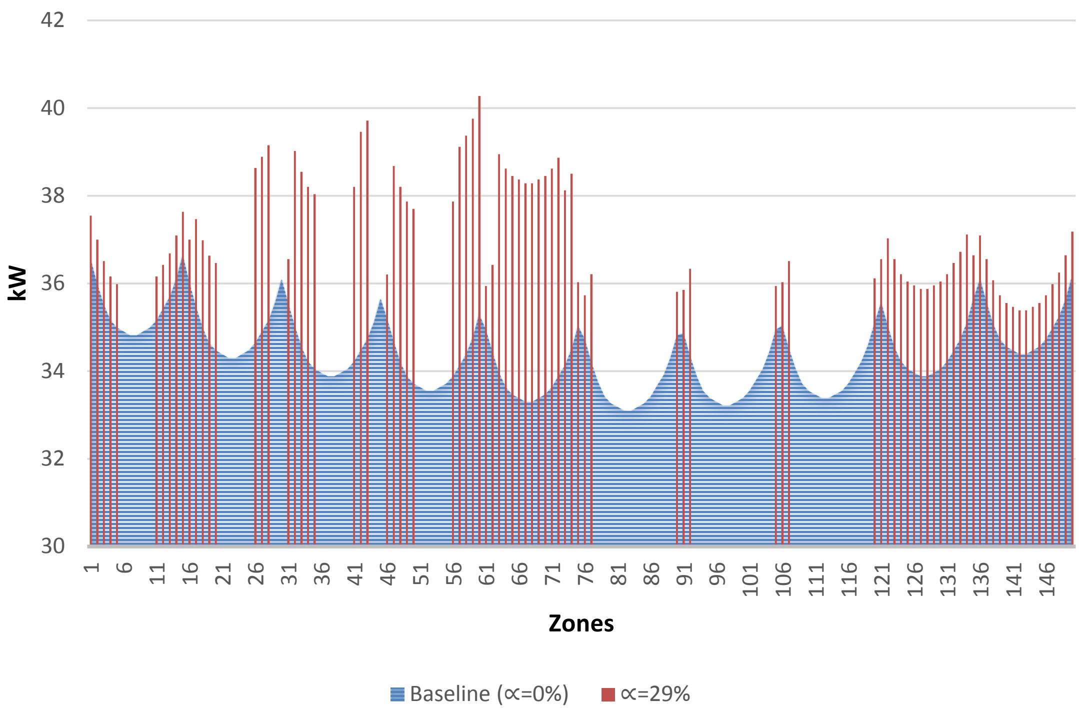


Fig. 29. Baseline Energy vs. Energy with User Preference α = 0.29 (Model #2)

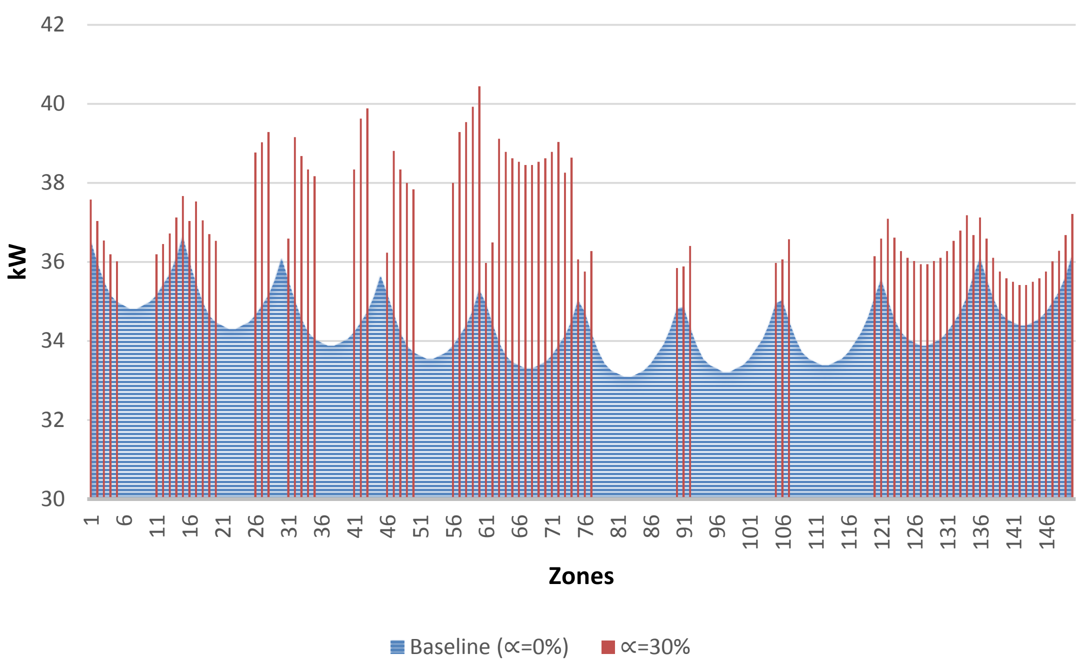


Fig. 30. Baseline Energy vs. Energy with User Preference α = 0.30 (Model #2)

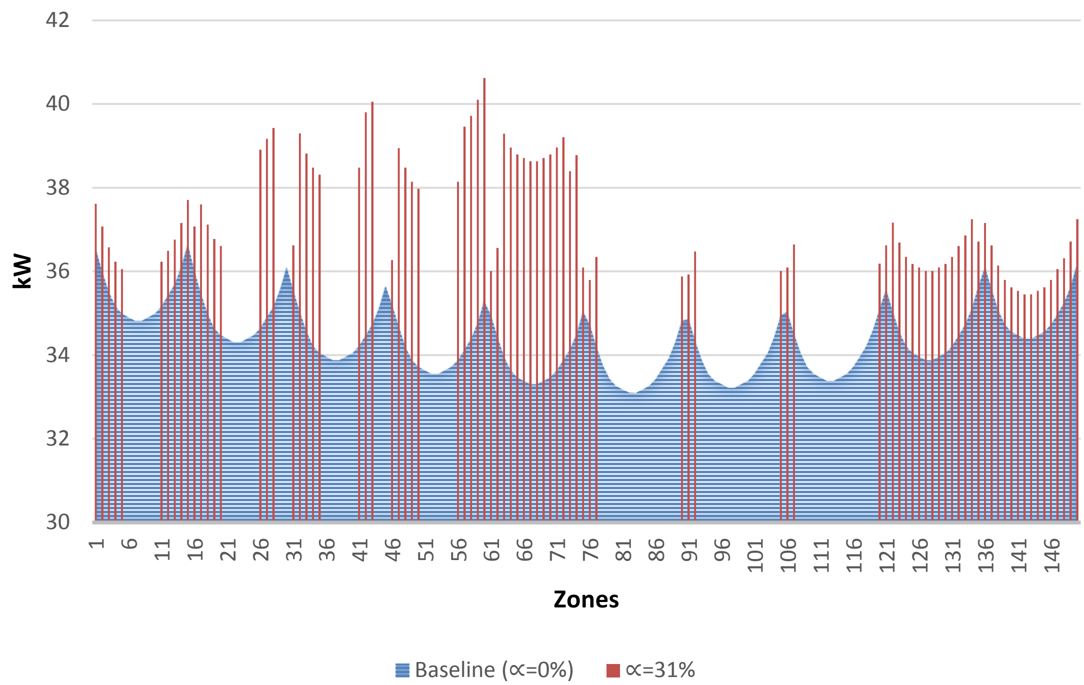


Fig. 31. Baseline Energy vs. Energy with User Preference α = 0.31 (Model #2)

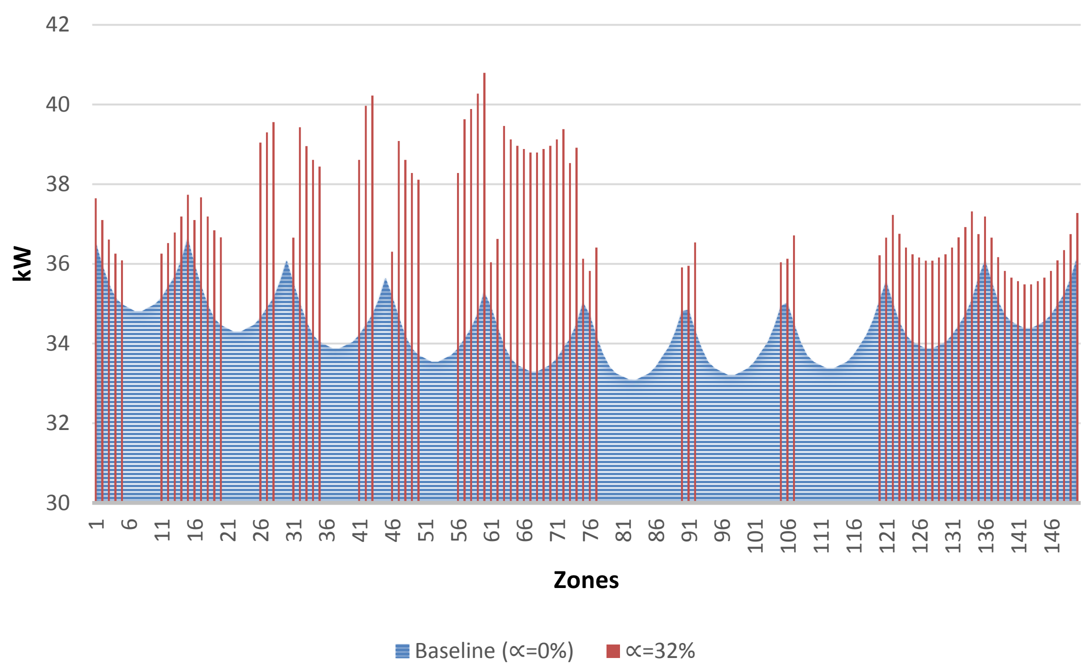


Fig. 32. Baseline Energy vs. Energy with User Preference α = 0.32 (Model #2)

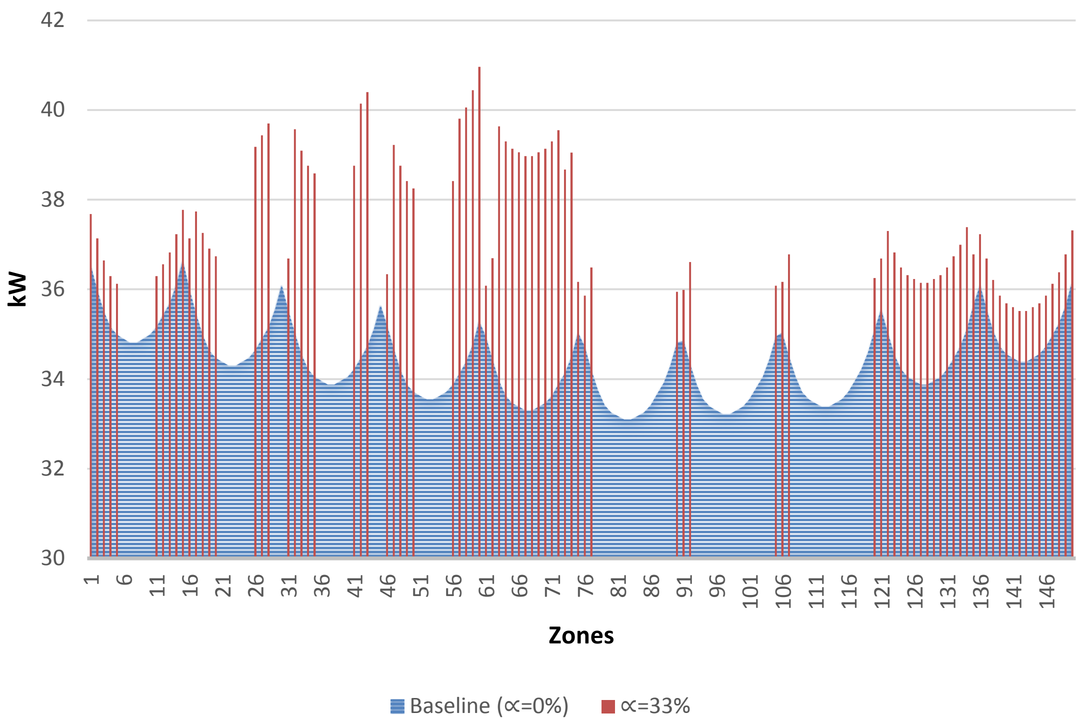


Fig. 33. Baseline Energy vs. Energy with User Preference α = 0.34 (Model #2)

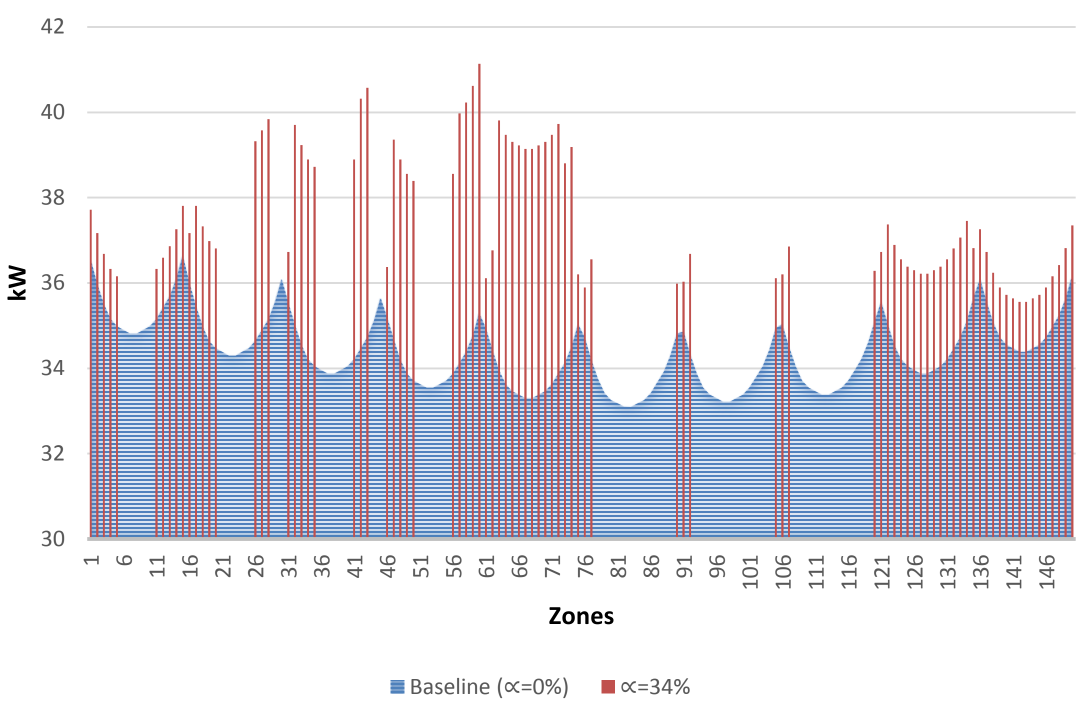


Fig. 34. Baseline Energy vs. Energy with User Preference α = 0.34 (Model #2)

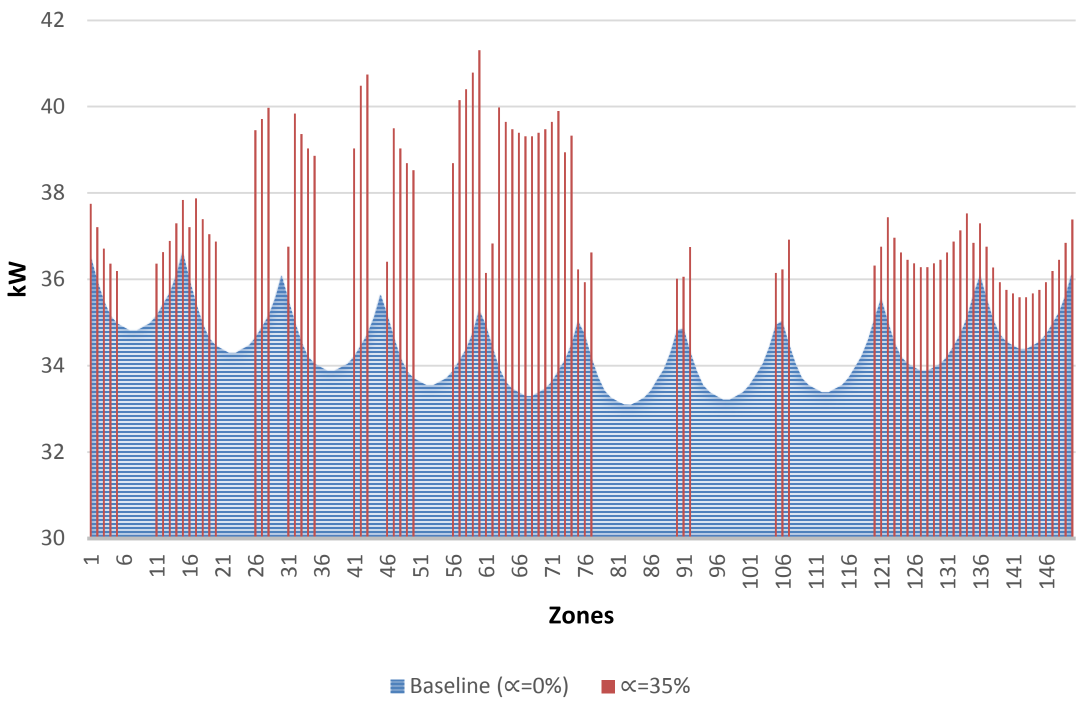


Fig. 35. Baseline Energy vs. Energy with User Preference α = 0.35 (Model #2)

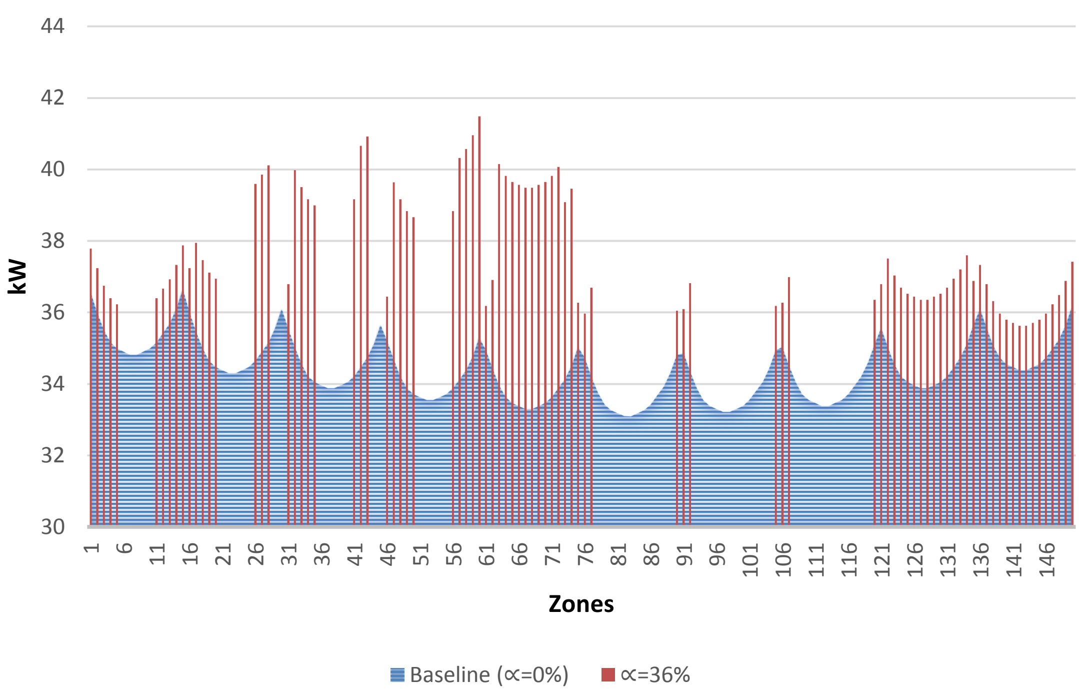


Fig. 36. Baseline Energy vs. Energy with User Preference α = 0.36 (Model #2)

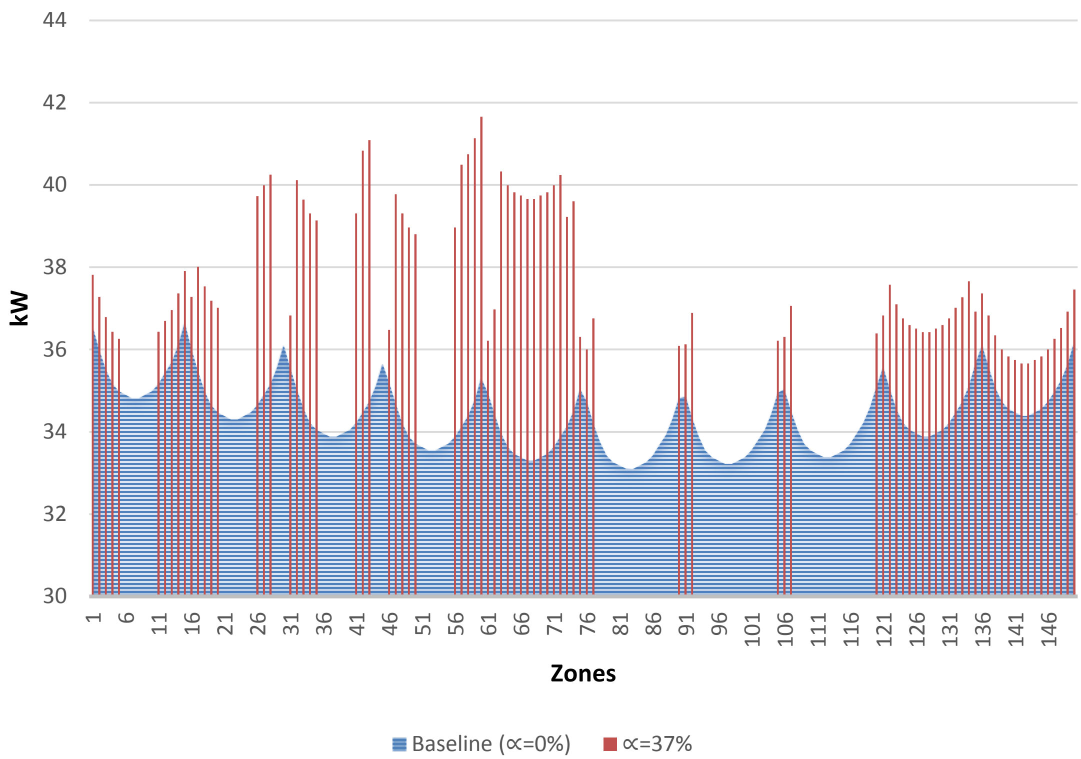


Fig. 37. Baseline Energy vs. Energy with User Preference α = 0.37 (Model #2)

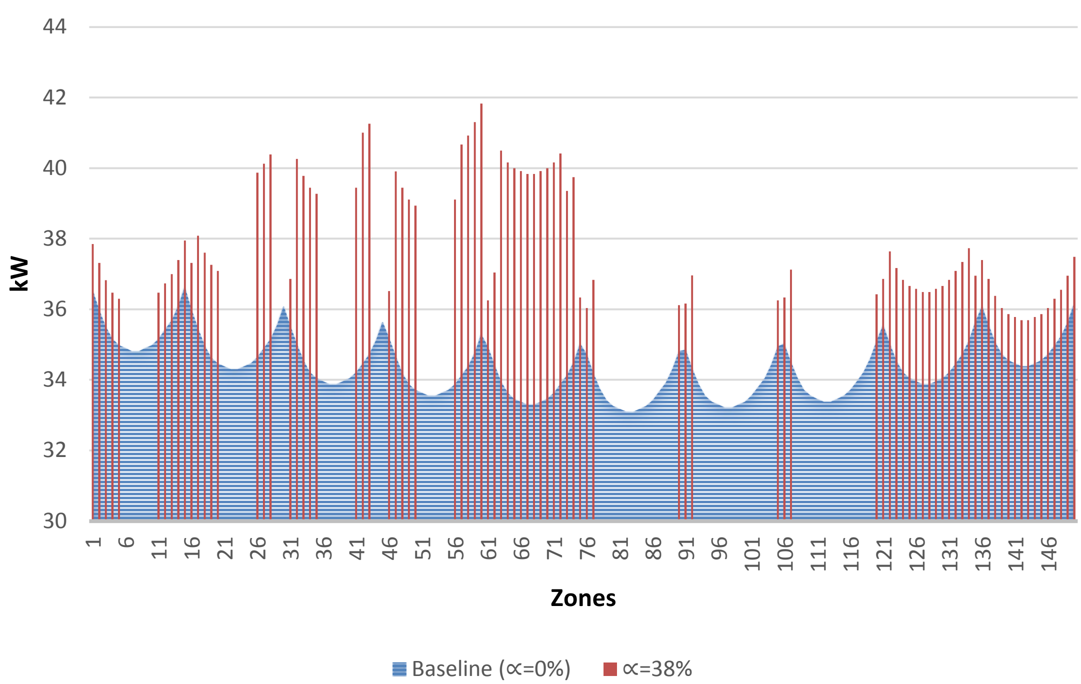


Fig. 38. Baseline Energy vs. Energy with User Preference α = 0.38 (Model #2)

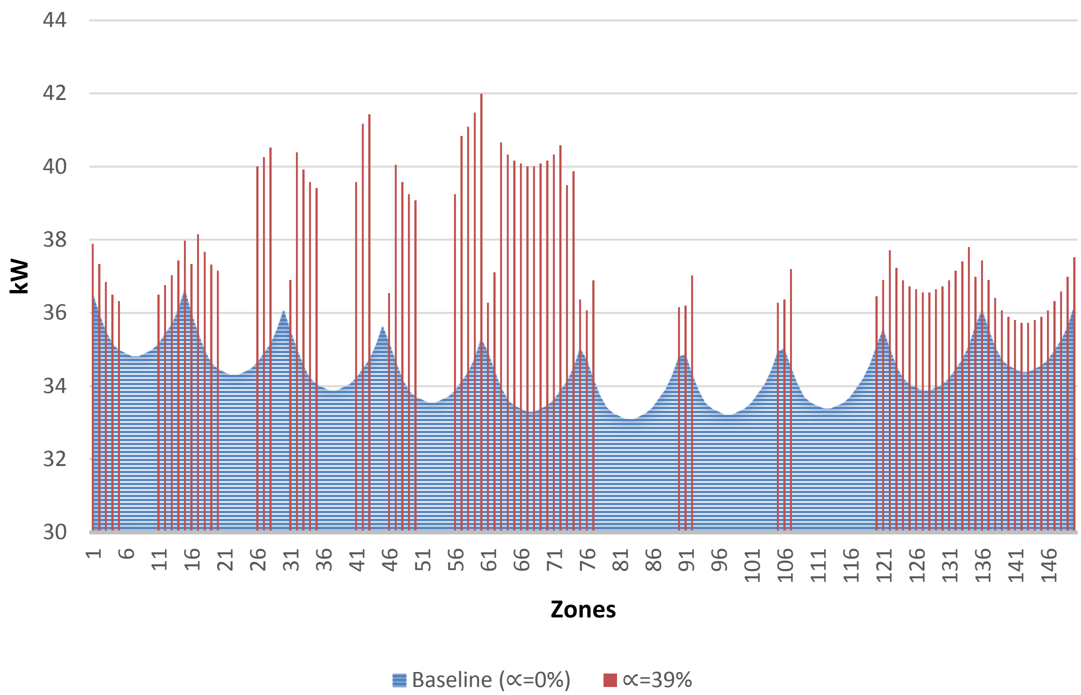


Fig. 39. Baseline Energy vs. Energy with User Preference α = 0.39 (Model #2)

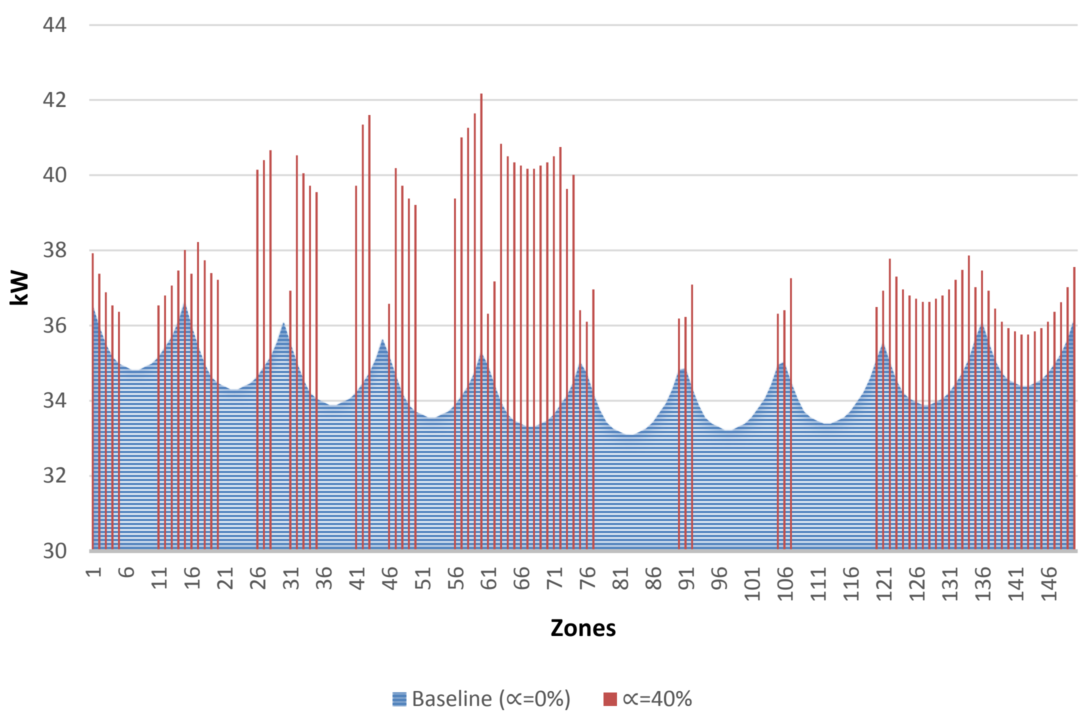


Fig. 40. Baseline Energy vs. Energy with User Preference α = 0.40 (Model #2)

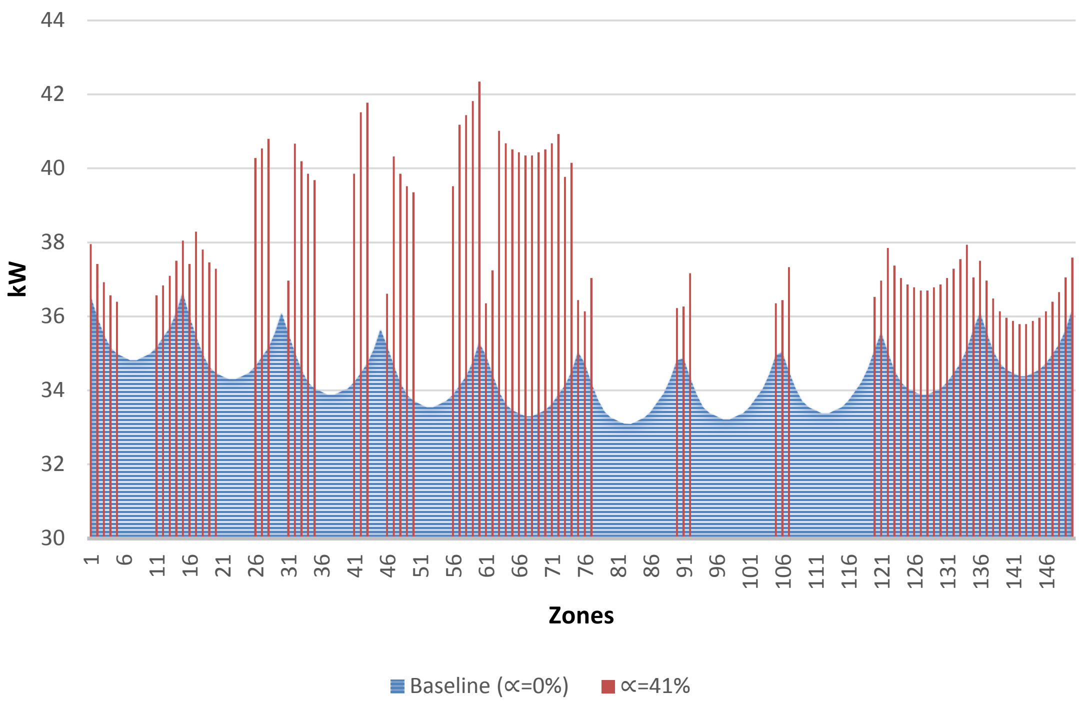


Fig. 41. Baseline Energy vs. Energy with User Preference α = 0.41 (Model #2)

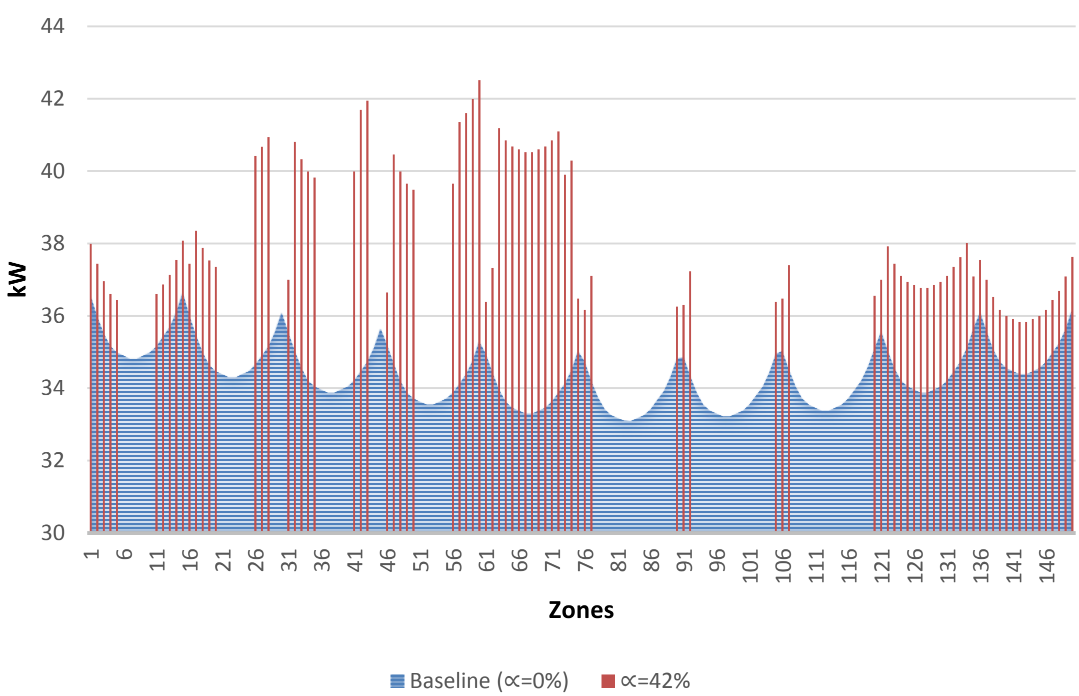


Fig. 42. Baseline Energy vs. Energy with User Preference α = 0.42 (Model #2)

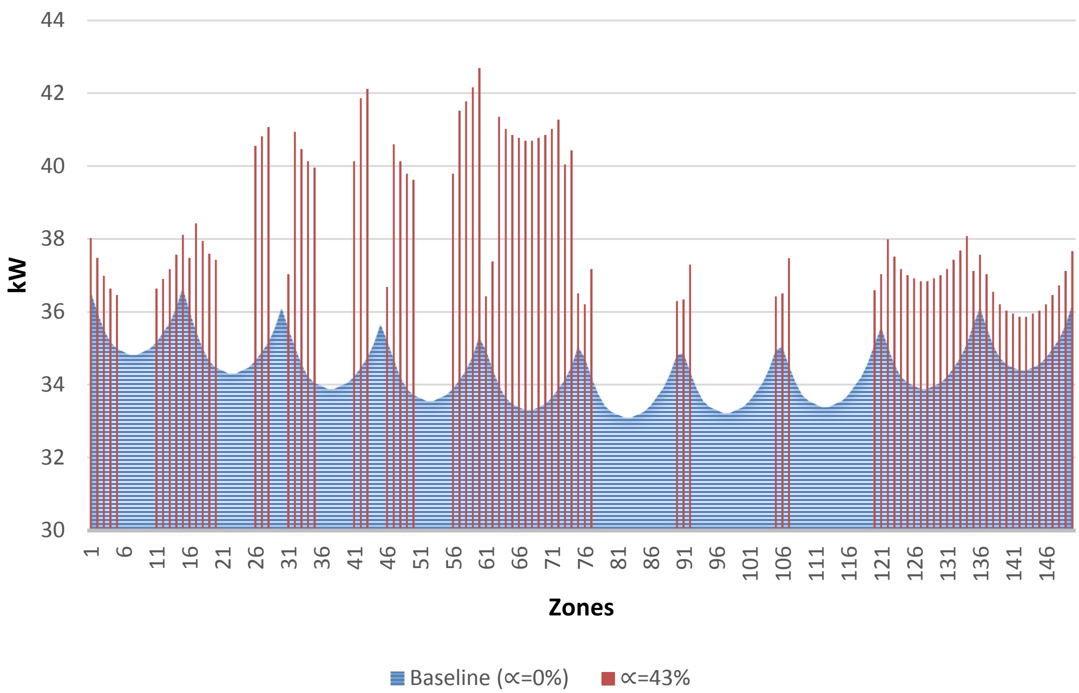


Fig. 43. Baseline Energy vs. Energy with User Preference α = 0.43 (Model #2)

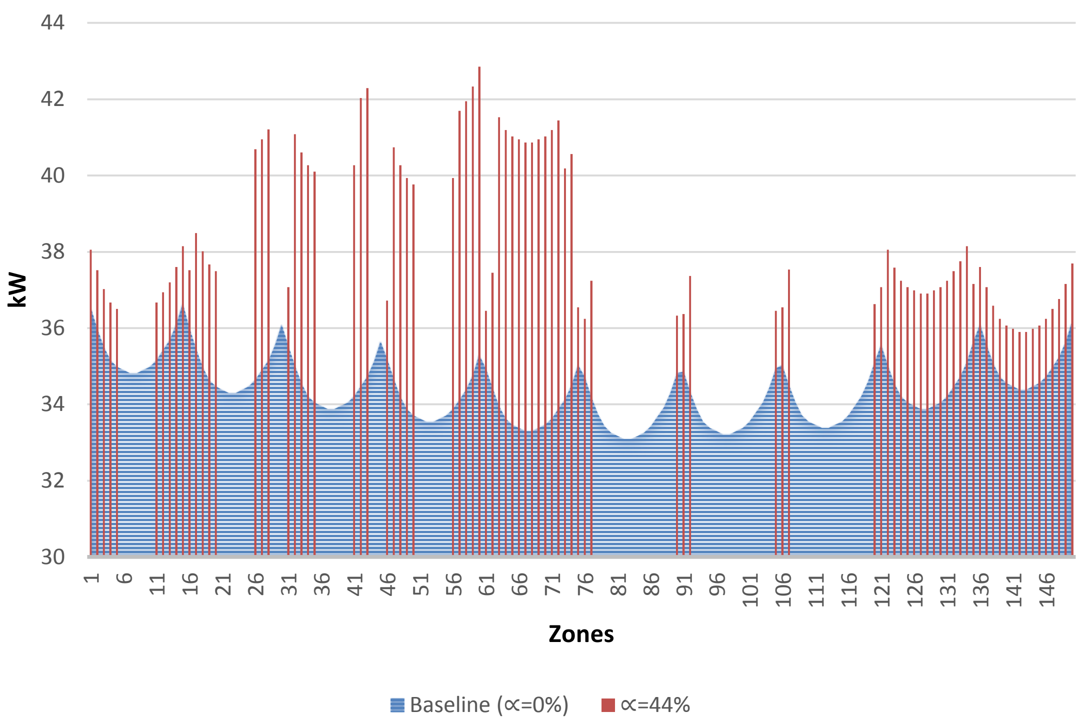


Fig. 44. Baseline Energy vs. Energy with User Preference α = 0.44 (Model #2)

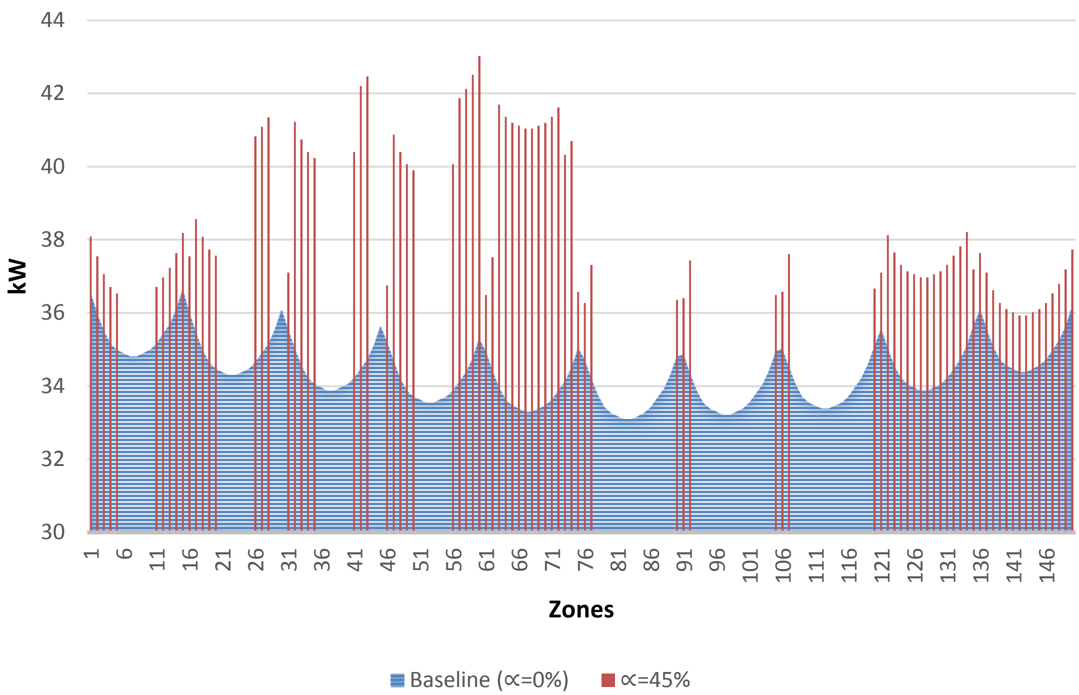


Fig. 45. Baseline Energy vs. Energy with User Preference α = 0.45 (Model #2)

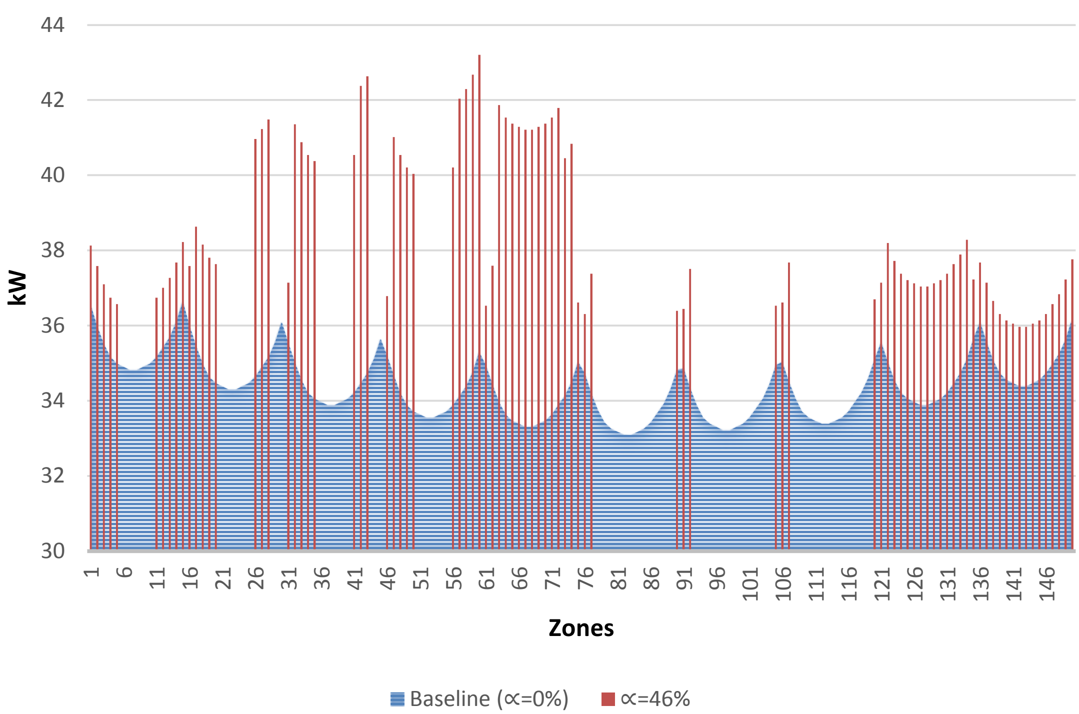


Fig. 46. Baseline Energy vs. Energy with User Preference α = 0.46 (Model #2)

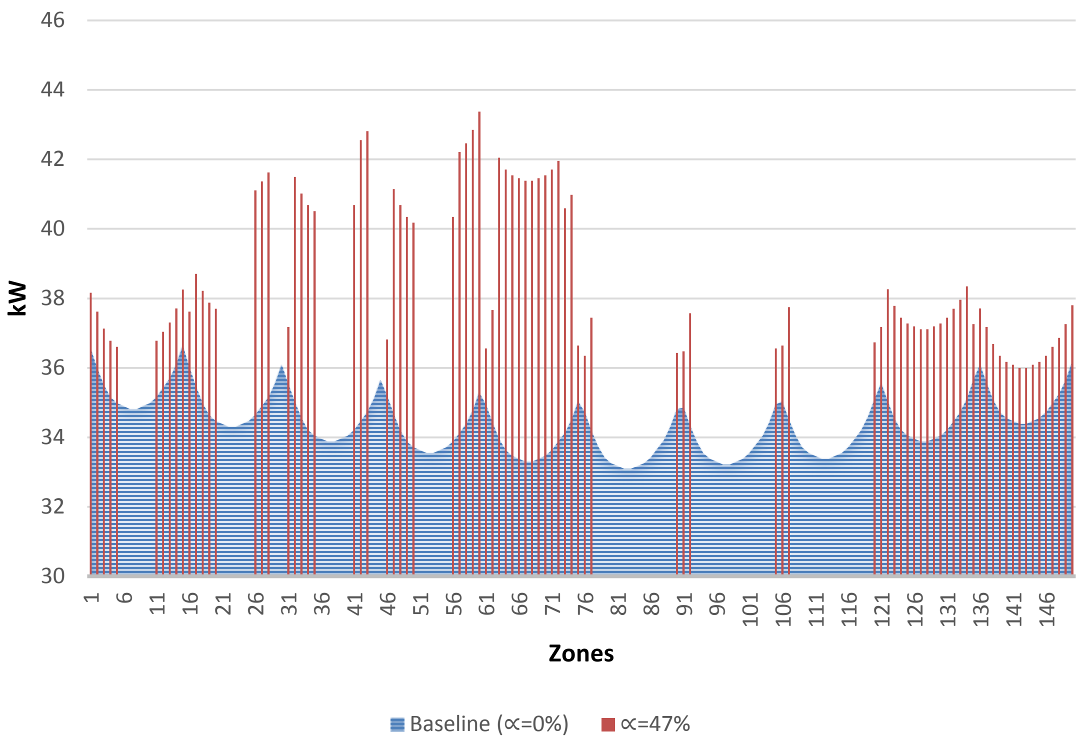


Fig. 47. Baseline Energy vs. Energy with User Preference α = 0.47 (Model #2)

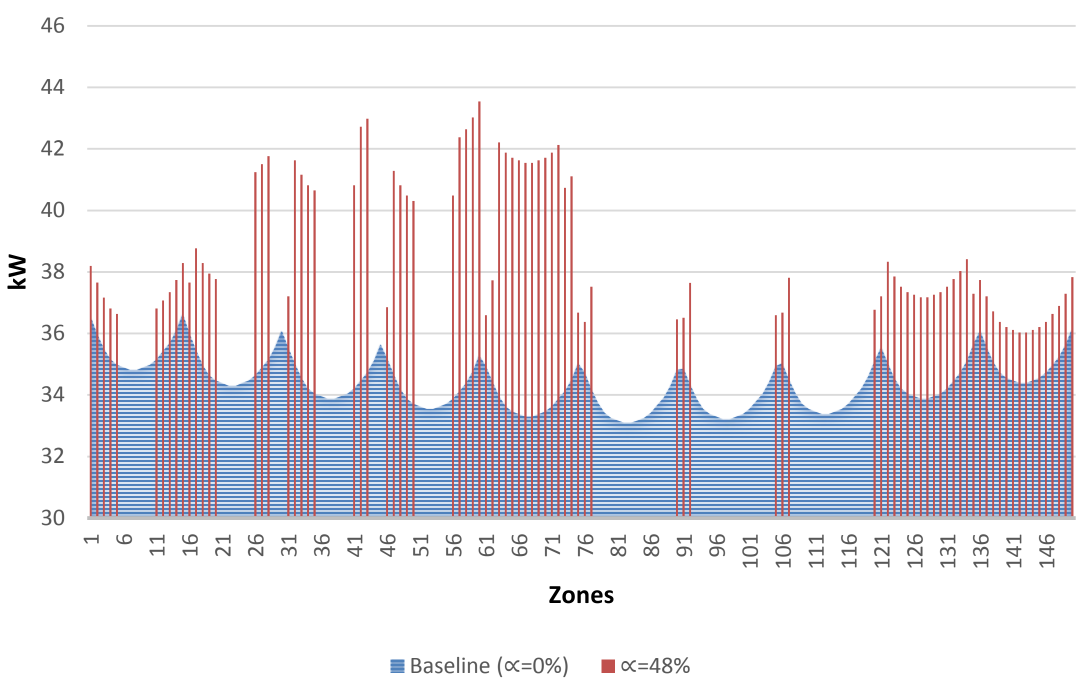


Fig. 48. Baseline Energy vs. Energy with User Preference α = 0.48 (Model #2)

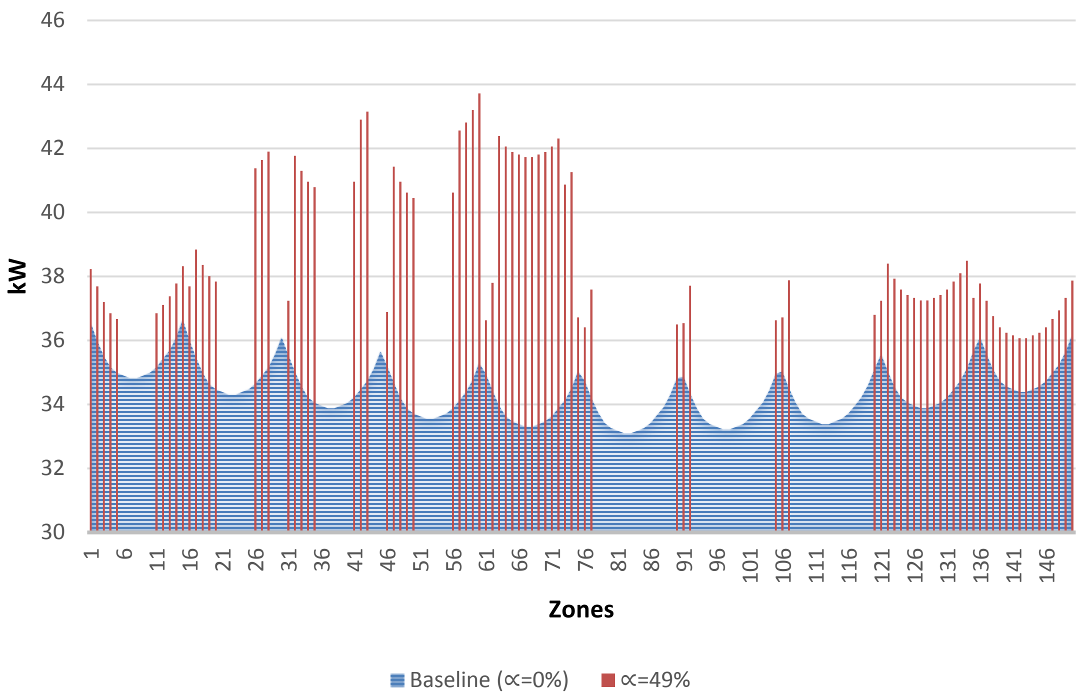


Fig. 49. Baseline Energy vs. Energy with User Preference α = 0.49 (Model #2)

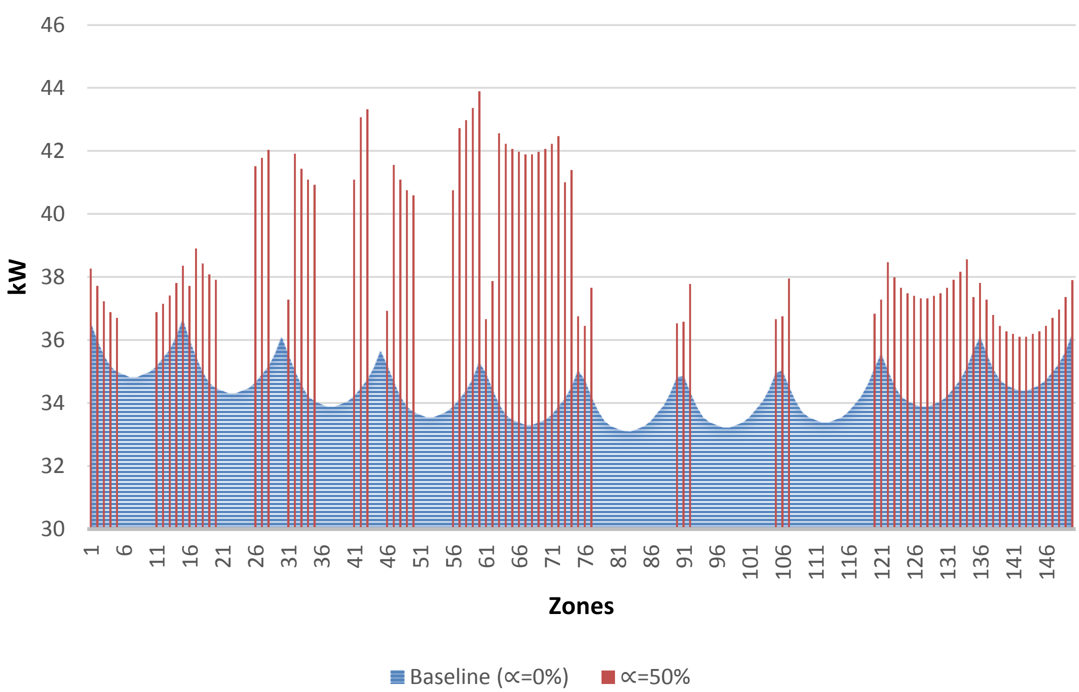


Fig. 50. Baseline Energy vs. Energy with User Preference α = 0.50 (Model #2)

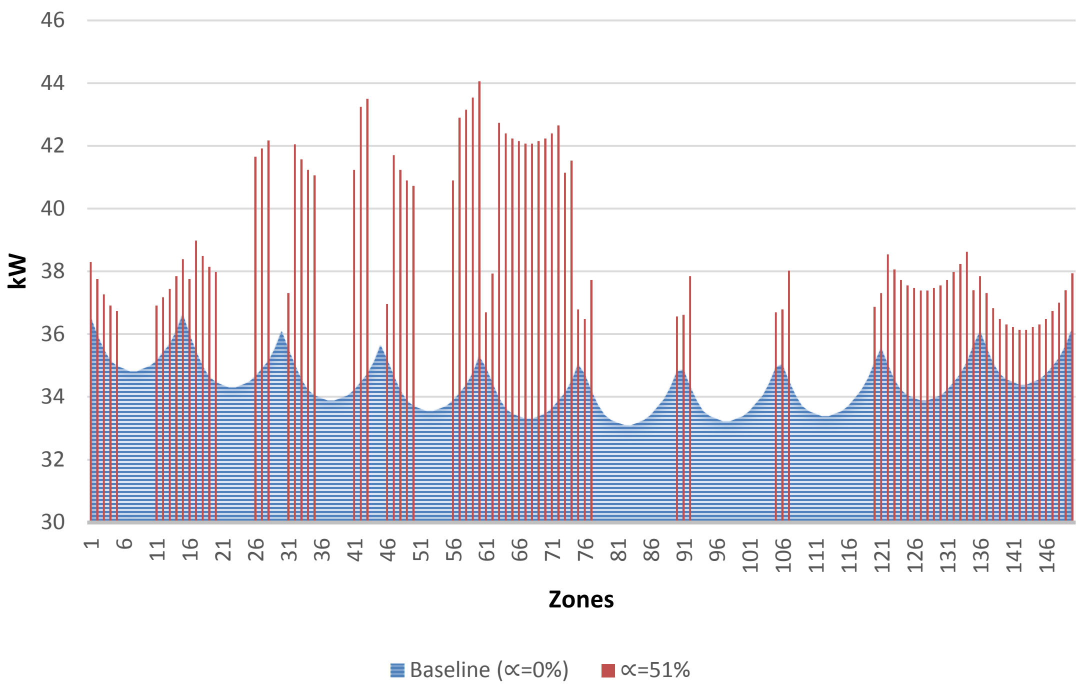


Fig. 51. Baseline Energy vs. Energy with User Preference α = 0.51 (Model #2)

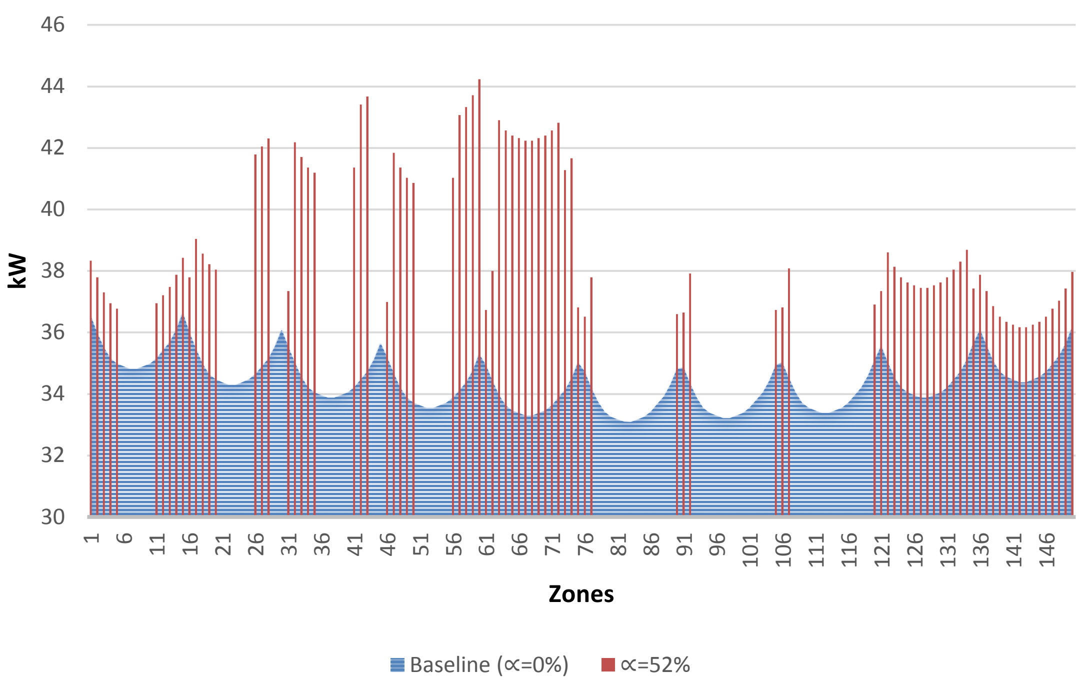


Fig. 52. Baseline Energy vs. Energy with User Preference α = 0.52 (Model #2)

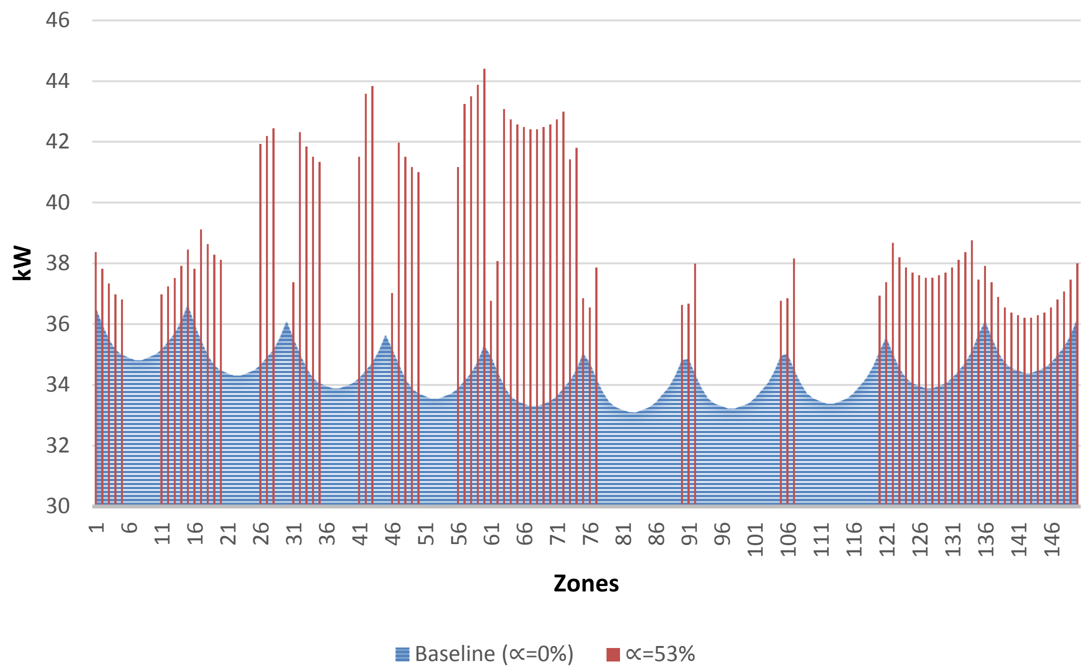


Fig. 53. Baseline Energy vs. Energy with User Preference α = 0.53 (Model #2)

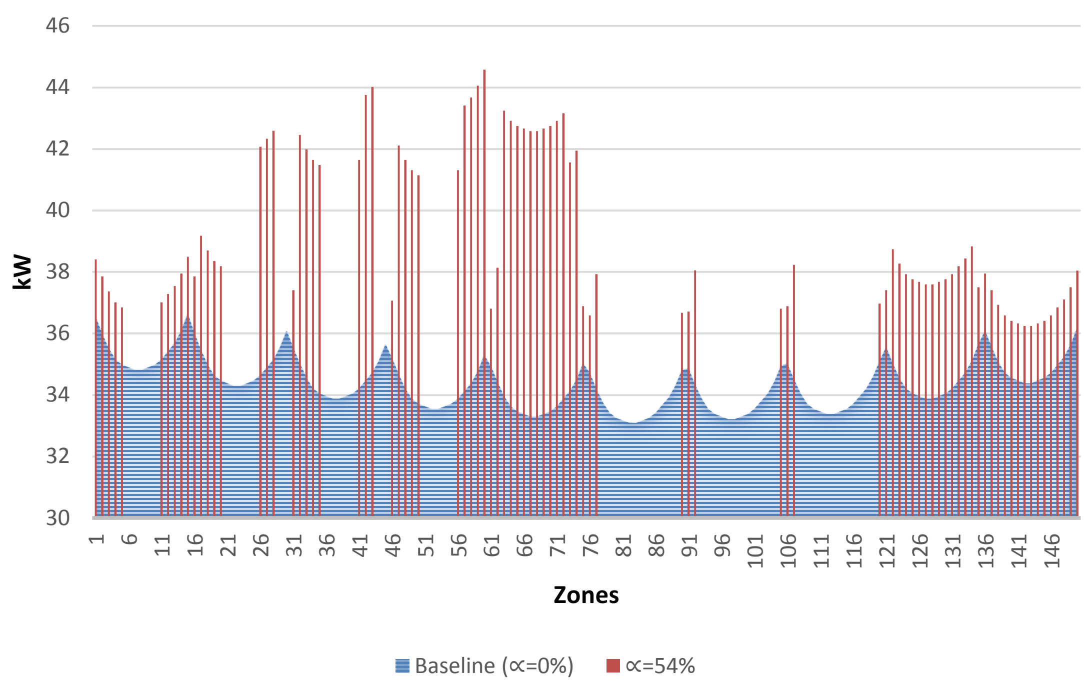


Fig. 54. Baseline Energy vs. Energy with User Preference α = 0.54 (Model #2)

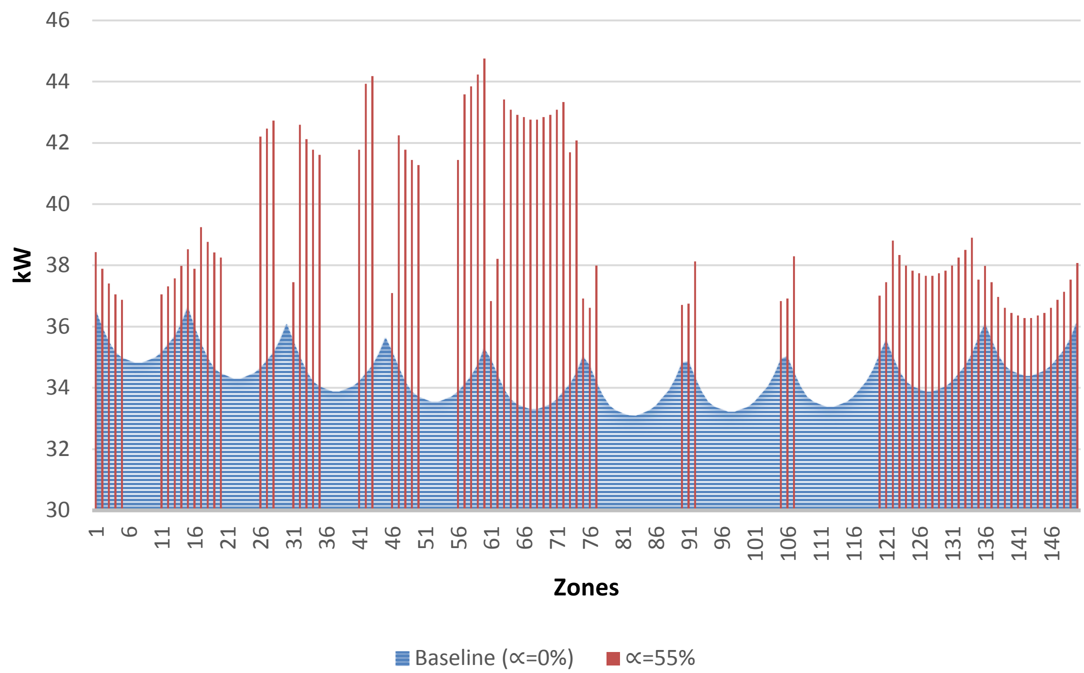


Fig. 55. Baseline Energy vs. Energy with User Preference α = 0.55 (Model #2)

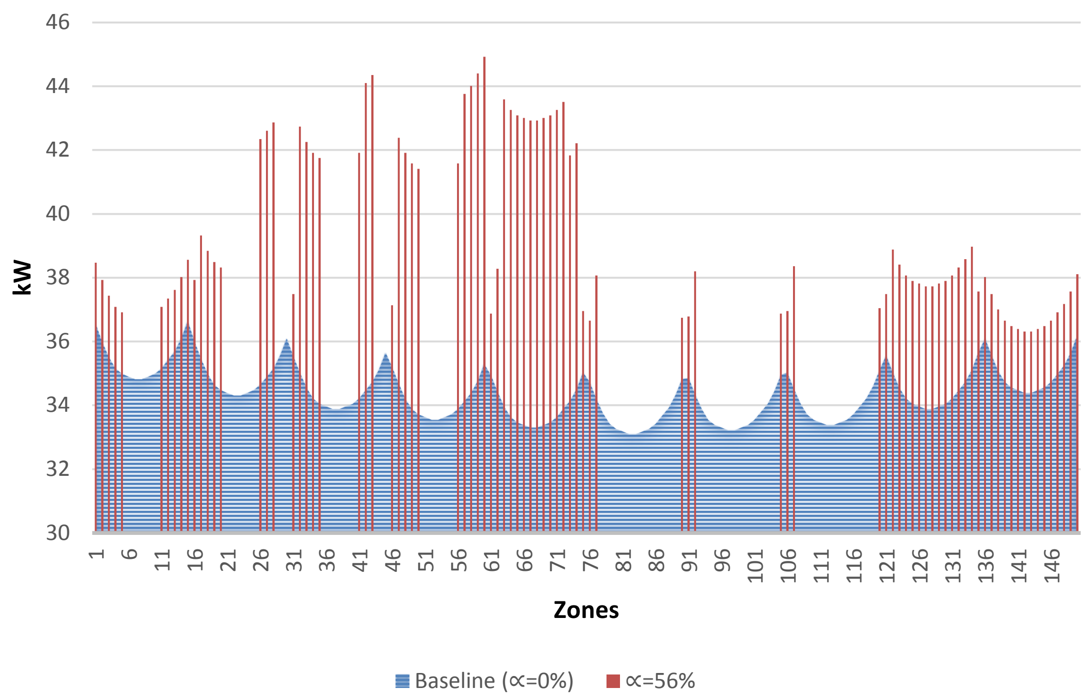


Fig. 56. Baseline Energy vs. Energy with User Preference α = 0.56 (Model #2)

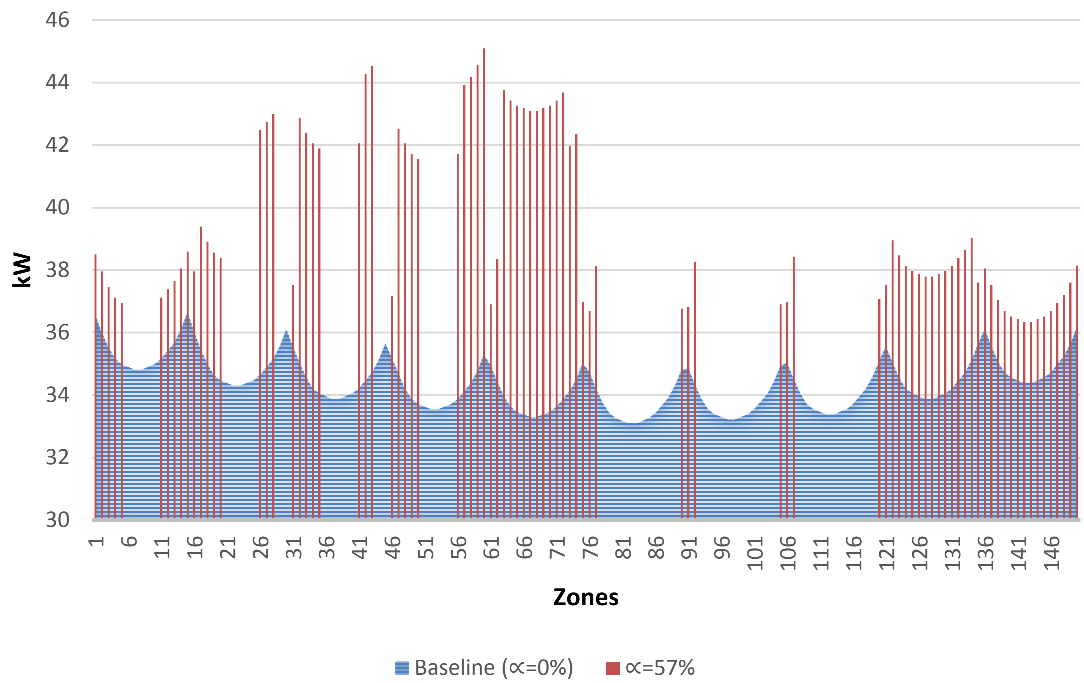


Fig. 57. Baseline Energy vs. Energy with User Preference α = 0.57 (Model #2)

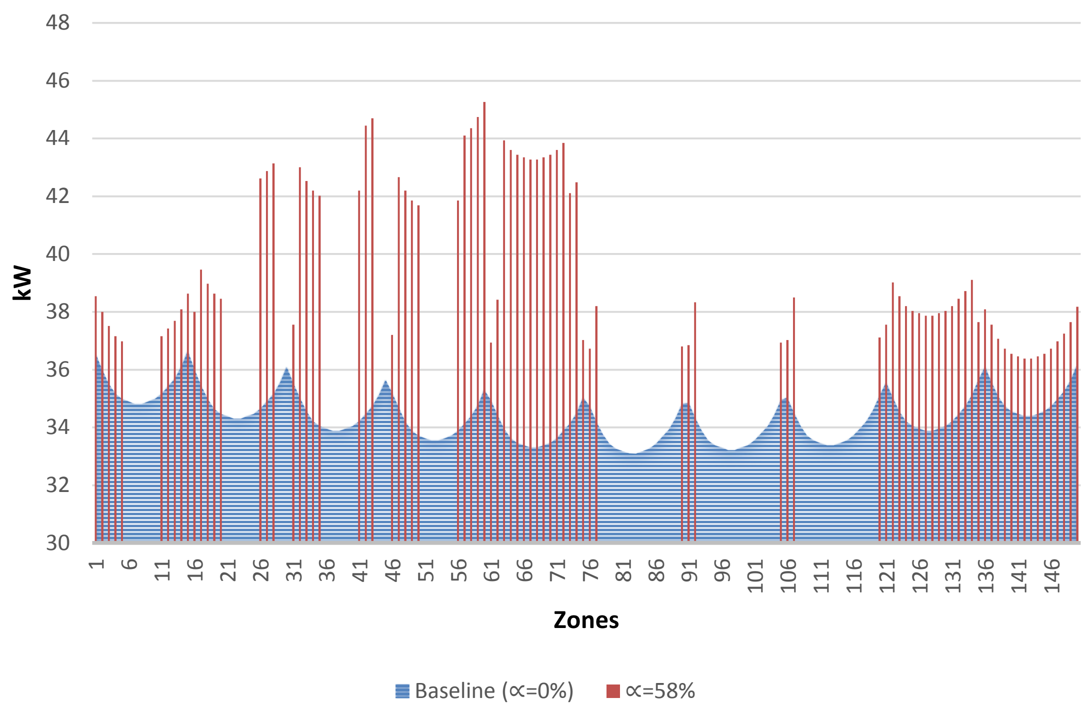


Fig. 58. Baseline Energy vs. Energy with User Preference α = 0.58 (Model #2)

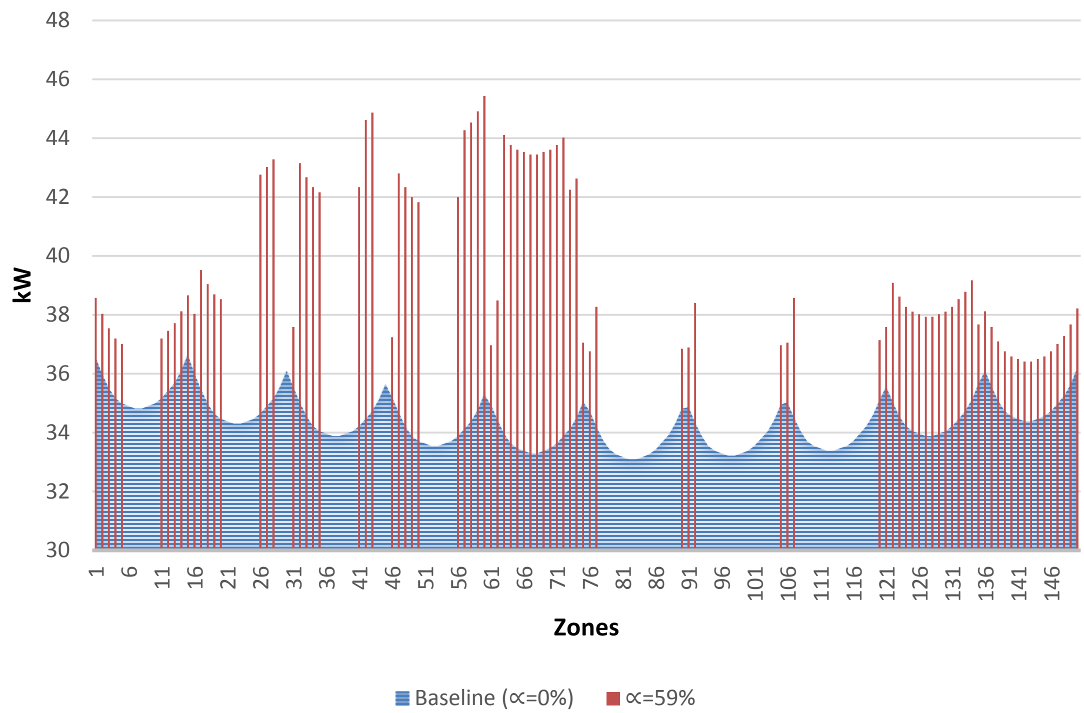


Fig. 59. Baseline Energy vs. Energy with User Preference α = 0.59 (Model #2)

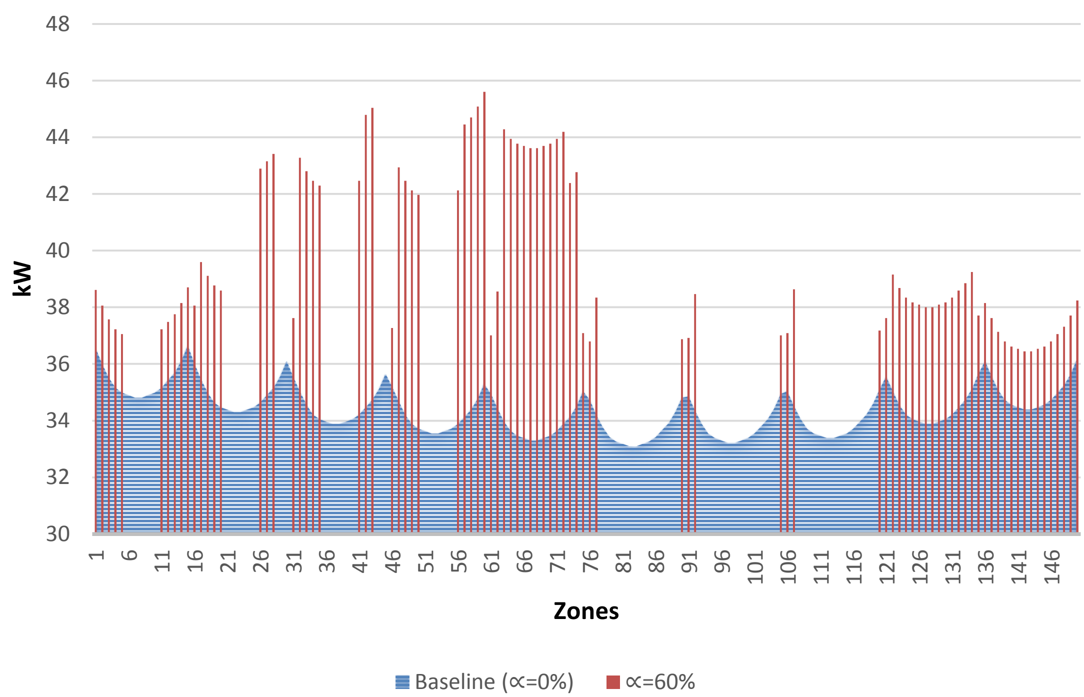


Fig. 60. Baseline Energy vs. Energy with User Preference α = 0.60 (Model #2)

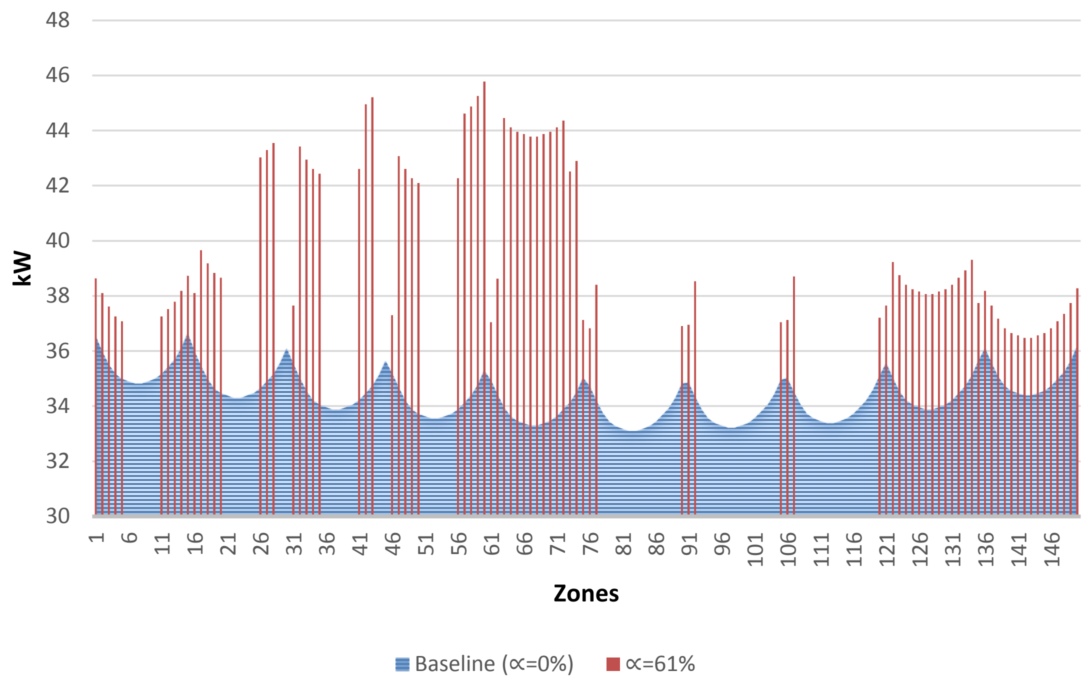


Fig. 61. Baseline Energy vs. Energy with User Preference α = 0.61 (Model #2)

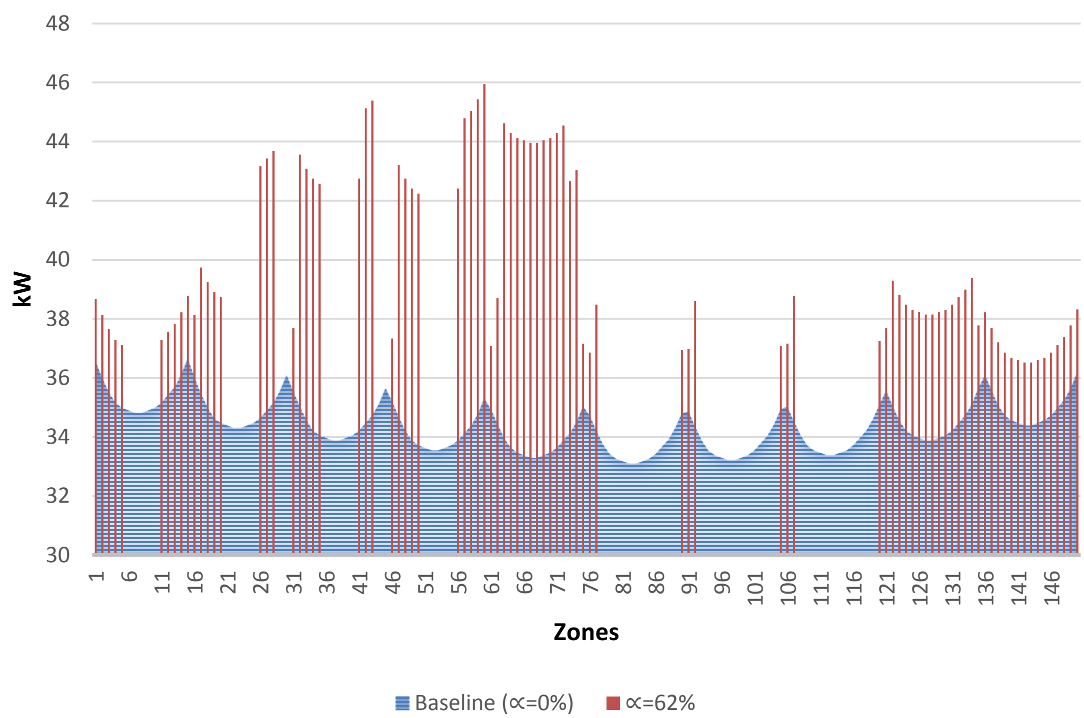


Fig. 62. Baseline Energy vs. Energy with User Preference α = 0.62 (Model #2)

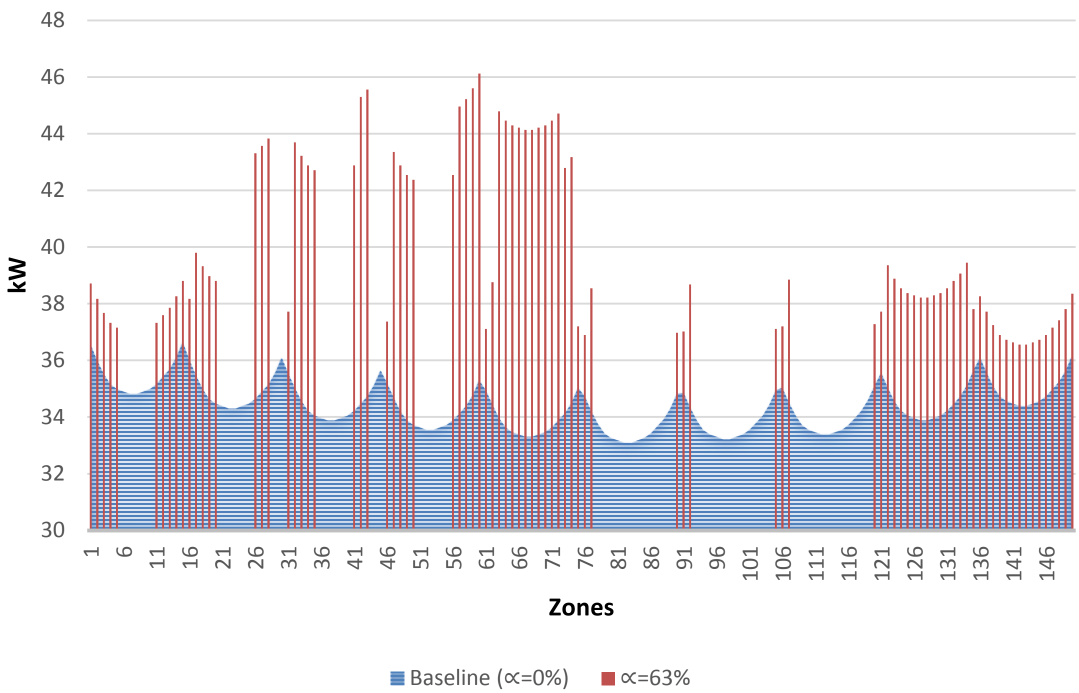


Fig. 63. Baseline Energy vs. Energy with User Preference α = 0.63 (Model #2)

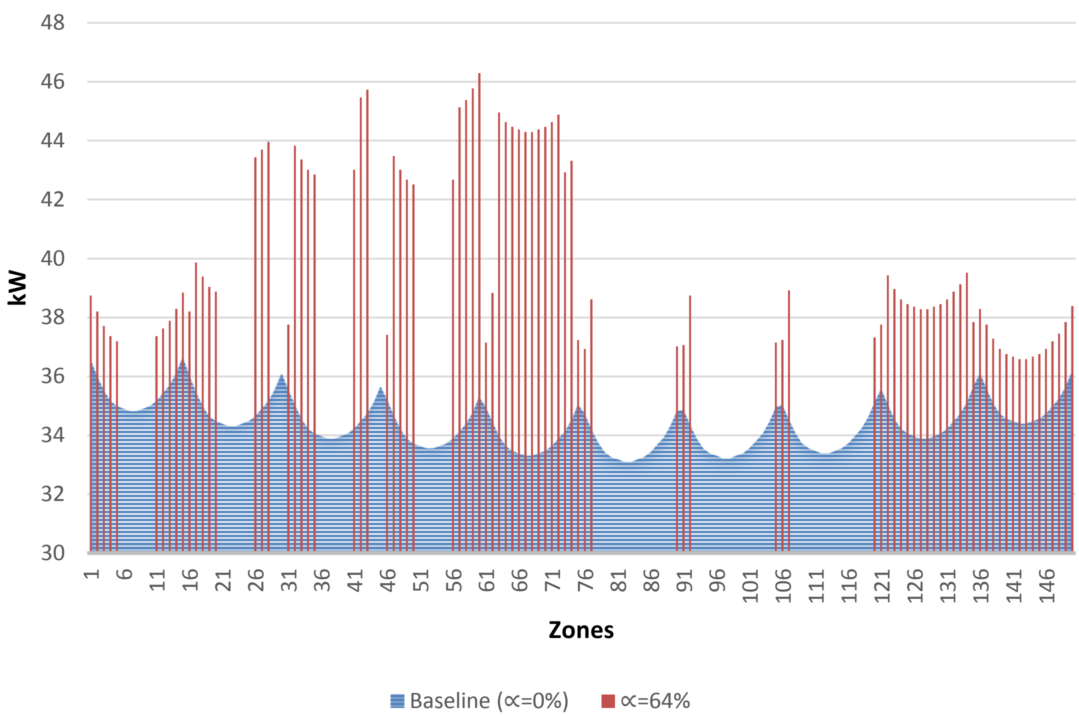


Fig. 64. Baseline Energy vs. Energy with User Preference α = 0.64 (Model #2)

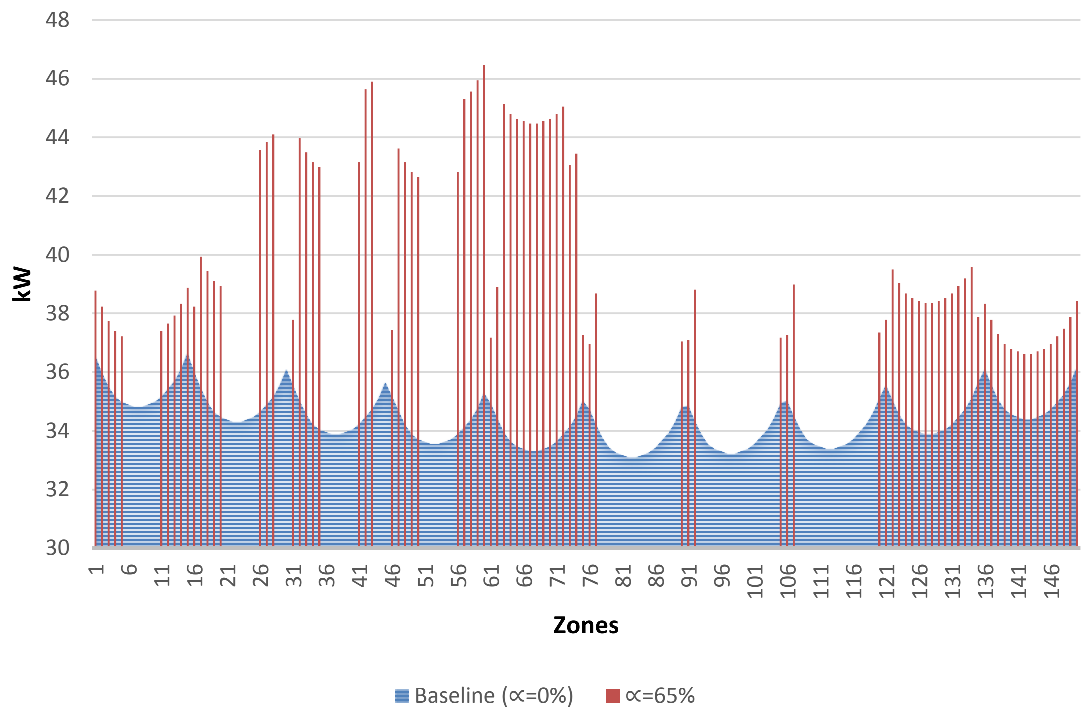


Fig. 65. Baseline Energy vs. Energy with User Preference α = 0.65 (Model #2)

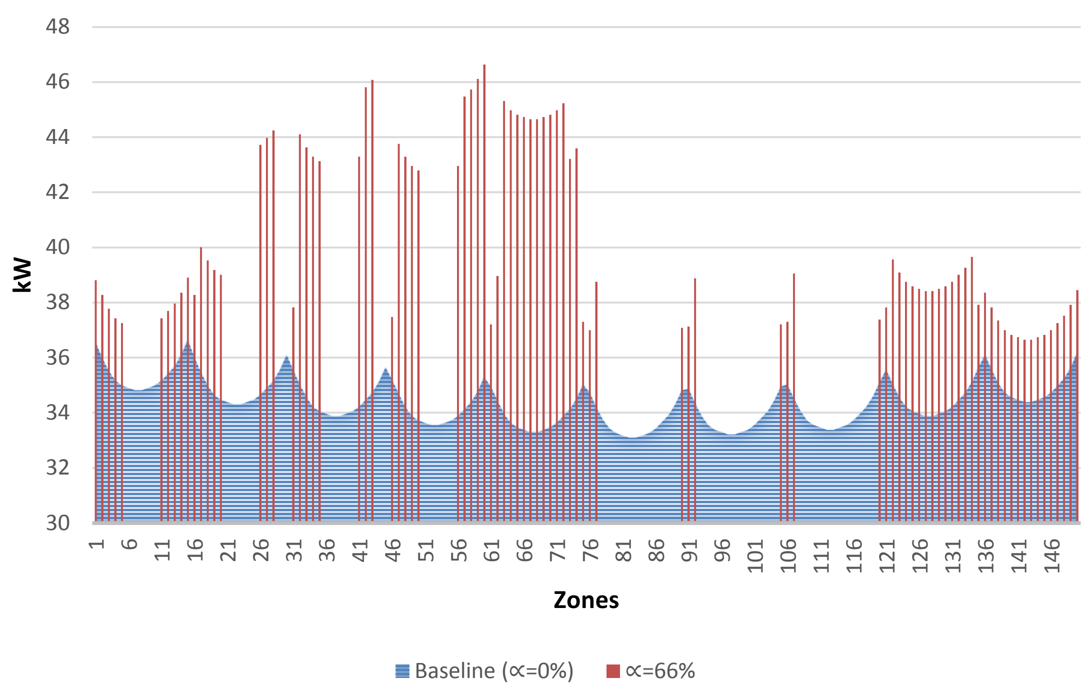


Fig. 66. Baseline Energy vs. Energy with User Preference α = 0.66 (Model #2)

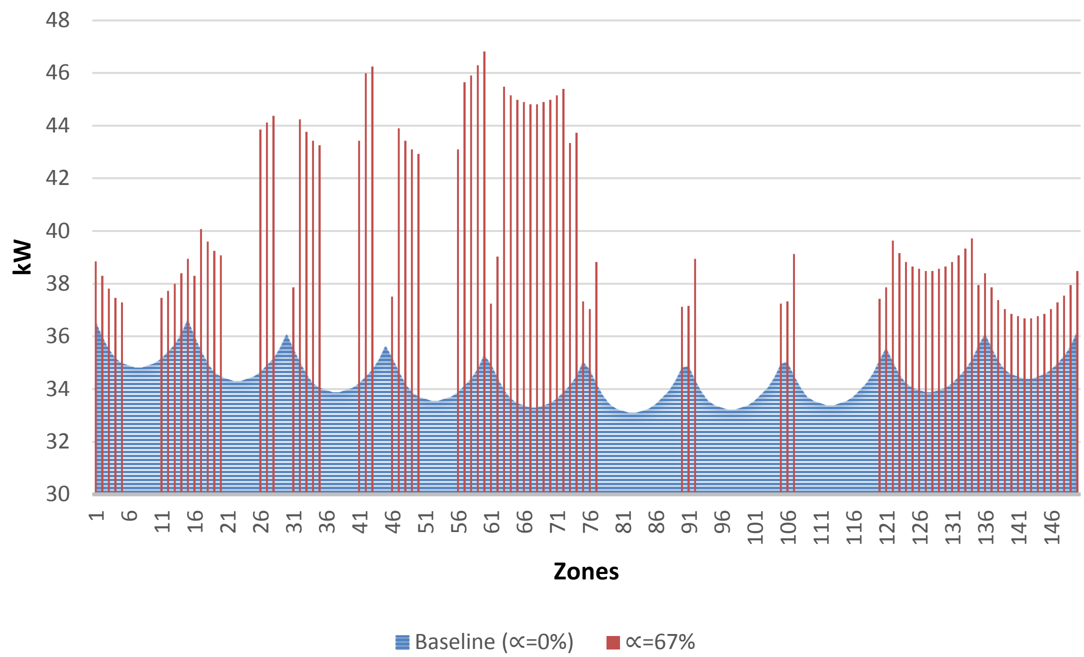


Fig. 67. Baseline Energy vs. Energy with User Preference α = 0.67 (Model #2)

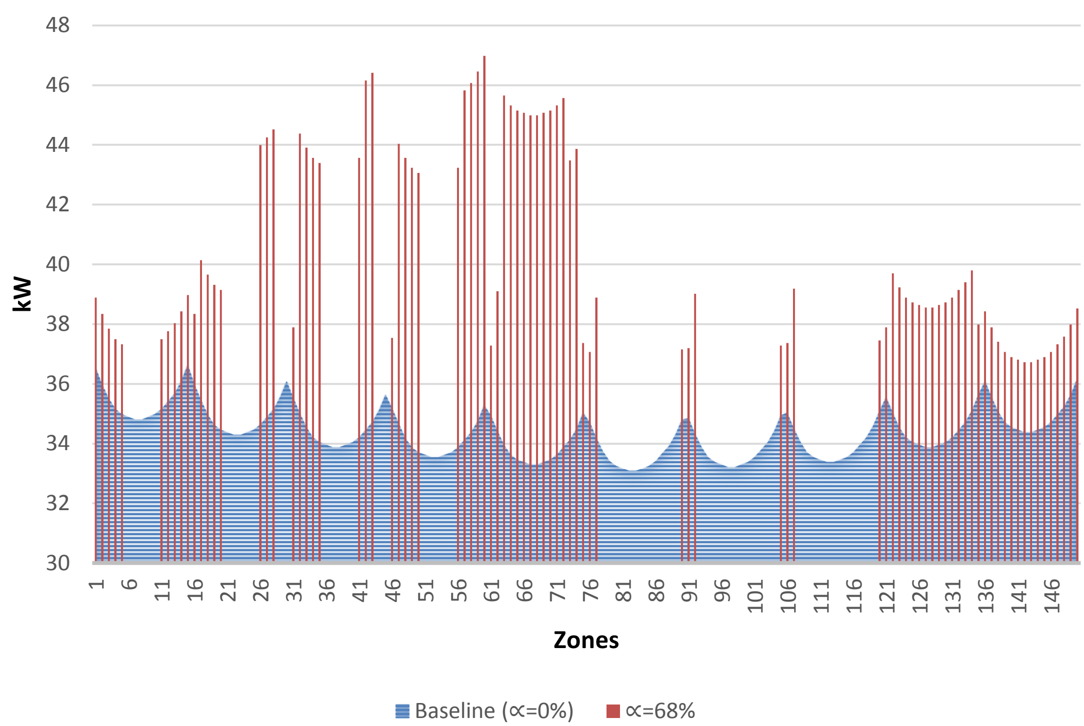


Fig. 68. Baseline Energy vs. Energy with User Preference α = 0.68 (Model #2)

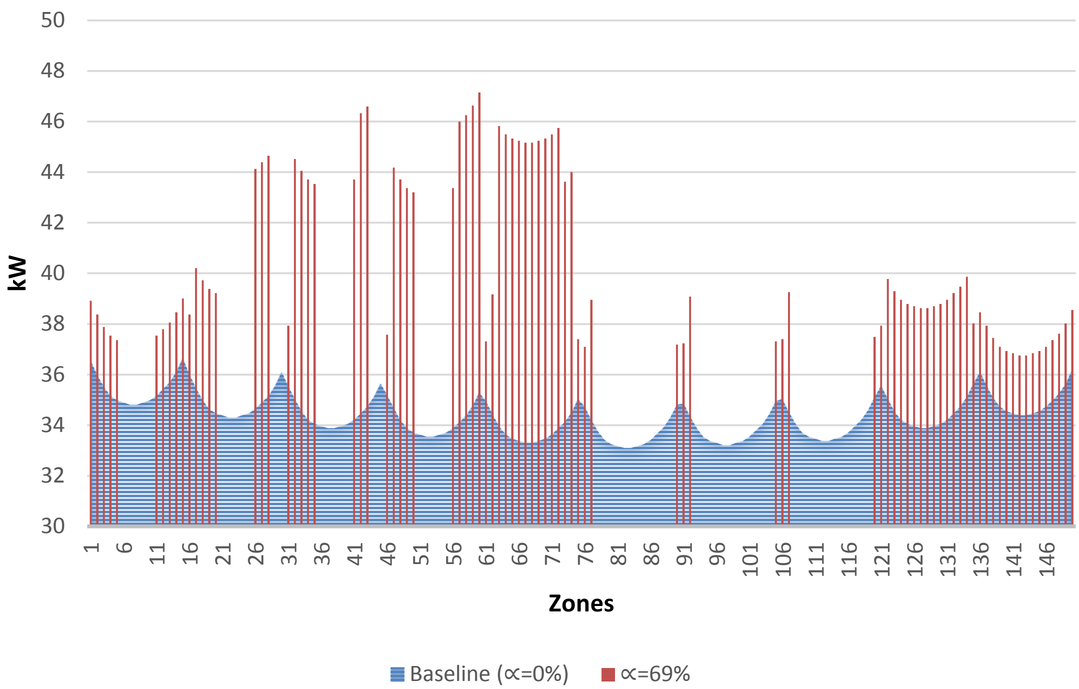


Fig. 69. Baseline Energy vs. Energy with User Preference α = 0.69 (Model #2)

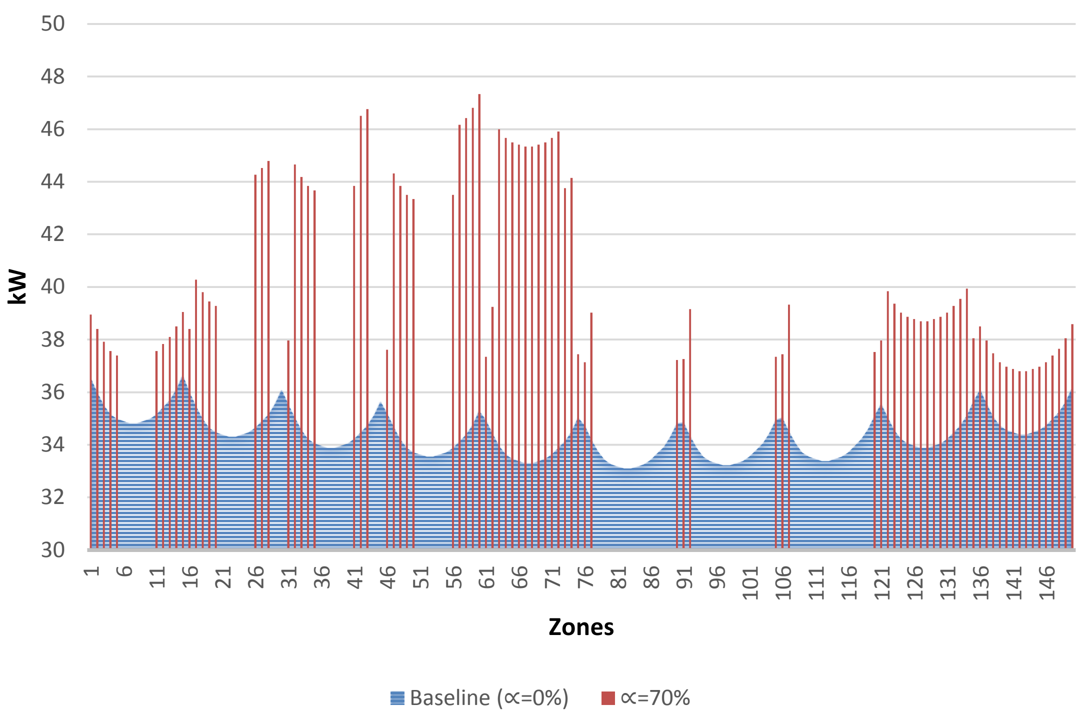


Fig. 70. Baseline Energy vs. Energy with User Preference α = 0.70 (Model #2)

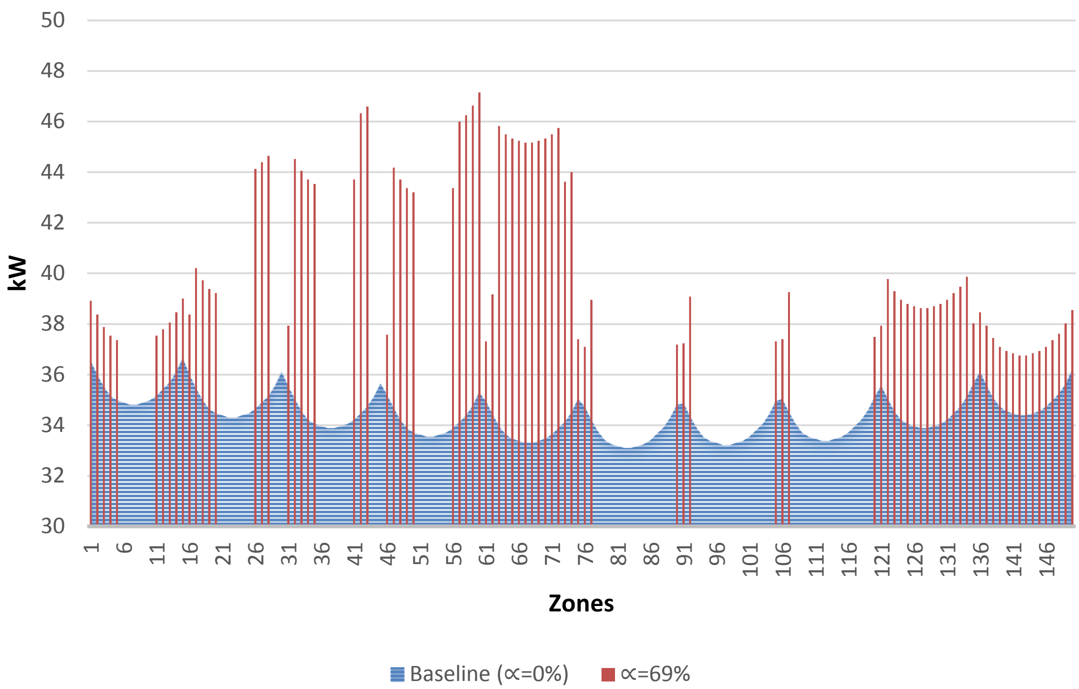


Fig. 69. Baseline Energy vs. Energy with User Preference α = 0.69 (Model #2)

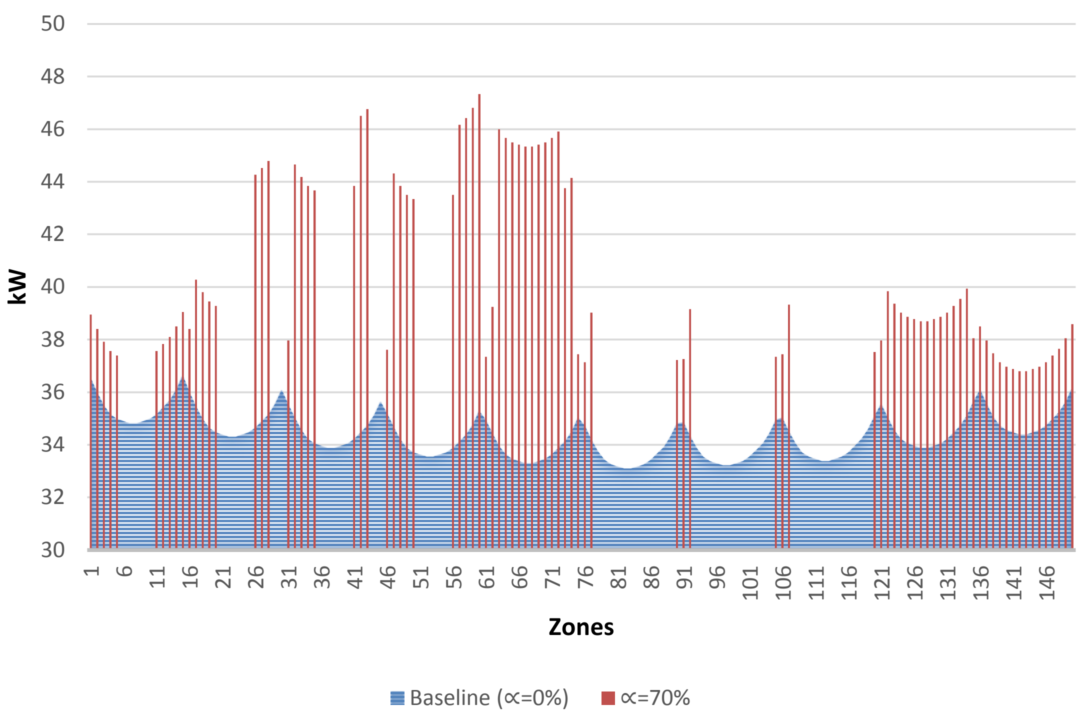


Fig. 70. Baseline Energy vs. Energy with User Preference α = 0.70 (Model #2)

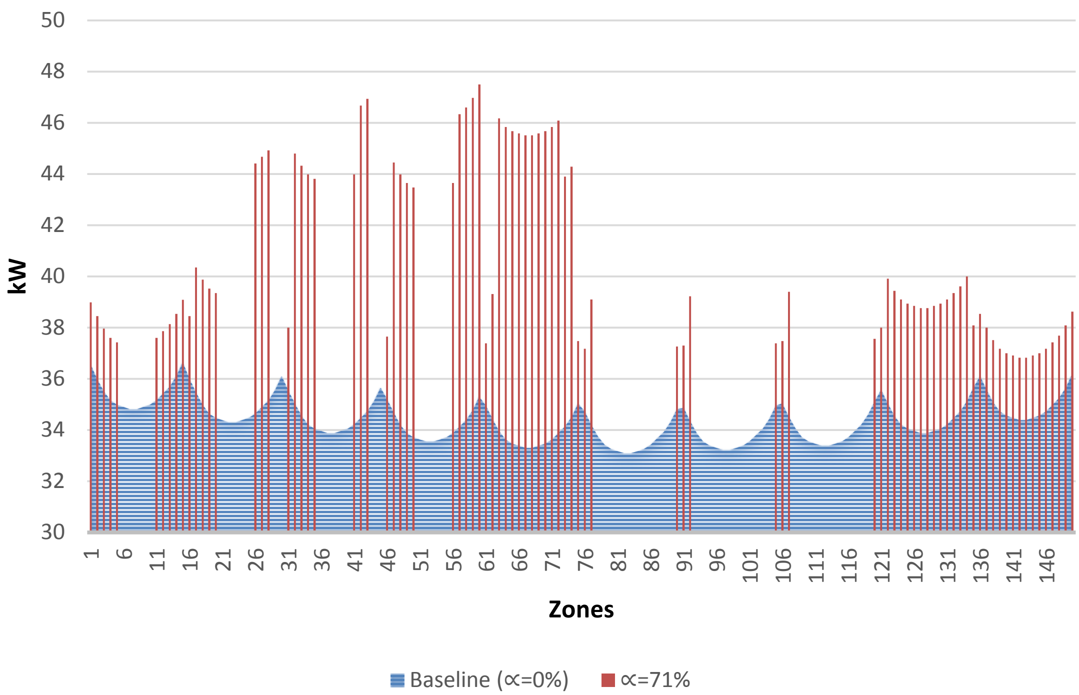


Fig. 71. Baseline Energy vs. Energy with User Preference α = 0.71 (Model #2)

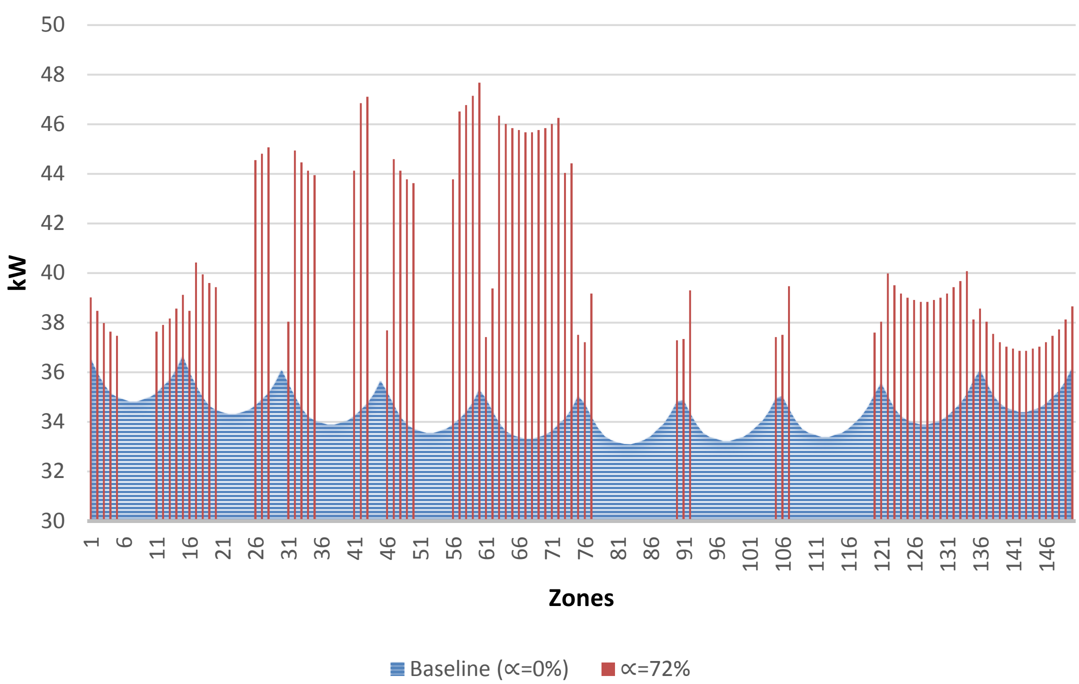


Fig. 72. Baseline Energy vs. Energy with User Preference α = 0.72 (Model #2)

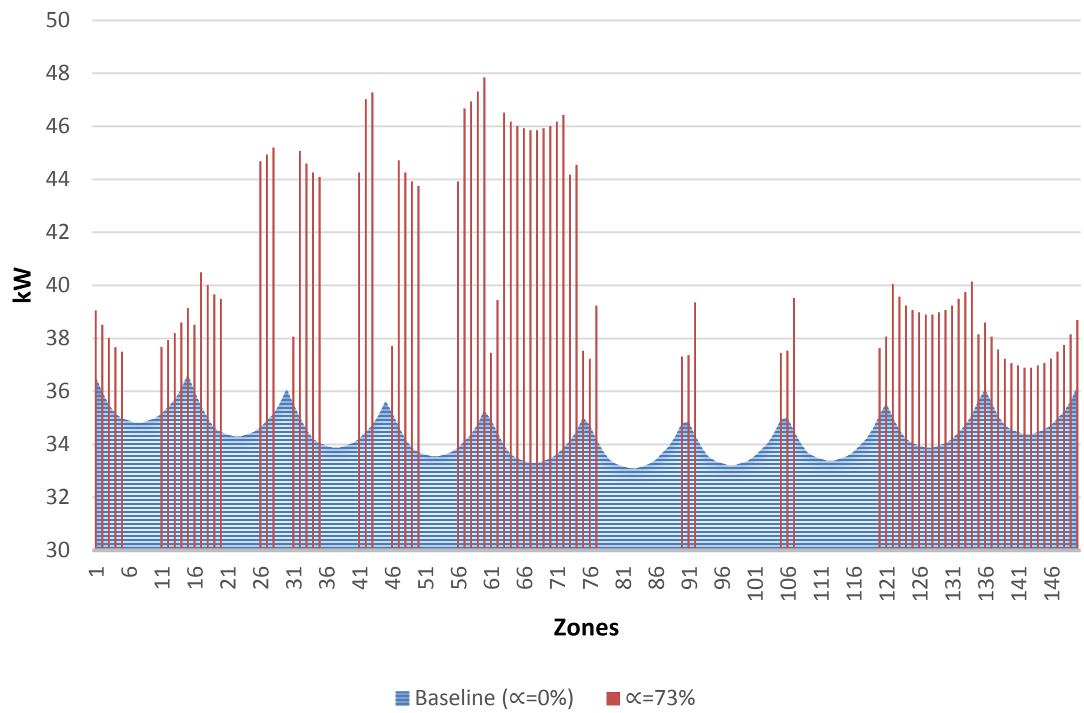


Fig. 73. Baseline Energy vs. Energy with User Preference α = 0.73 (Model #2)

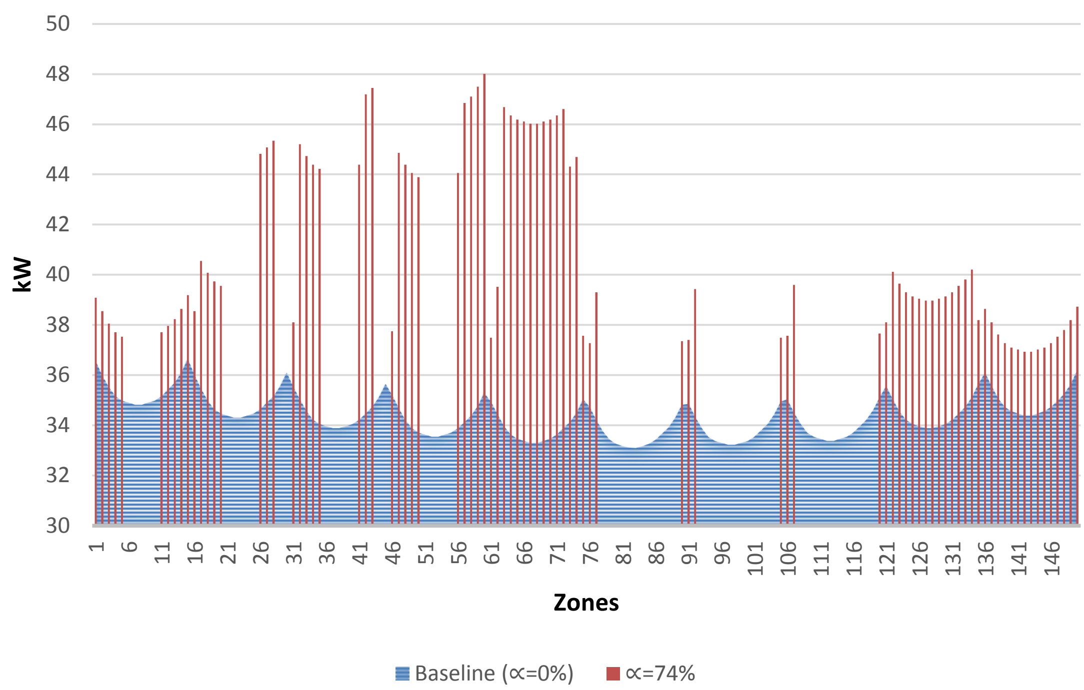


Fig. 74. Baseline Energy vs. Energy with User Preference α = 0.74 (Model #2)

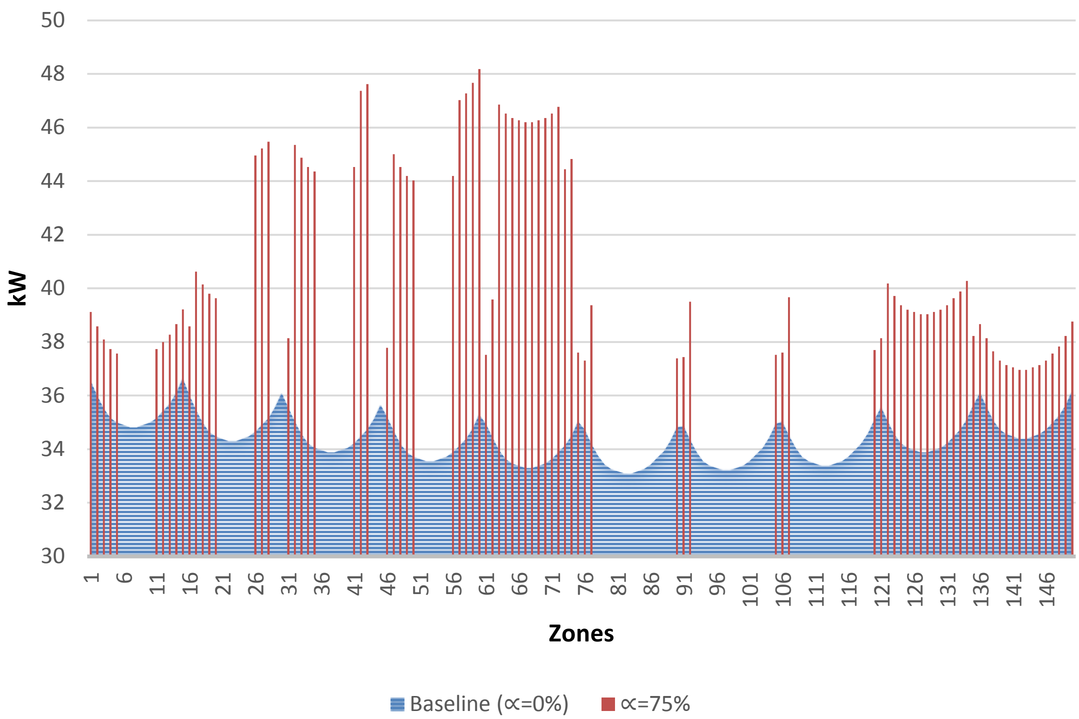


Fig. 75. Baseline Energy vs. Energy with User Preference α = 0.75 (Model #2)

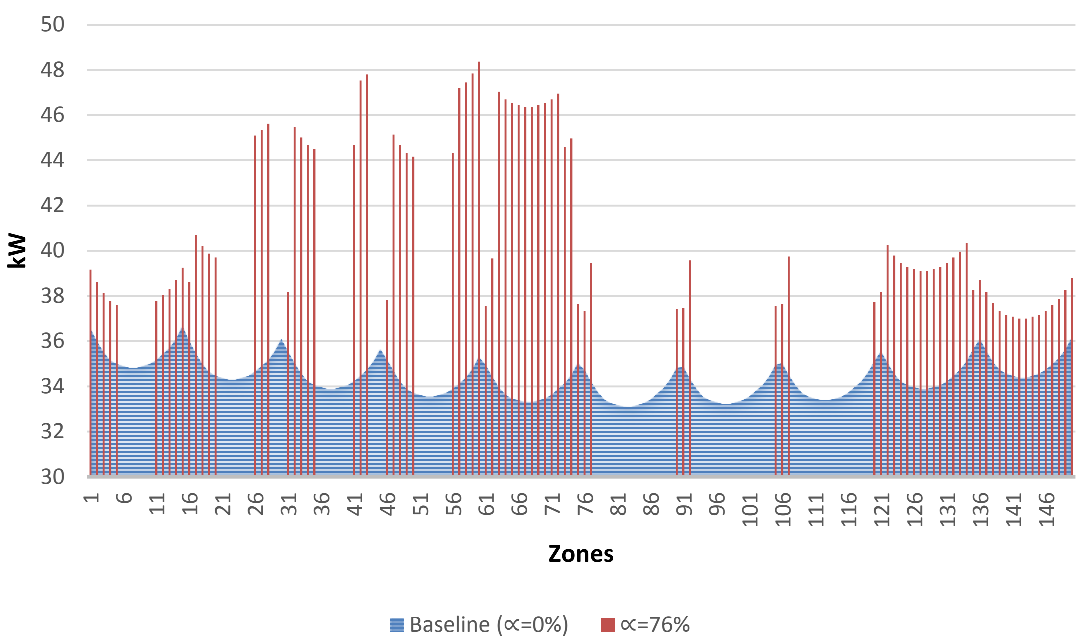


Fig. 76. Baseline Energy vs. Energy with User Preference α = 0.76 (Model #2)

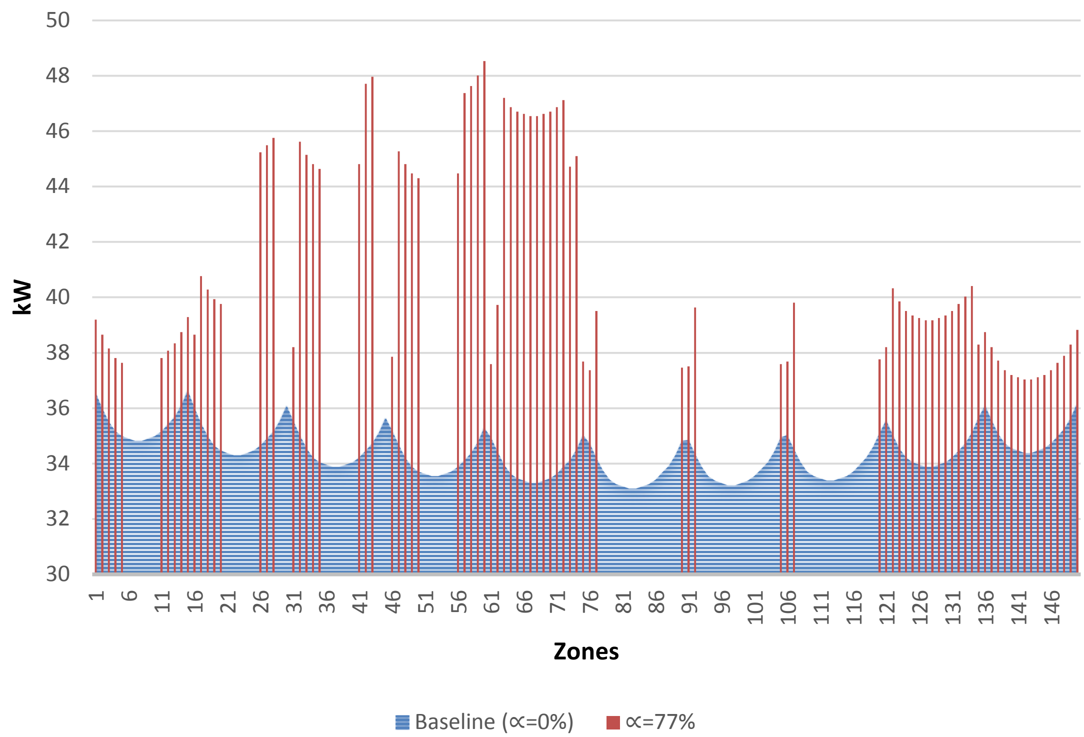


Fig. 77. Baseline Energy vs. Energy with User Preference α = 0.77 (Model #2)

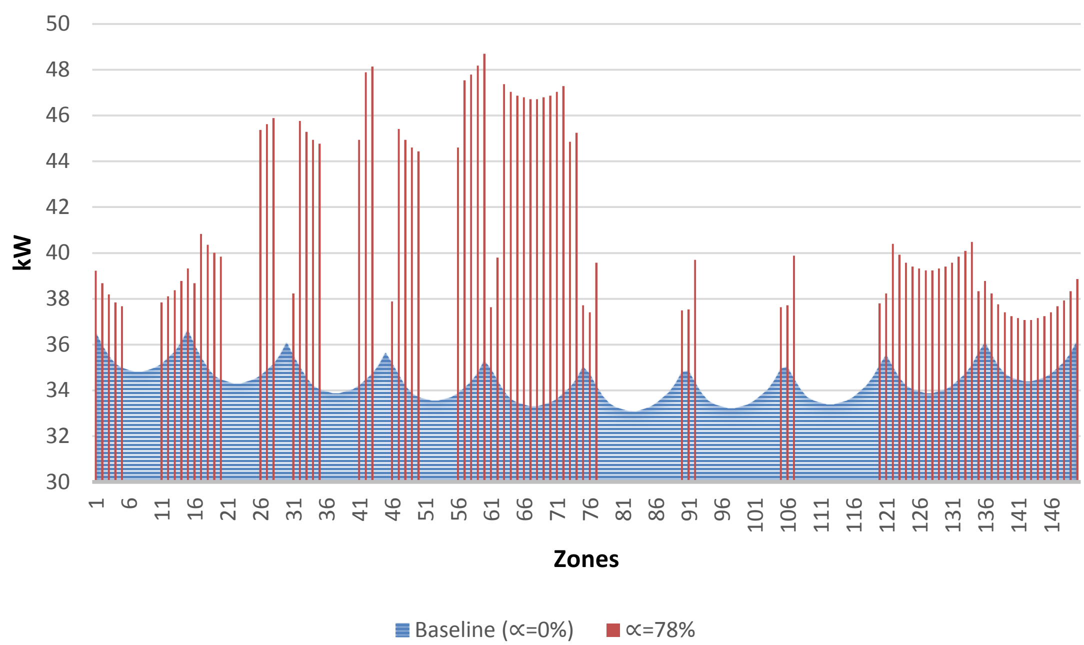


Fig. 78. Baseline Energy vs. Energy with User Preference α = 0.78 (Model #2)

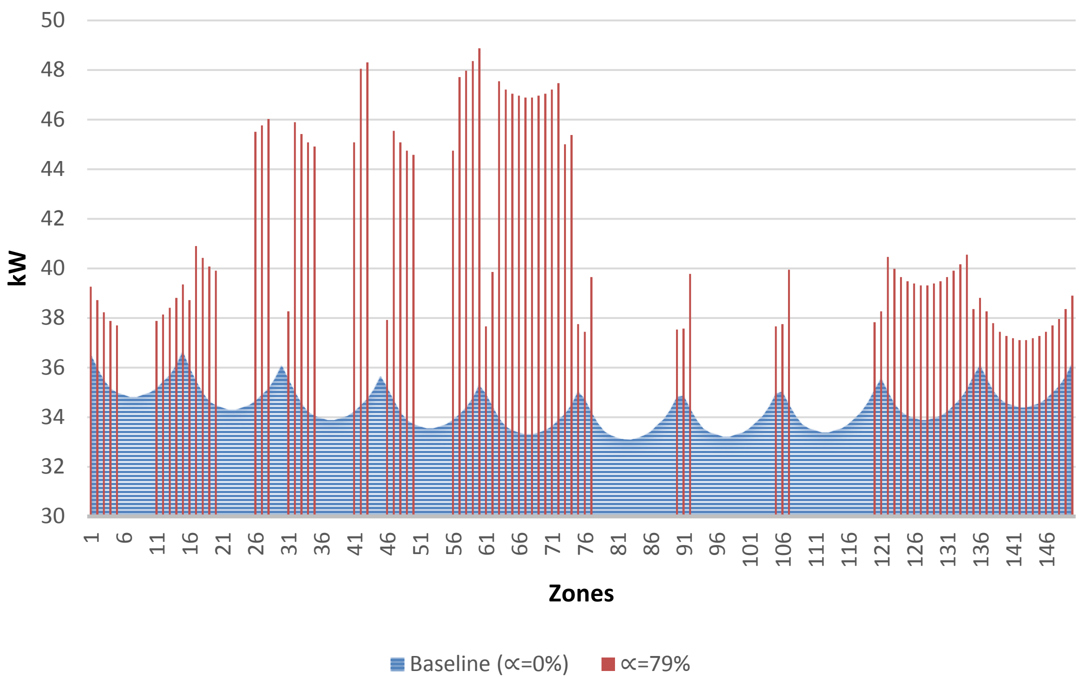


Fig. 79. Baseline Energy vs. Energy with User Preference α = 0.79 (Model #2)

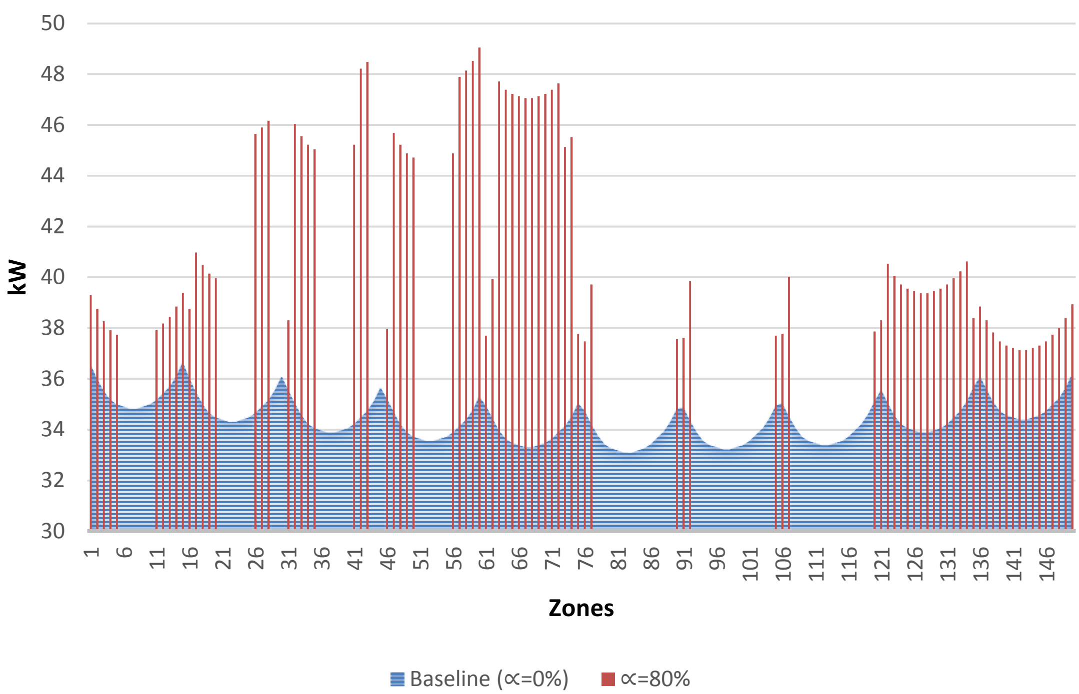


Fig. 80. Baseline Energy vs. Energy with User Preference α = 0.80 (Model #2)

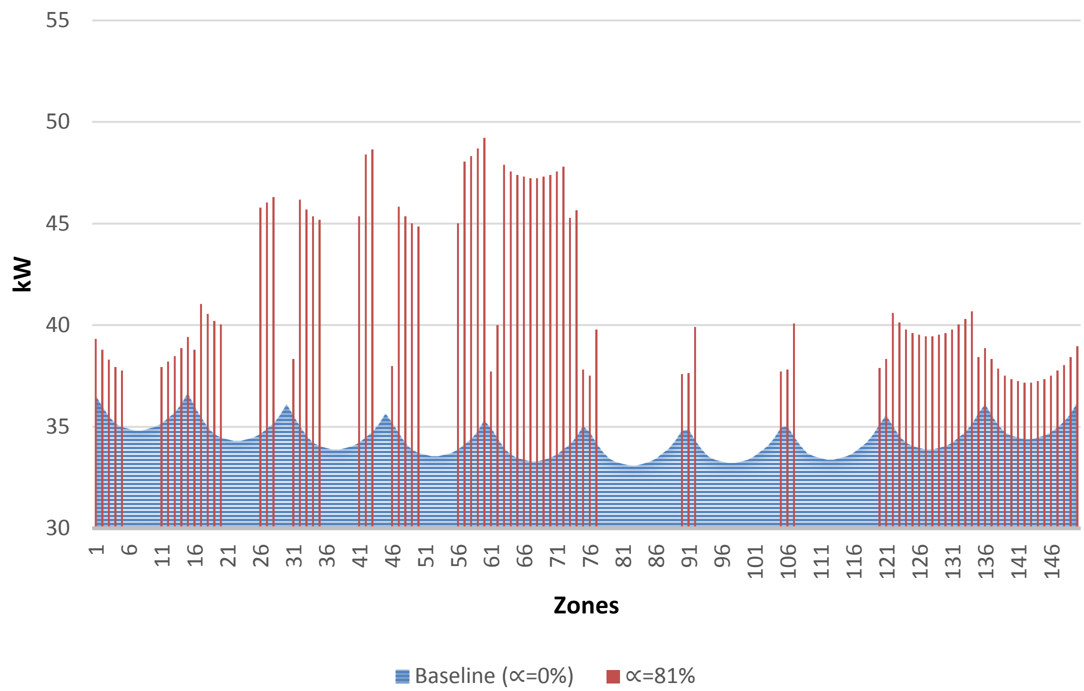


Fig. 81. Baseline Energy vs. Energy with User Preference α = 0.81 (Model #2)

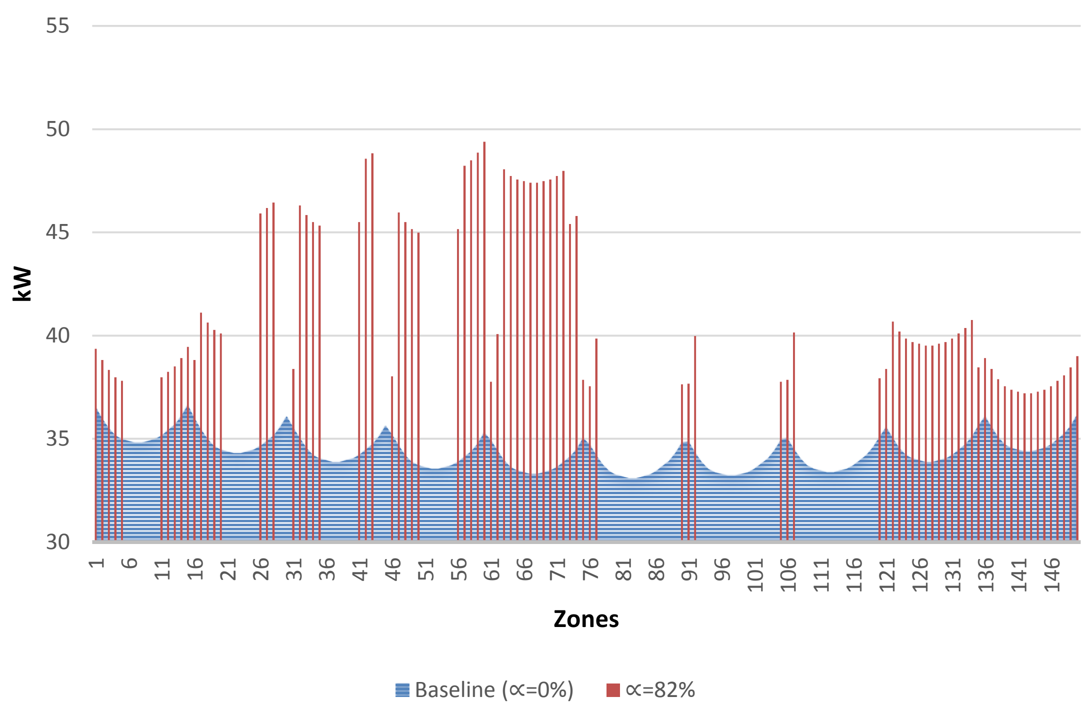


Fig. 82. Baseline Energy vs. Energy with User Preference α = 0.82 (Model #2)

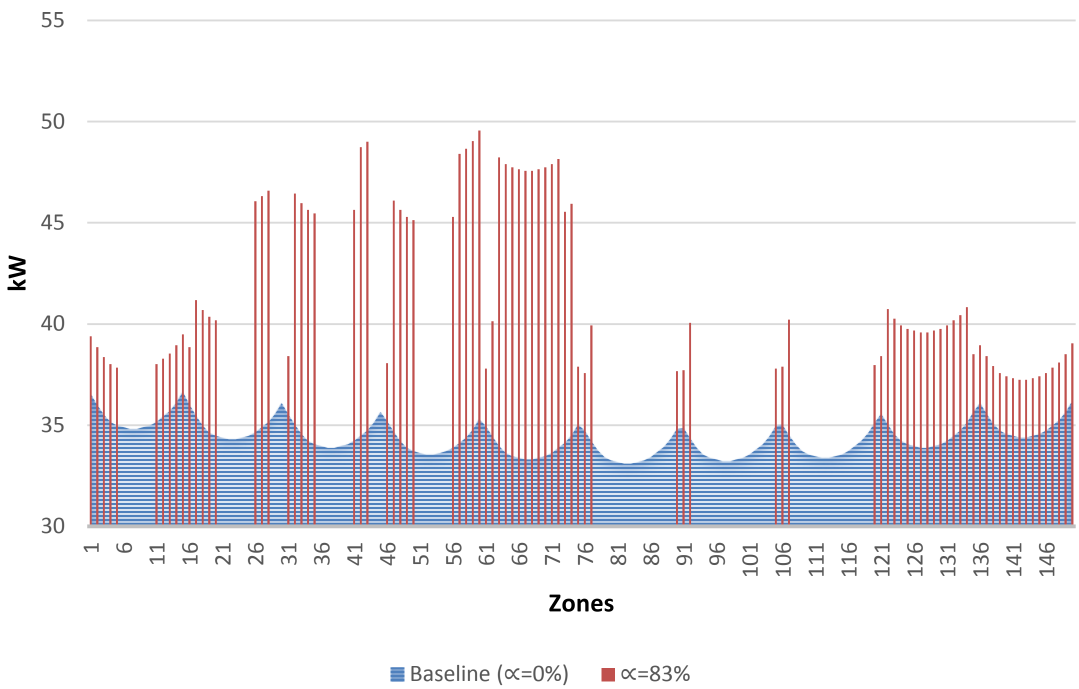


Fig. 83. Baseline Energy vs. Energy with User Preference α = 0.83 (Model #2)

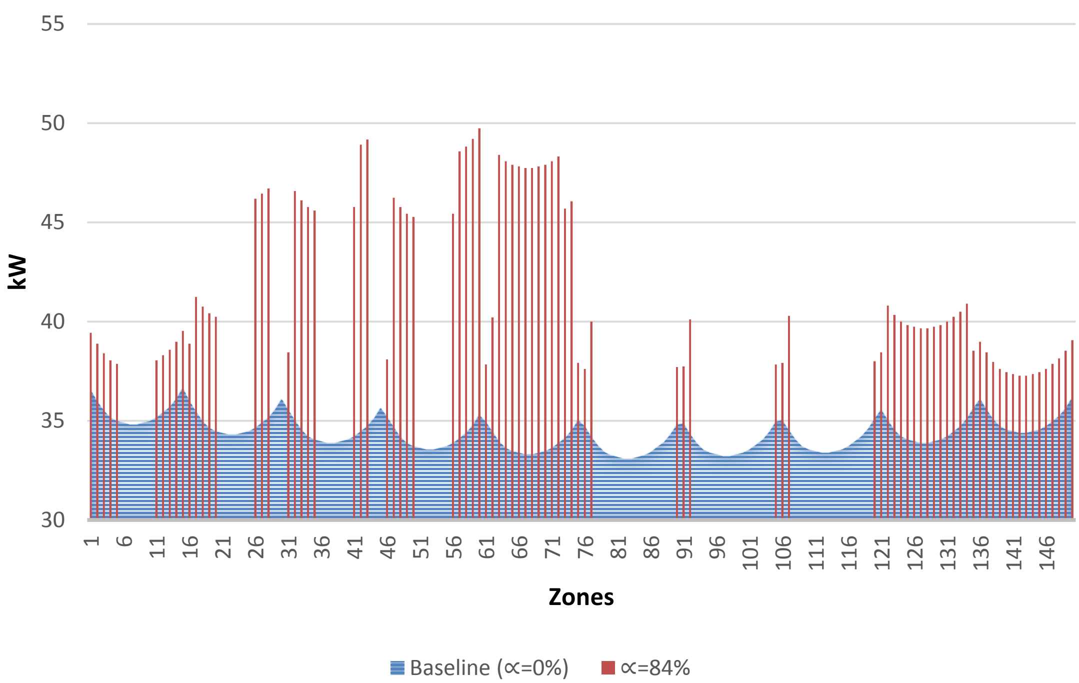


Fig. 84. Baseline Energy vs. Energy with User Preference α = 0.84 (Model #2)

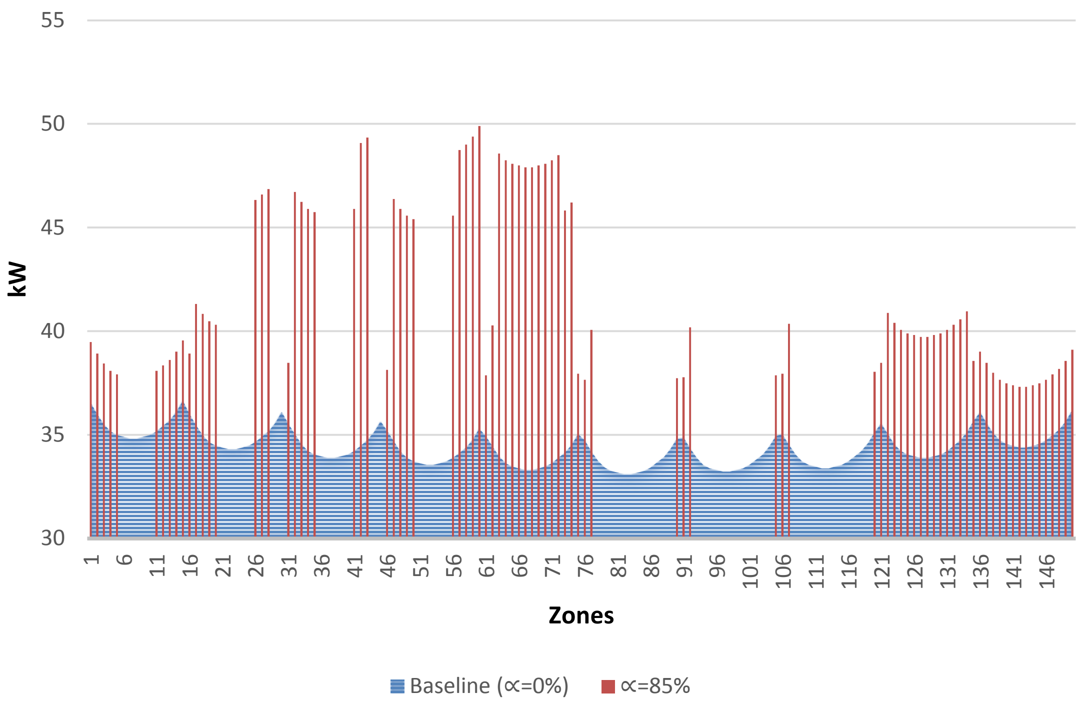


Fig. 85. Baseline Energy vs. Energy with User Preference α = 0.85 (Model #2)

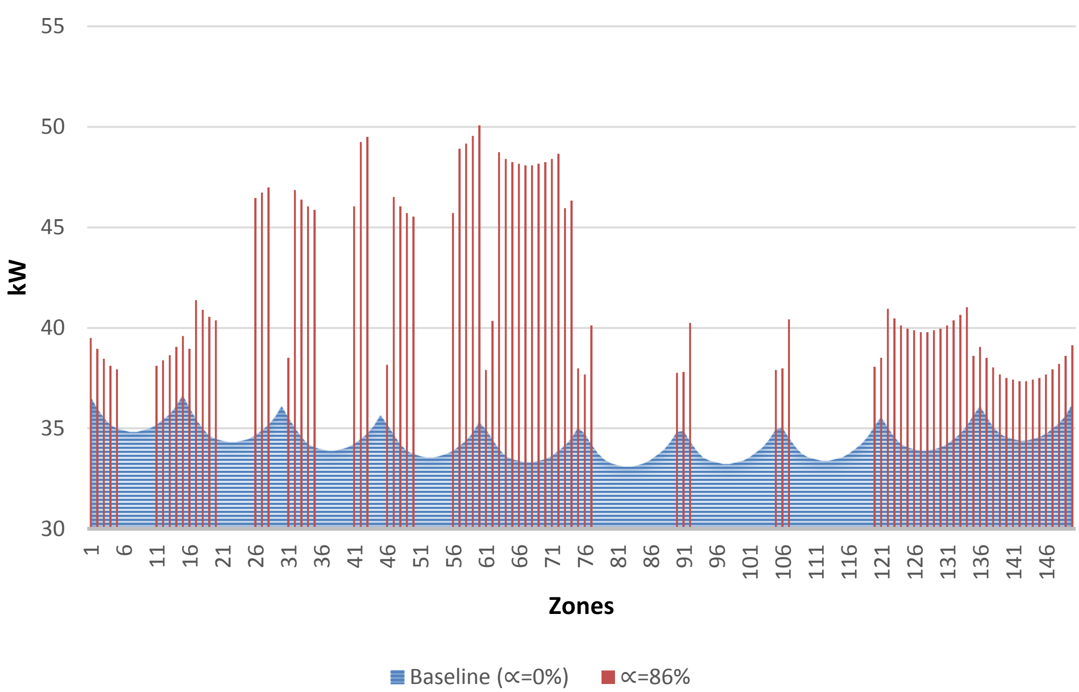


Fig. 86. Baseline Energy vs. Energy with User Preference α = 0.86 (Model #2)

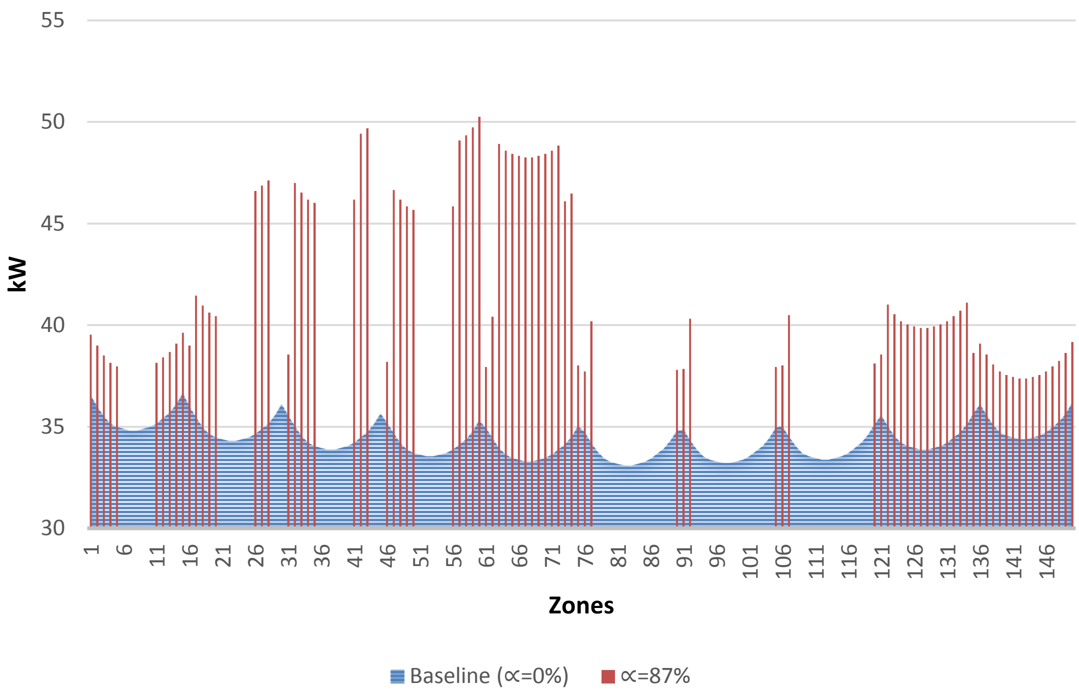


Fig. 87. Baseline Energy vs. Energy with User Preference α = 0.87 (Model #2)

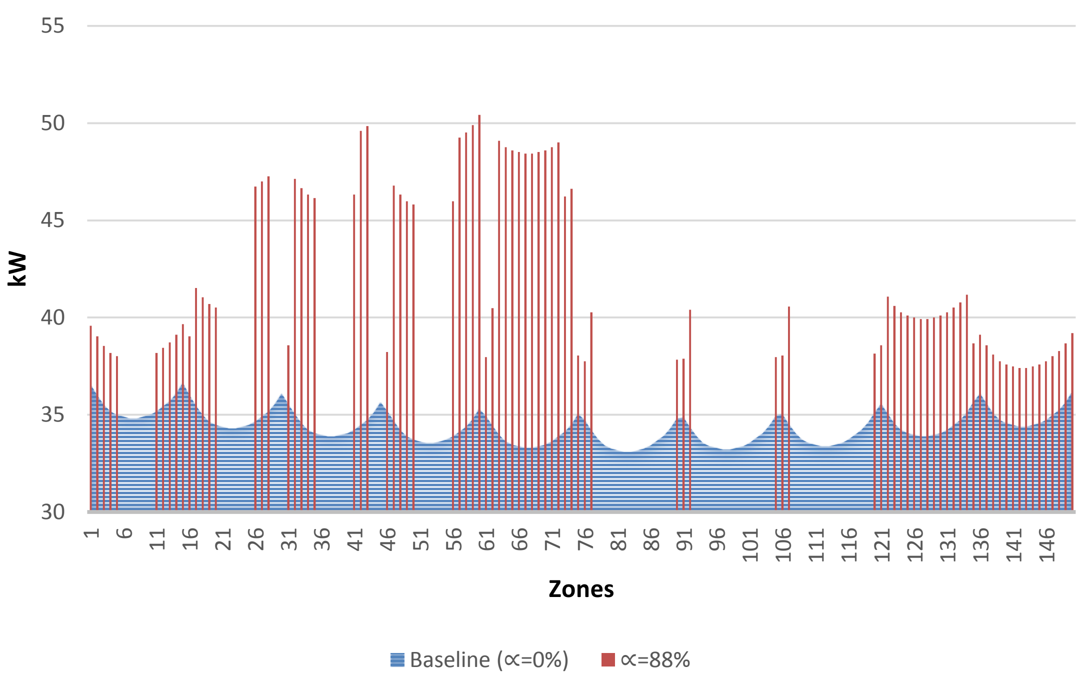


Fig. 88. Baseline Energy vs. Energy with User Preference α = 0.88 (Model #2)

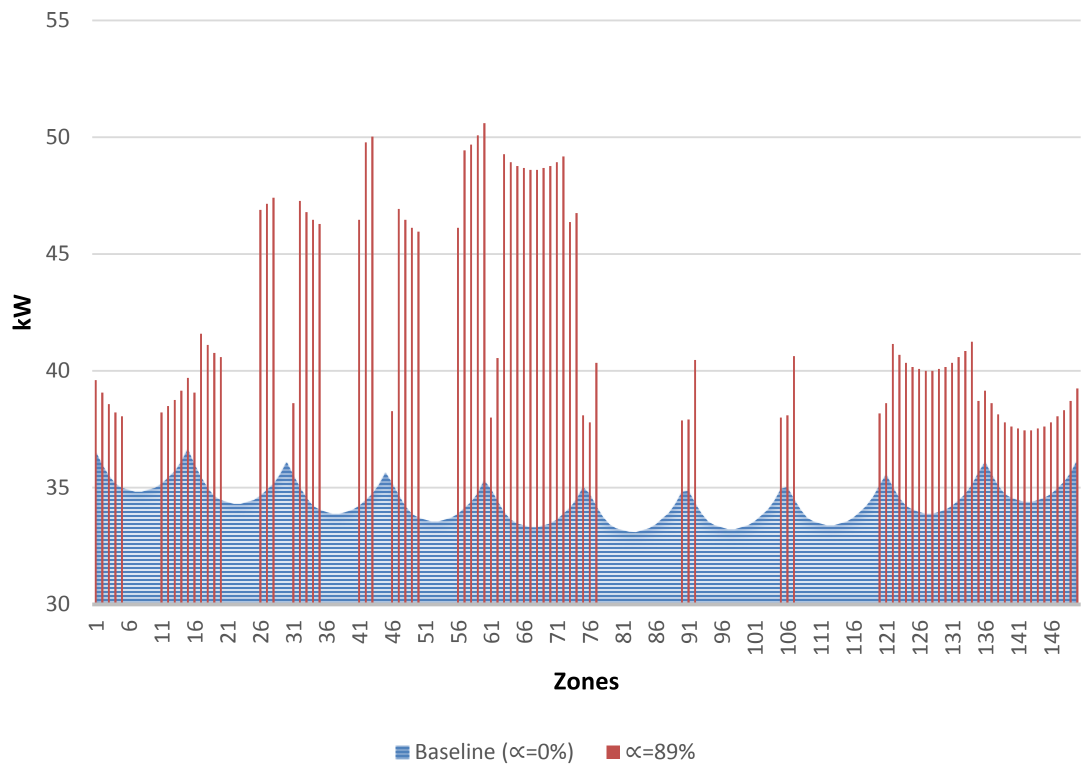


Fig. 89. Baseline Energy vs. Energy with User Preference α = 0.89 (Model #2)

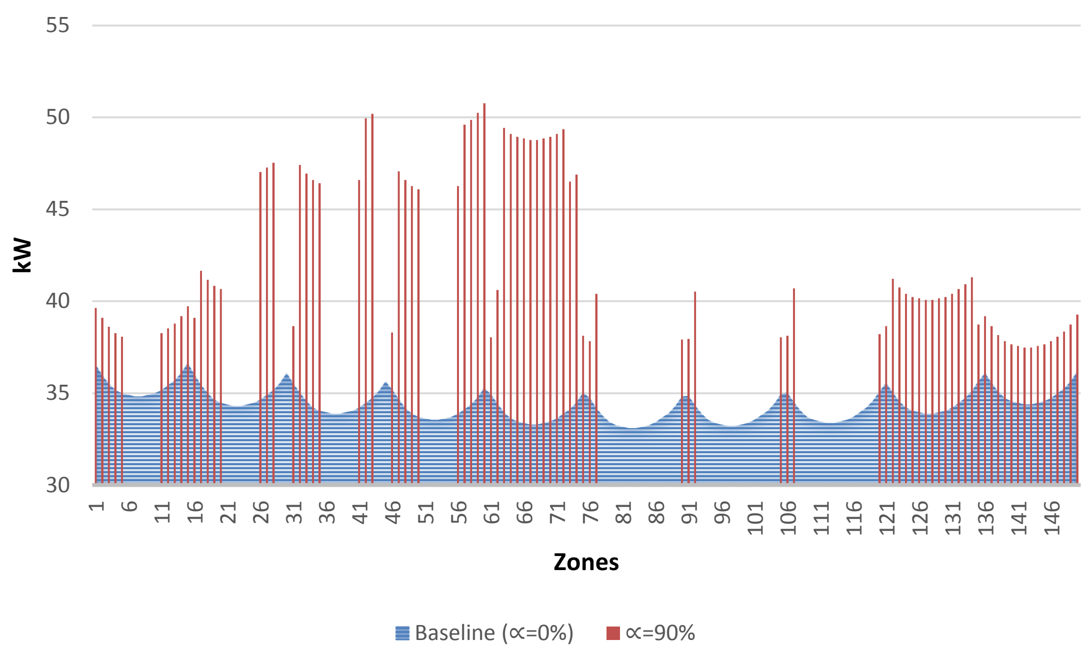


Fig. 90. Baseline Energy vs. Energy with User Preference α = 0.90 (Model #2)

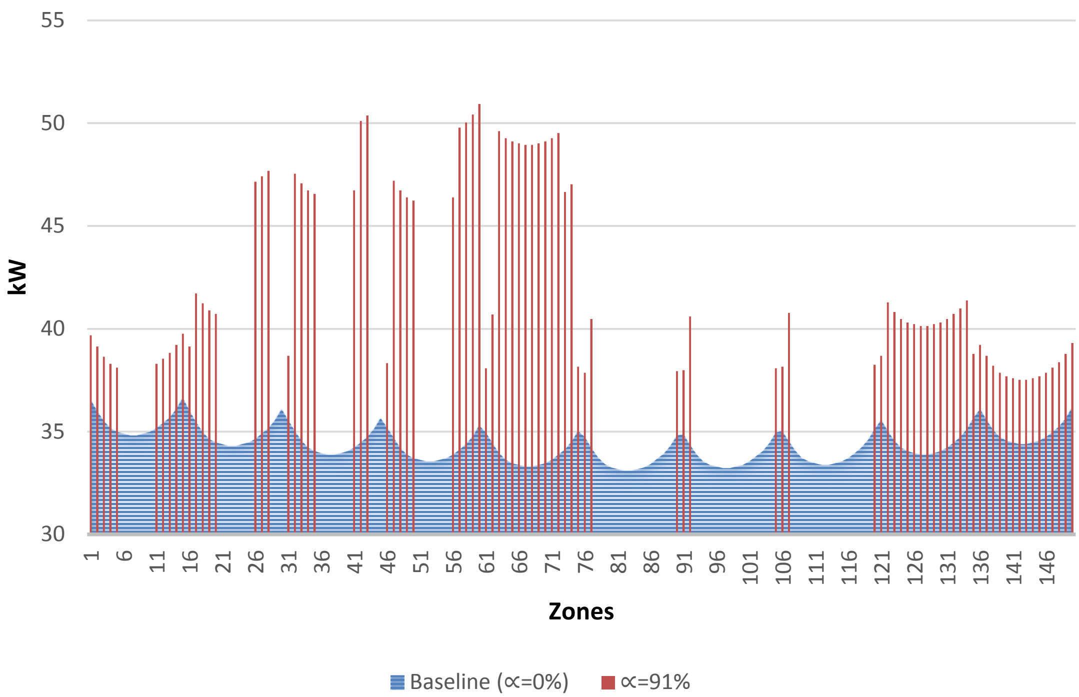


Fig. 91. Baseline Energy vs. Energy with User Preference α = 0.91 (Model #2)

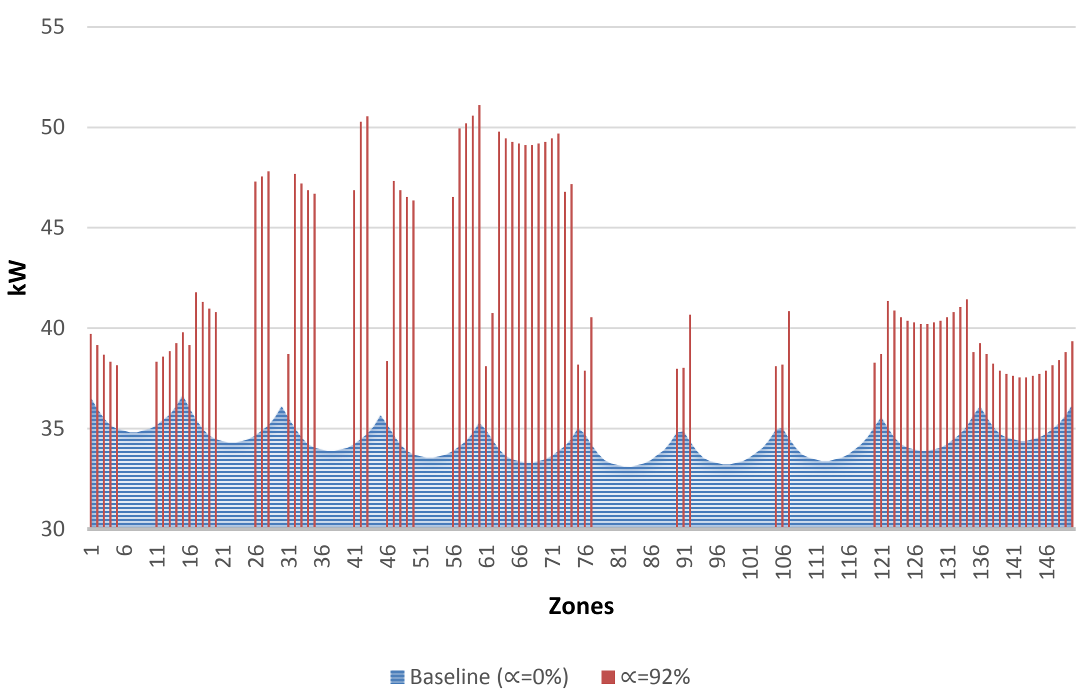


Fig. 92. Baseline Energy vs. Energy with User Preference α = 0.92 (Model #2)

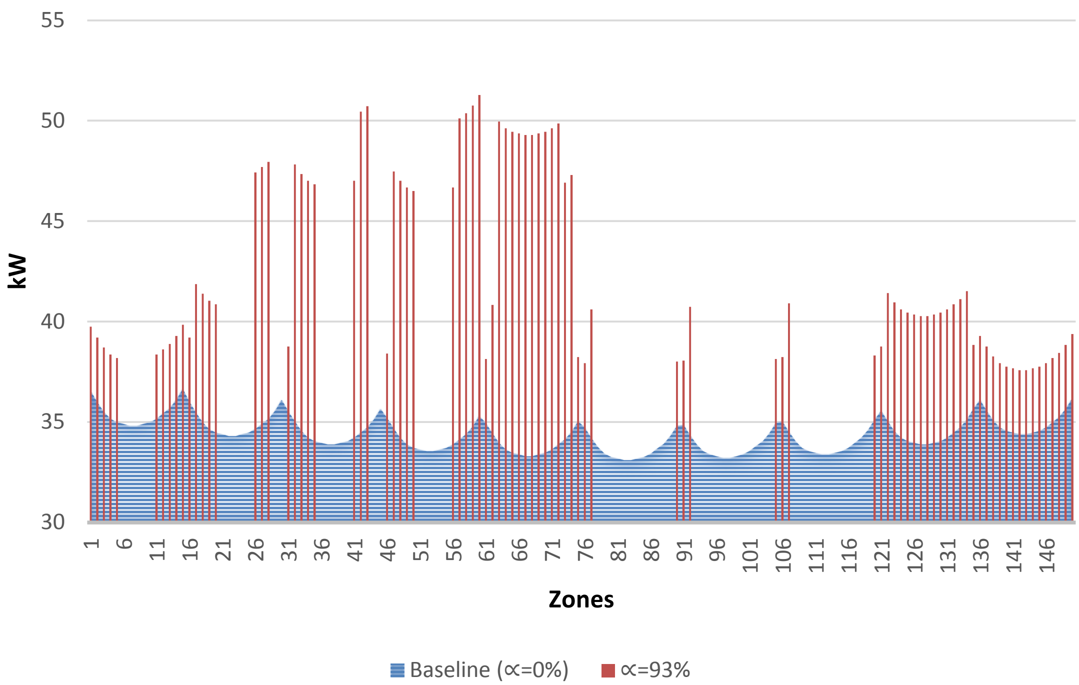


Fig. 93. Baseline Energy vs. Energy with User Preference α = 0.93 (Model #2)

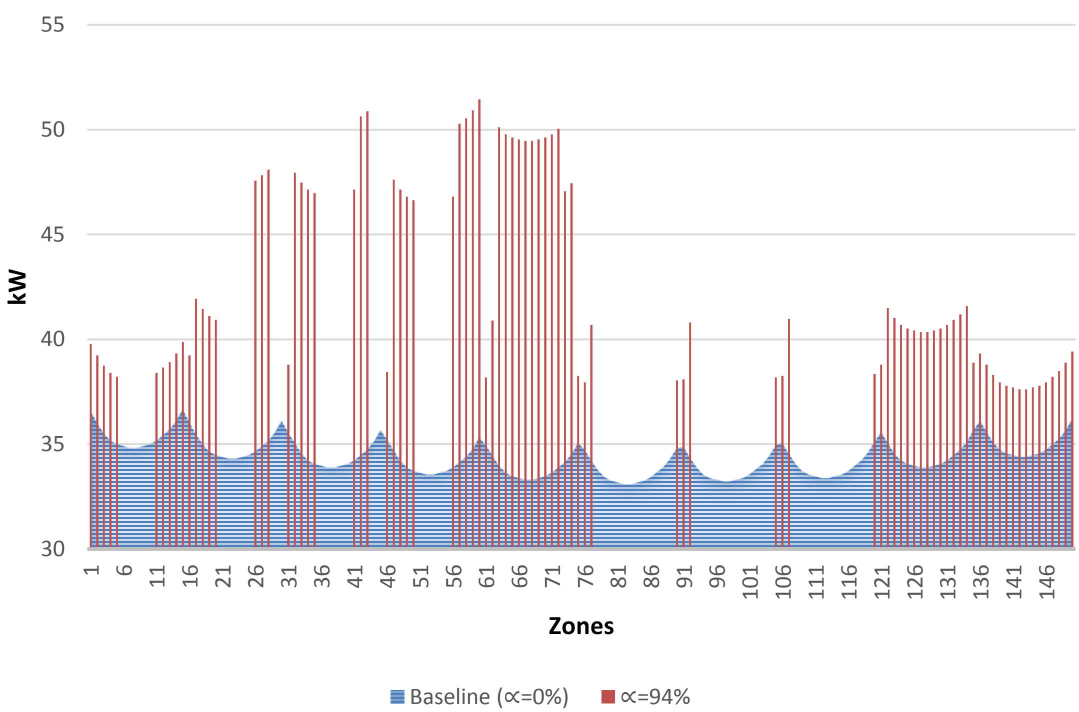


Fig. 94. Baseline Energy vs. Energy with User Preference α = 0.94 (Model #2)

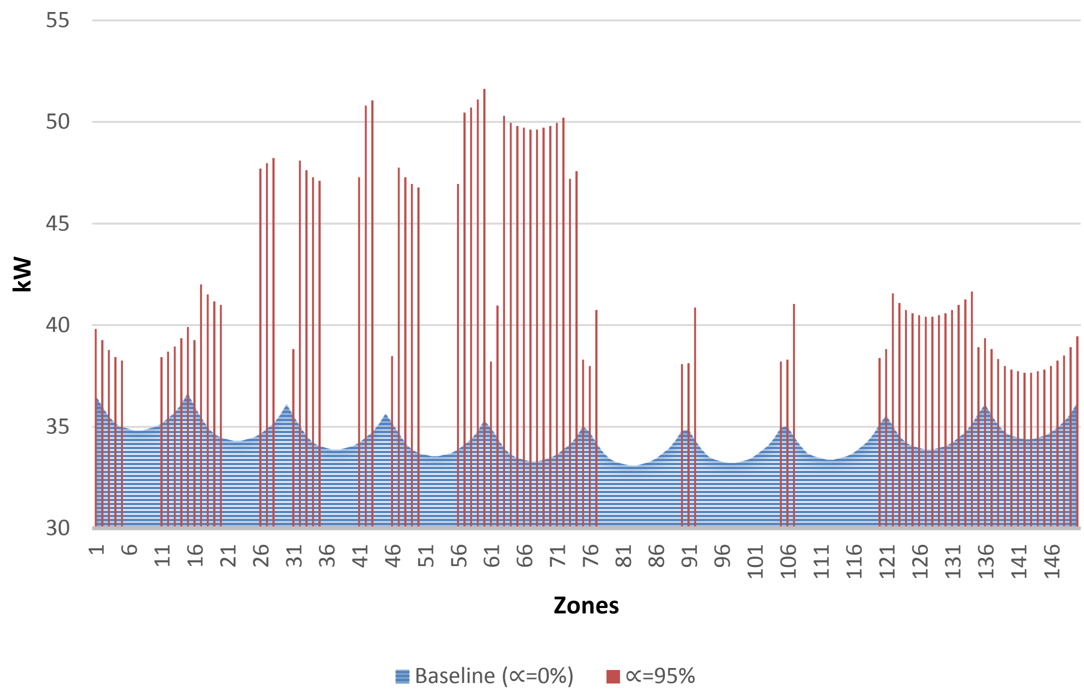


Fig. 95. Baseline Energy vs. Energy with User Preference α = 0.95 (Model #2)

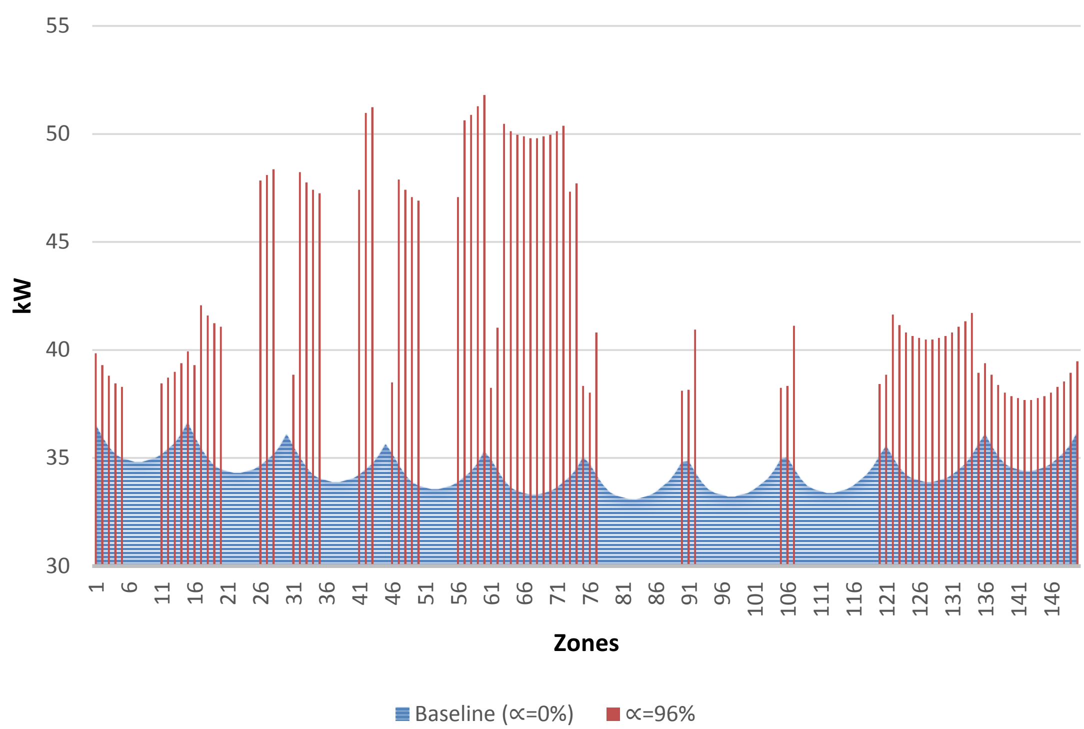


Fig. 96. Baseline Energy vs. Energy with User Preference α = 0.96 (Model #2)

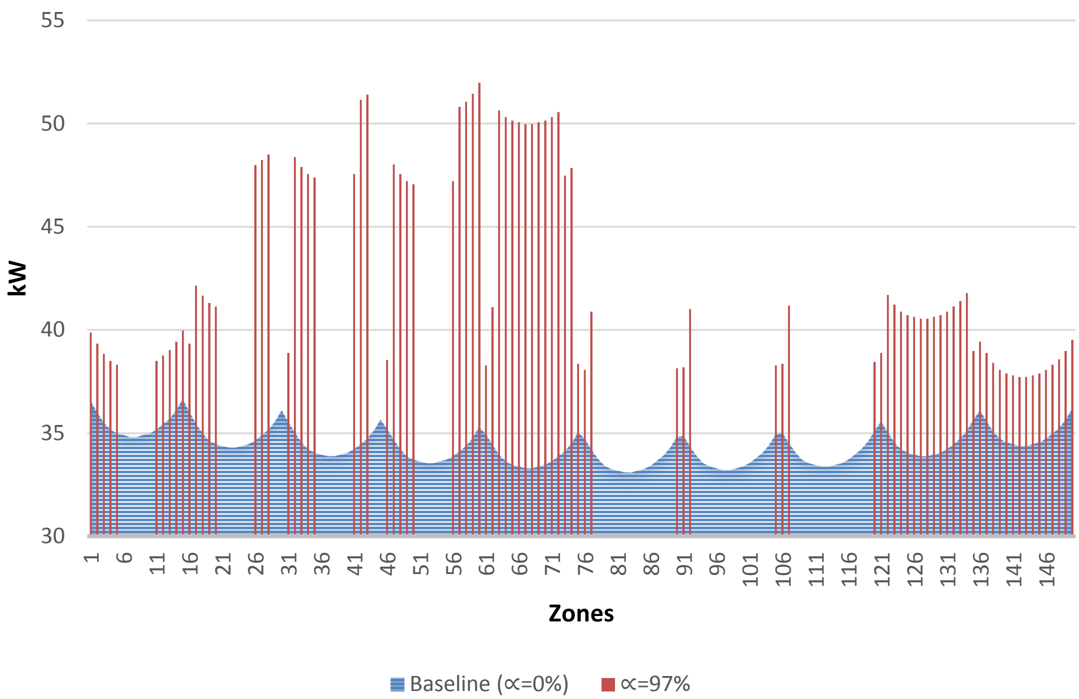


Fig. 97. Baseline Energy vs. Energy with User Preference α = 0.97 (Model #2)

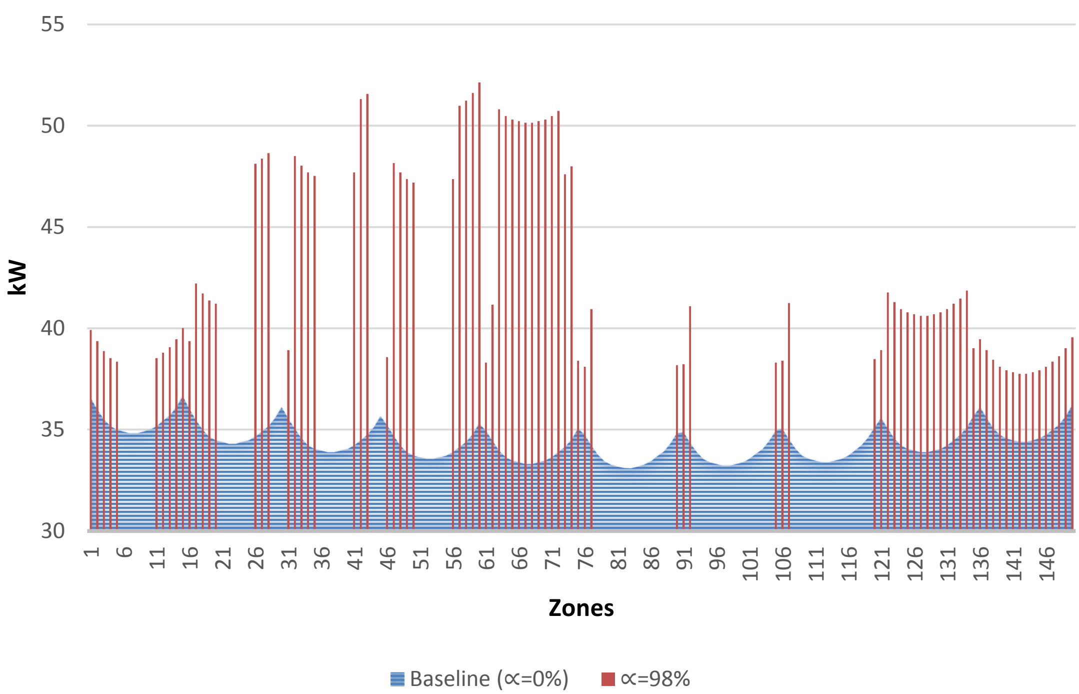


Fig. 98. Baseline Energy vs. Energy with User Preference α = 0.98 (Model #2)

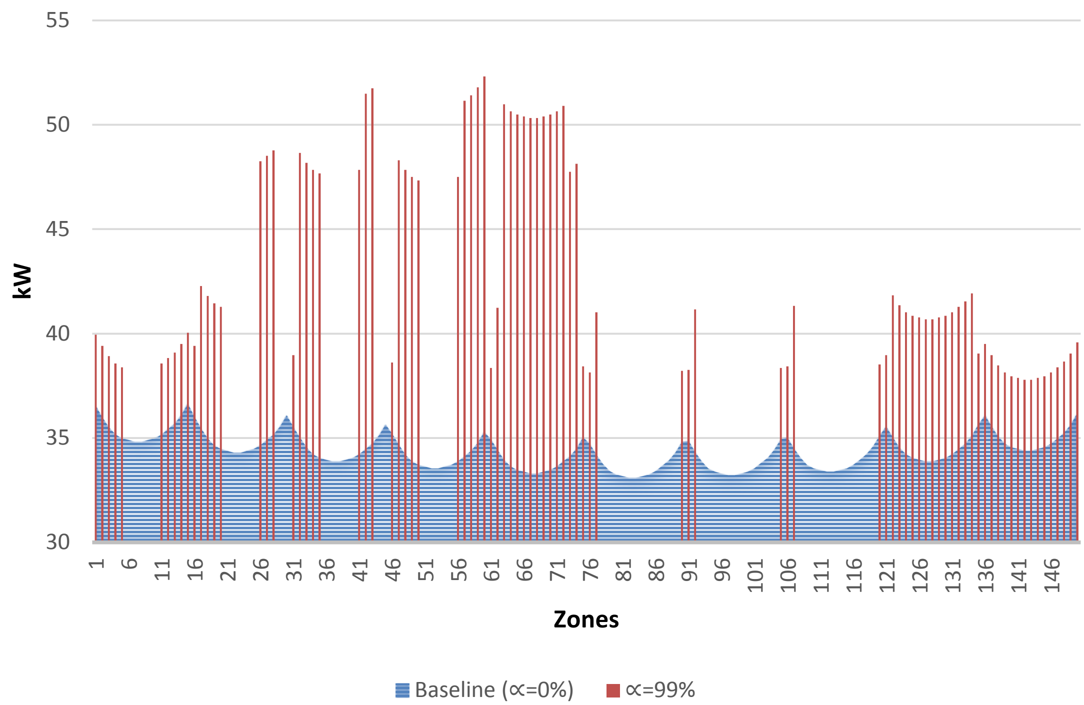


Fig. 99. Baseline Energy vs. Energy with User Preference α = 0.99 (Model #2)

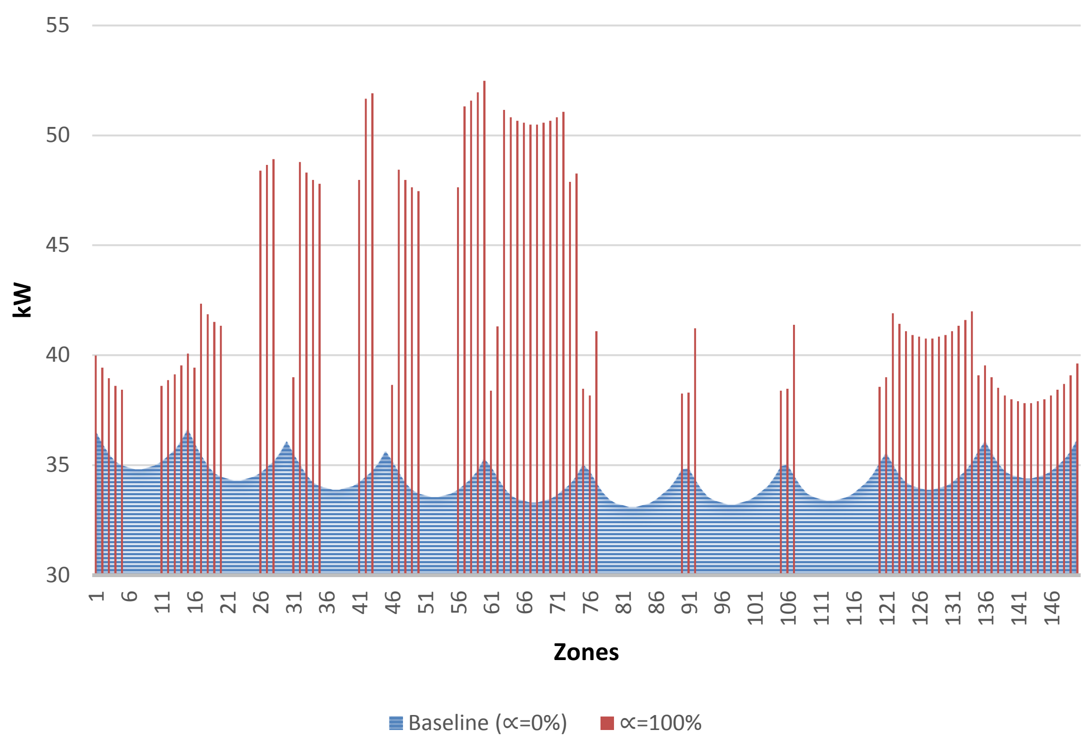


Fig. 100. Baseline Energy vs. Energy with User Preference α = 1 (Model #2)